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1957 SAE TRANSACTIONS

New Drive Lines For New Engines

William P. Michell, Dana Corp.

This paper was presented at a meeting of the SAE Metropolitan Section, New York, Nov. 20, 1956.

WHETHER for new or old engines there is a continuing battle going on in transmission and axle concepts. With a recheck of field use and economics of the engine and drive line in freight hauling, together with a study of new engines available, we may be able to outline features leading to better designs.

Most of our data will come from riding loaded truck-trailers with suitable instruments, checking the why, where, when, and how.

Engines

For any 4-stroke-cycle engine let:

P = Brake mean effective pressure, psi
N = Rpm
D = Displacement, cu in.

Then:

$$\text{bhp} = \frac{PND}{792{,}000}$$

All efforts toward more power must turn on one of these three variables.

P is a function of fuel, breathing, combustion, and friction. It has been continuously improved but is reaching the point of diminishing returns.

N is roughly limited by piston speed, which affects life of bores, rings, bearings, and the like, as well as friction losses.

Increasing D is the expensive way to get power, since displacement governs the weight and cost.

Piston speed, which limits N, is $\frac{2N \times \text{stroke}}{12}$. Then

THE selection of transmissions for trucks, with particular emphasis on how they affect the performance of the vehicle, is covered in this paper—the L. Ray Buckendale Lecture for 1956.

The author also appraises, briefly, the automatic transmission for truck use.

He foresees early progress in this direction for short haul, city, and off-highway units, where shifting and clutch problems are most acute. He expects, however, that it will be some time before they are adopted for highway freight vehicles.

Table 1—Averages for 6- and 8-Cyl Truck Engines

	Average of 36 6-Cyl Truck Engines	Average of 18 8-Cyl Truck Engines
Bore/Stroke Ratio	0.885	1.085
Displacement, cu in.	511.	357
Maximum Torque, ft-lb	413 at 1294 rpm	321 at 2078 rpm
Torque/Displacement	0.809	0.9
Maximum Bmep, psi	123	136
Horsepower at Maximum Torque	102	127
Maximum Horsepower	177	184
at Governed Rpm	2794	3789
Hp/Displacement	0.346	0.515
Bmep at Governed Rpm, psi	104	112
Piston Speed at Governed Rpm, fpm	2340	2255

to raise N and use our displacement more often we must reduce the stroke, increasing the bore to keep D constant (increase the bore/stroke ratio to get more speed).

The 8-cyl engine for the same displacement and bore/stroke ratio has less stroke and can run faster. A trend to higher bore/stroke ratios, however, has come in strongly on the new 8-cyl engines; these engines have the advantage of being shorter than in-line sizes, which is important in truck work; they can stand more bore diameter, without excessive length.

There are really two trends:

(a) Greater bore/stroke ratio.
(b) En-banc eights for shorter length.

Both result in higher speeds.

Table 1 shows the averages for all listed sixes in truck sizes above 248 cu in. and eights above 224 cu in.

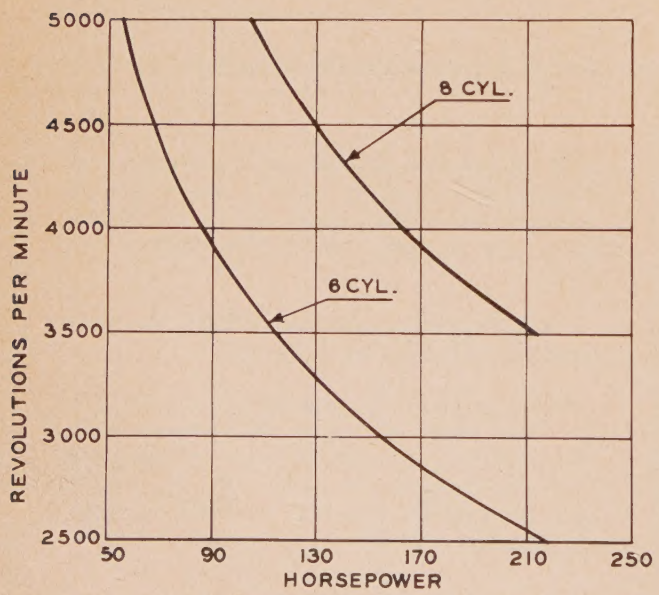

Fig. 1—Speed versus horsepower for 6-cyl and 8-cyl engines, using 1956 average design factors

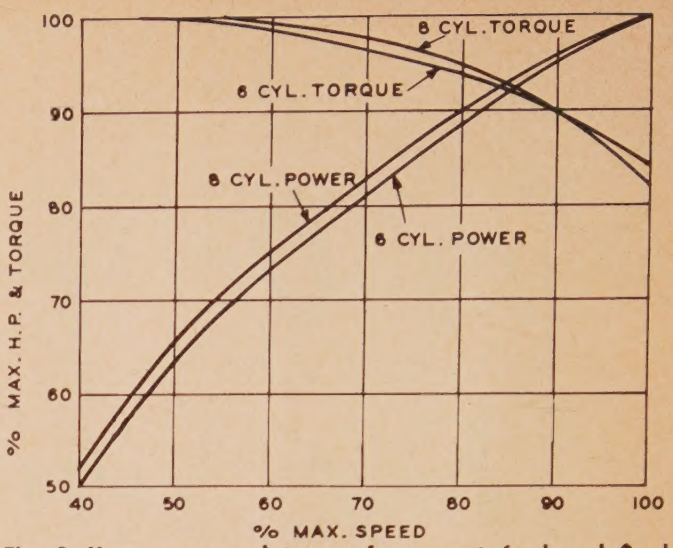

Fig. 2—Horsepower and torque for current 6-cyl and 8-cyl truck engines

Taking these figures as the present state of the art, we can tie horsepower and speed together, using averages for P and bore/stroke ratio. For sixes:

$$\text{hp} = \frac{1.38 \times 10^9}{N^2}$$

For eights:

$$\text{hp} = \frac{2.63 \times 10^9}{N^2}$$

These relationships are plotted in Fig. 1.

Note in Table 1, as regards eights:

(a) The bore/stroke ratio has gone up.
(b) The displacement is down.
(c) The bmep is up, due to better breathing, and so forth.
(d) Horsepower per cubic inch is up.
(e) The piston speed is slightly reduced.
(f) The top speed is sharp increased (power up 4%, speed up 36%).

Table 2—Powers of Ratios

Power											
1	1.18	1.20	1.22	1.24	1.26	1.28	1.30	1.32	1.34	1.36	1.38
2	1.39	1.44	1.49	1.54	1.59	1.64	1.69	1.74	1.80	1.85	1.90
3	1.64	1.73	1.82	1.90	2.00	2.10	2.20	2.30	2.41	2.52	2.63
4	1.94	2.05	2.22	2.36	2.52	2.68	2.86	3.04	3.22	3.42	3.63
5	2.28	2.46	2.70	2.92	3.17	3.42	3.71	3.98	4.32	4.65	5.0
6	2.70	2.99	3.30	3.64	4.01	4.40	4.83	5.29	5.79	6.33	6.9
7	3.18	3.58	4.01	4.51	5.05	5.61	6.26	6.98	7.76	8.61	9.5
8	3.76	4.30	4.91	5.59	6.35	7.21	8.16	9.22	10.4	11.71	13.1
9	4.44	5.15	6.00	6.91	8.00	9.2	10.6	12.2	13.9		
10	5.23	6.19	7.31	8.59	10.	11.8	13.8				
11	6.15	7.41	8.90	10.6	12.6	15.1					
12	7.29	8.92	10.9	13.3							
13	8.58	10.7	13.3								
14	10.10	12.8									
15	11.9										
16	14.2										

Powers of ratios higher than 1.38 can be found by reading lines 2, 4, 6, 8, 10.

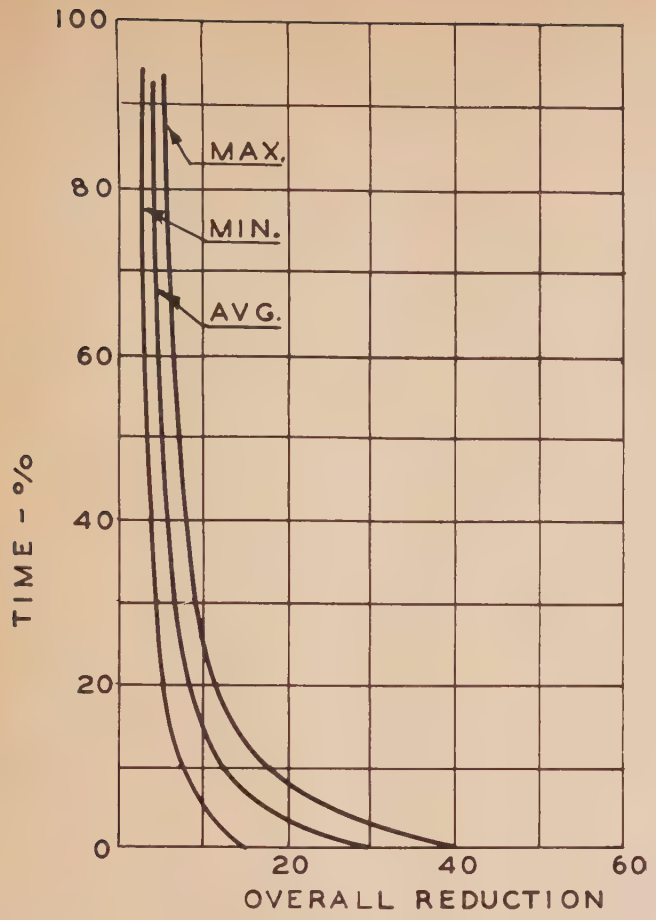

Fig. 3—Overall reduction versus per cent time more reduction required

Horsepower at any speed will be delivered to the tires through gears which bring the speed down to the legal or economic norm.

From a transmission point of view, we are concerned most with the shape of the horsepower curves.

Fig. 2 gives the per cent horsepower delivered at all speeds, by the current 6- and 8-cyl engines, as shown in the averages (Table 1).

While the typical eight runs much faster than the six, its speed is reduced by the rear-axle gear so that percentage plotting for the two engines in Fig. 2 gives us a true picture of engine horsepower as geared down.

Nothing in the shape of the curves indicates notable differences in truck performance, giving proper gear setup. Road tests, including records of average speed and of time spent in various gears on test runs, confirm this view.

The Road

Total grades and their severity govern the design of the transmission. Contour or grade maps of all highways can be bought from state highway departments, but effective grades are hard to appraise from maps. On light, rolling grades the truck speed may be little affected due to the high speed reached downhill.

Time in various gears in actual hauling has been recorded for many years. I discovered that plotting such data in the form of probability curve (Fig. 3) results in generally useful information. Translating the data into apparent gradeability required (Fig. 4) permits use for any combination on the road. A trial set of gear ratios for a given truck can be turned into gradeability for each ratio and the time in each gear can be found from the general curve. This can be used to find time loss due to inadequate ratios, and to calculate life of gears and bearings.

Such road record curves are implied in the design of every transmission. Unfortunately, we cannot design for the average curve. Some compromise closer to maximum severity is usually used to avoid failures.

The Job

Transmissions could be tailored to every individual hauling job; a wide variety of gear combinations and sizes do exist.

To bring out the problems we will work with a common unit, a highway hauler, most widely used. The

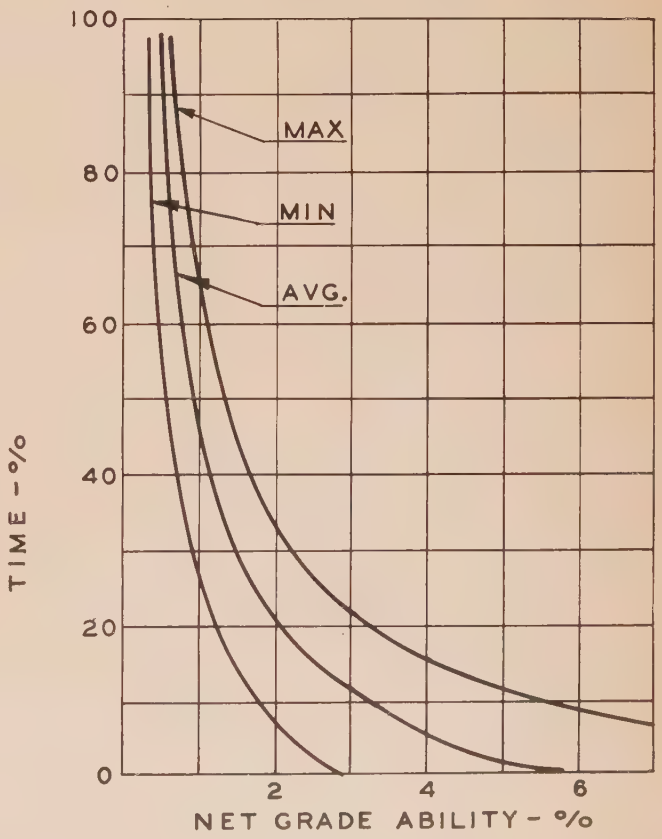

Fig. 4—Per cent net gradeability versus per cent time more gradeability required

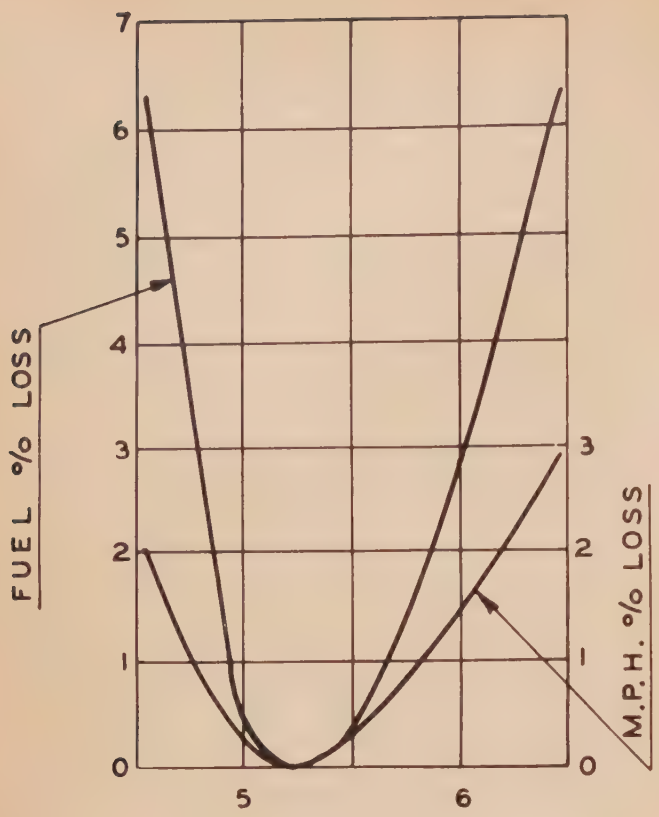

Fig. 5—Axle ratio versus fuel and speed

following specifications will come close to typifying this field:

Gross Combination Weight, lb	55,000 (max legal in many states)
Engine Hp	150 net
Tires	10.00 x 20
Maximum Level Speed:	
Flat Bed (with 50 sq ft frontal area), mph	56
Van (with 100 sq ft frontal area), mph	48

Basic Concepts—Gear-set Ratios

Factors which concern the transmission man will be defined next.

Ratio: a number expressing torque multiplication or speed reduction (overdrive is a fraction such as 0.8).

The confusion between lower, higher, low numerical, and high numerical ratios can be avoided by calling the ratios faster or slower. A fast axle ratio might be 4.5, a slow one 8.

Ratio step: the slower ratio divided by the faster. If a 2-speed axle has ratios of 6.0 and 8.0 the step is 1.33 (and similarly for a transmission).

Splitter ratio(s): If the transmission has ratio steps of 1.77 a 2-speed axle or auxiliary with a step of 1.33 will "split" the box ratios and double the number of speeds.

Progression is:

Low-box	Low axle
Low-box	High axle
Second-box	Low axle
Second-box	High axle

and so forth

The splitter step here should be the square root of the main box step; with the 3-speed auxiliary, its step would be the cube root.

Tachometer driving: Practically all truck engines have an optimum, best cruising or governed speed and a minimum permitted speed. Let the desired top speed be 100%; when the engine slows down by the amount of the ratio step (to 75% if the step is 1.33) the driver should downshift, getting back to full speed.

Lugging: For any driving condition where a further opening of the throttle results in no road speed increase, the result is short engine life.

Starting: Empirically and by test it has been established that about 10% net gradeability is necessary to start a truck or truck-trailer with an ordinary clutch.

Grades: Over 6% grades are seldom found on main roads; adequate starting on such a grade would call for 16% net or real gradeability (10% plus 6%). This is a fair figure for an average rig as designed at the factory.

Overall range: Again empirically the overall ratio for highway truck-tractor-trailer rigs is about 10/1. This is most commonly obtained by a 5-speed transmission with, say, 7.5 range and an axle with a 1.33 step. If an overdrive transmission is used, the low ratio divided by the high gives the range. Prevalence of this 10/1 range is due largely to availability of commercial units. The bulk (60-80%) of the units we are discussing will have about this gear range; some operators who do not need them specify 2-speed axles with a view to later resale value. Higher power in the engine reduces the spread or overall range of the transmission. High speed has no effect.

Let:

N = Engine speed at peak hp
T = Engine torque at peak hp
R_H = Top gear ratio (overall)
R_L = Low gear ratio (overall)

Some maximum road speed is required. This is a function of power, not gears; but the suitable gear must be available.

$$\text{Maximum road speed} = \frac{N \times 2\pi \times \text{tire radius}}{R_H}$$

Some maximum gradeability is required in low, at full engine speed:

$$\text{Maximum gradeability} = \frac{T \times R_L}{\text{tire radius} \times \text{gross weight}}$$

All the factors in these equations are set in advance except engine and gear ratios, covered by factors N,

T, R_H, and R_L. Now multiply the two equations together and lump all the constants into K.

$$K = \frac{N\,T\,R_L}{R_H}$$

$\frac{R_L}{H_H}$ = Transmission range (lowest overall ratio divided by highest)

NT = Torque times speed, or power

Then our equation is:

$$\text{hp} \times \text{range} = \text{constant}$$

Low power requires high range and vice versa; engine speed does not enter in the range requirement.

Range may be obtained in main box, auxiliary, 2-speed axle, or torque converter combinations, but it must be adequate. The above formula is put to practical use when a high horsepower engine comes in; the range is reduced by the use of a faster low gear; this permits upping the rating of the gearbox, since tooth loads are benefited. It could also permit closing up the ratio steps, or the use of fewer steps.

It would be interesting to tabulate successful service constants using this formula, for every service.

Axle Ratio

The selection of axle ratio is most critical; with a "fast" ratio we cut down the engine revs per mile and gain fuel economy but, of course, lose gradeability in high gear where we spend most time. Then minor grades call for gearshifting; the fast axle can be carried too far, fuel-wise.

A cure for this condition would be a very close step next to high; one large truck builder likes a combination of gearbox and axle giving top speeds of 60 in high, 50 in the next step, and 40 in the next step. Running on level at 50 mph in high gear, the engine is making only 5/6 of maximum or governed speed; when light, adverse grades are encountered the next (50-mile) gear gives more gradeability and keeps the engine loaded for good economy. This 60- to 50-mile step is 1.2 in the gear step, unusually close but very desirable.

Fig. 5 shows the results of road tests on one run with varying axle ratios with a 10-speed drive line. The step next to high was 1.3. This shows the critical nature of the axle ratio. The comparative road test seems to be the safest method of selection.

Ratio Steps

From the power-speed graph (Fig. 2) for our high-speed engine we can construct a graph of power delivered through the ratio steps. If the step is 1.5 next to high the truck will enter this gear with the engine at 67% of top speed and accelerate to 100%. The usable power can be found from the mean of the engine power curve from 67% to 100%.

This graph of the effect of ratio steps on our average high-speed eight is shown in Fig. 6.

This shows a constant gain in usable power with closer steps; road tests continually verify this.

In practice the gear steps in the past have run from 1.25 to 1.40 for gas jobs and as low as 1.18 for heavy diesel rigs. The less droop in the horsepower curve the greater the gain from close steps. New eights have considerably more droop than sixes.

Comparing the ratio step performance of our fast eight with that of a slower engine with less droop in the horsepower curve in Fig. 6, we see that a 1.33 ratio step on the new engine delivers as much power as a 1.28 step on the older engine. A 1.56 step is as good on the new engine as 1.47 on the old. Close steps are, however, still advantageous with new engines.

Reducing the engine speed by governor setting would cut out part of the drooping end of the curve and make the new engines act more like the older ones, ratio-wise. Such a reduction usually occurs, in practice; we are dealing here with the engine maker's rating, usually set fast to show maximum horsepower for sales reasons.

Limits to finer steps are:

1. Cost and weight of the transmission. A 10 to 1 gear range with 1.2 steps, for instance, would require 13 steps, 14 ratios (see Table 2).

2. Physical ability of the driver. Many road grades vary to the extent that they could be called vertical curves. A driver continually shifting to suit such grades will finally give up, exhausted, and run in some slower gear which is sure to get him over the hill.

3. Time lost due to shifts. The gain due to short time use of a ratio may be lost in the time of shifting—2 or 3 sec. Evidently an automatic transmission would wash out the fatigue and time loss factors.

There is, none the less, a constant demand for the

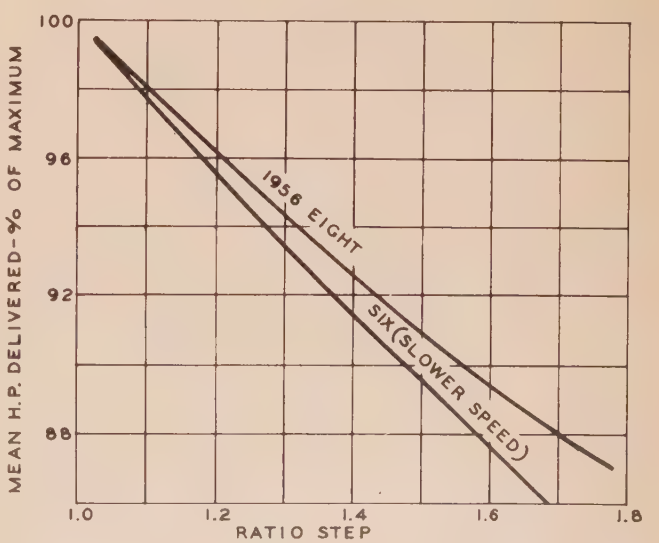

Fig. 6—Mean power delivered through ratio steps for 1956 8-cyl truck engine

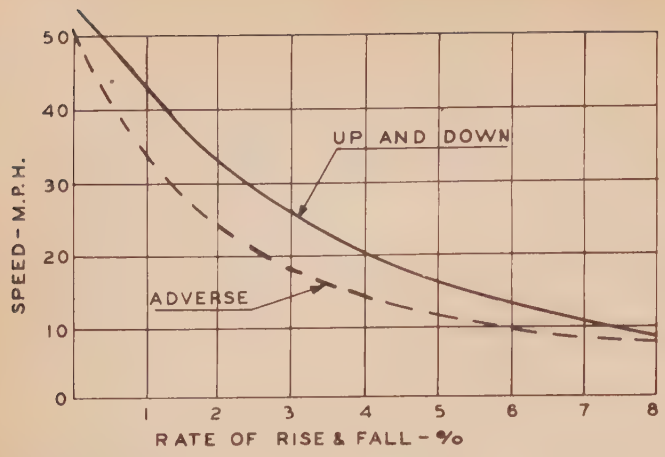

Fig. 7—Hill speeds for truck of 55,000-lb gcw, 150 hp net, and 50 sq ft front area

closer steps. Huge amounts of study and testing have gone into designs for close ratios. Some typical solutions follow. By consulting Table 2 one can readily find real numbers.

Current Practice—Gear Steps

(A) Five-speed transmission 1-S^2-S^4-S^6-S^8 and 2-speed axle 1-S, giving 10 equal steps from 1 to S^9. This is the most widely used.

(B) Same with 2-speed auxiliary. This may have 1-S-S^2-S^3-S^4 in the main box and 1-S^5 in the auxiliary, and likewise with 4-speed main box.

(C) Four-speed transmission with 2-speed axle giving eight equal steps from 1 to S^7. Used considerably on the West Coast for very short tractors.

(D) Main box, five speeds, 1-S^3-S^6-S^9-S^{12} with 3-speed auxiliary 1-S-S^2, giving 15 speeds with step S, low of S^{14}. Widely used on very heavy combinations.

(E) Same with 4-speed main box, 12 speeds. Similarly used on heavy highway combinations.

(F) Main box, with five speeds and auxiliary with four speeds, giving 20 evenly stepped speeds, recently introduced.

(G) A newly demanded combination has a 5-speed main box and an auxiliary with three splitter ratios plus a deep low. This gives 15 speeds for the highway plus five very low speeds for off-highway operation.

These combinations are all successful; we recognize, though, that the progression is locked to even steps by the use of 5 x 2, 4 x 3, or similar multiplications. Any attempt to vary the steps will come out rather badly.

There is less need of close steps at the slow end, so these rigid formulas involve some wasted steps.

A 5-speed main box and 2-speed axle combination is usually driven as an 8-speed unit, omitting first-high axle and second-high axle.

Study of Fig. 3, which shows most of the truck time in top gears, and actual experience show that very close steps near the top speed and wider steps at the low end would be a desirable solution.

Some new attempts along these lines, still in the early road test stages, are:

(H) Main box 1-S-S^2-S^4-S^7, two-speed axle 1-S^3 ($S^3 = 2$), giving six close steps from 1 to S^5, two lower steps S^7, S^{10}.

(I) Main box 1-S-S^2-S^3-S^6, two-speed axle, 1-S^4 ($S^4 = 2.5$), giving eight close steps from 1 to S^7, plus one low step, S^{10}.

In each case (G and H) the main box is shifted down through the close steps, then the axle is shifted down, then all main box steps including the lower wide steps are used. This permits close steps in the high end, wide steps at the bottom, and cuts out most of the axle shifting, simplifying driving.

Another 2-speed axle combination is:

(J) Main box 1-S-S^2-S^3, axle 1-S^4 ($S^4 = 3.5$).

This gives eight even speeds; the top four, controlled in transmission, cover must ordinary road work and could readily be gathered to give closer steps near the top. Gathered steps appearing again in the low range would look rather odd, but the low range is seldom used except in starting.

(K) Main box with close or gathered steps plus torque converter.

We can have:

Main box 1-S-S^2-S^3-S^n ($S^n = 5$), torque converter 1-2 (or 2.5 or 3) stall torque.

The torque converter, with main box in low, handles all work below 5 to 1, and locks up or goes into the coupling phase in main box ratios faster than 5 to 1.

The sequence in "K" uses the converter in the only way it can be used to advantage and is the choice in some designs of automatics. One new automatic now uses the arrangements with six main box speeds:

1 to S^5, where $S = 1.39$ and $S^5 = 5.29$; converter stall torque is 2.8.

Another converter job, hand shifted, uses a 5-speed main box with about 5 to 1 low but with speeds gathered closer near the top rather than even steps, a desirable arrangement.

The above list by no means exhausts all the gear combinations in use; this entire paper could be used up in discussion of the pros and cons of the various types. Standards minded engineers look for a trend and there is no trend, except to greater diversity.

Functioning is finally the criterion, and I believe the combination which provides close steps, fast easy shifts, and reliability will endure.

One key to success is the driver; rigorously to shift even 1.33 steps for 8 hr in hills calls for a lot of work on any hand-shifted job. A story comes in of drivers who can save 8% of their own allowed time with careful shifting, but prefer the loss of time to the fatigue involved; this with a 1.33 step combination.

More modern roads still have hills, but the gradi-

ents do not change every few yards, which will help the close-step manual-shift jobs.

Standard Step Combination

To reduce the inventory at the truck maker's level, a desirable standardization might be:

1. Let all transmissions have the same step "S" between speeds.
2. Let all 2-speed auxiliaries or axles have step $\sqrt{S}$.
3. Let splitter 3-speed auxiliaries have step $\sqrt[3]{S}$.
4. Where greater range is needed use an auxiliary with the above splitter step or steps plus a deep step for range in low.

This would insure that any combination sold would have equal and usable steps. Boxes with overdrive or with a close step next to high would still have the standard step "S" in lower speeds.

Gear Set Appraisal

There are no mysterious reasons for odd steps or wide speed differences in steps in a gear combination. If the steps are not reasonably uniform, poor design or selection can be assumed.

Let a 5-speed box have ratios: $1\text{-}R_4\text{-}R_3\text{-}R_2\text{-}R_1$, and the 2-speed splitter have: 1-S; resulting ratios are: $1\text{-}S\text{-}R_4\text{-}R_4S\text{-}R_3\text{-}R_3S\text{-}R_2\text{-}R_2S\text{-}R_1\text{-}R_1S$.

Dividing each ratio into the one after it gives the step; unless the steps are reasonably close there is something wrong. Three or four per cent step variation should be maximum. An exception can be made for the special setup designed for one or two close steps near the top, with sacrifice elsewhere.

This appraisal gives all the information needed, without bar graphs of road speeds or horsepower curves for each gear ratio.

Unfortunately, we find in use too many bad combinations due to carelessness or use of stock items. A vehicle should not be penalized throughout its life for such reasons.

Performance and Fuel References

Most data showing road test results have been taken from Research Report No. 9-A, Highway Research Board, using some simplification.

The research report data checks well with recent results. New averages of 17 engineering test runs with various engine-transmission combinations show composite results:

Gross Combination Weight, lb	52,225
Speed, Overall, mph	34.43
Speed, Road (average), mph	37
Miles per Gallon	3.82
Engine Hp—Gross	155
Net Hp	140 (est.)

Table 3—Power-Speed and Fuel Cost

(55,000-lb, van—level road)

Road Speed, mph	Rolling Resistance, hp	Air Resistance, hp	Total Hp	Hp-Hr per Ton-Mile	Mpg	Fuel Cost per Mile, ¢
30	45.3	14.5	59.8	0.073	5.8	0.036
40	65.6	34.0	99.6	0.091	4.5	0.045
50	88.8	66.8	155.6	0.114	3.71	0.056
60	114.0	115.0	229.	0.138	3.1	0.068
70	142.0	182.0	324.	0.169	2.5	0.083
80	173.0	272.0	445.	0.202	2.1	0.100

At 0.52 lb per net hp-hr fuel to engine.
Gasoline at 21 cents per gal.
Drive line losses omitted.

The research report uses rate of rise and fall of road as a parameter for fuel and speed.

Checking rise and fall by fuel and by speed shows fair concurrence with this report. Checking gas consumption on p. 39 and road speed on p. 54, we get 1.6 ft per 100 in each case for rate of rise and fall, for our 17 test runs.

Performance

Level Road Speed—There is a limit to the economical size of engine; savings in time are overbalanced by fuel cost if the engine is used to give high level road speeds.

Table 3 shows how our 55,000-lb rig would look with various horsepower engines run wide open on level ground, with a van trailer.

At 70 mph we are spending 0.31¢ per *gross* ton-mile for gasoline, plus other extras for tires and brakes. In the Midwest we may be hauling steel at a rate of 2.78¢ per *net* ton-mile, a one-way load.

Faster truck speeds do not seem attractive.

Considerable work is, however, being done with larger engines plus the use of a chassis speed governor to avoid uneconomic level speeds. This confines the full power of the engine to use on hills and acceleration, where these factors are serious.

Transmission designer can, however, feel assured that great increases in power are not on the way, to change his designs.

Grade Speeds—Overall Average—In Fig. 7 we see the average speeds our unit will make in hilly country of increasing severity.

Distance is measured on the road surface and, neglecting increased air resistance, the energy put in up-hill should be regained down-hill. Actually, of course, the speed becomes excessive and the energy is lost in the form of heat in the brakes.

The grades we are considering are only adverse for half the distance "up and down" in the curve shown. The curve marked "adverse" shows the speed if the grades were all adverse. On very severe grades the up-and-down speed approaches the adverse-grade speed; the down-hill temperature of the brakes is so high that safety considerations demand a slow speed to avoid the chance of losing the brakes by drum

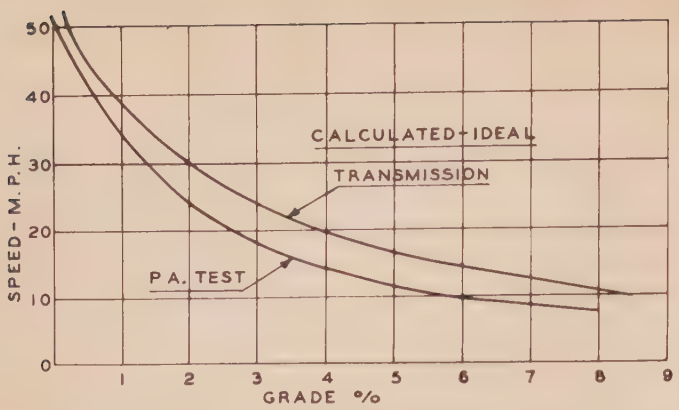

Fig. 8—Hill climbing for truck of 55,000-lb gcw, 150 hp net, 50 sq ft front area. Grades shown are averages of road variables in a given stretch

Table 4—Speeds and Shifts on Various Runs of Truck-Tractor-Trailer Unit

Terrain	Speeds in Gear Set	Shifts per Mile
Rolling	5	1.13
Medium	5	1.51
Severe	5	3.06
Rolling	8	1.79
Medium	8	2.44
Severe	8	3.46
Rolling	10	1.83
Medium	10	2.60
Severe	10	3.30
Rolling	15	2.16
Medium	15	2.60
Severe	15	3.60

Average Gcw: 52,225
Average Hp (Gross): 155
Average Mph: 34.43

expansion and fade. This is the standard argument for a built-in retarder (hydraulic or electric, for example).

The greatly reduced speed and increased fuel use on grades drive the truckers to toll roads in sections where they cut down grades.

Grades Speeds—Adverse—Fig. 8 shows actual speeds compared with calculated speeds on adverse grades for a unit with our specifications (55,000 lb and 150 hp) having an ideal transmission.

Actual speeds are seriously below what should have been obtained, due to:

1. Shift time.
2. Power unavailable due to ratio steps.
3. Driver not able to keep up with changing road requirements.

Lost time is largely chargeable to the transmission, and closing the gap between the actual and ideal curves has always been the goal. Test runs were made with 10- and 15-speed combinations.

Fig. 9 shows the situation for a 5-speed box where a fourth gear will not pull the hill and third runs the engine at 117 hp, top speed, instead of the 150 available.

We can see that if the engine is lugged down in fourth gear it will almost hold the load at 15 or 20 mph. This is bad driving, but not uncommon. The good driver will anticipate the condition and drop back to third gear, keeping his engine speed up. The continuous necessity of making such decisions plus the hard work involved in continuously shifting multi-speed combinations results in:

(a) Failure of gear set to deliver maximum hp to axle.
(b) Wide differences between driving times.

Acceleration—The acceleration time for a highway combination is serious if many stops are made. A through run on Ohio main roads shows records of a stop every 3.53 miles, and 8.15 stops per hr.

Fig. 10 shows ideal acceleration time for a 55,000-lb, 150-hp rig.

The time lost per stop due to slower average speed while accelerating is 0.85 min if the road speed is 47 mph on the level.

At 50 stops per day, time loss accelerating would be 0.71 hr.

This type of service would call for gear steps delivering maximum horsepower for maximum acceleration, and if extreme, a larger engine.

Note the very short time in lower gears. This explains the skipping of steps below 20 mph—time-saving does not justify the extra shifts.

Gear Shifting and Shift Time

We are concerned with gear shifting on two major angles:

1. Engine power is cut off during shifting. This is

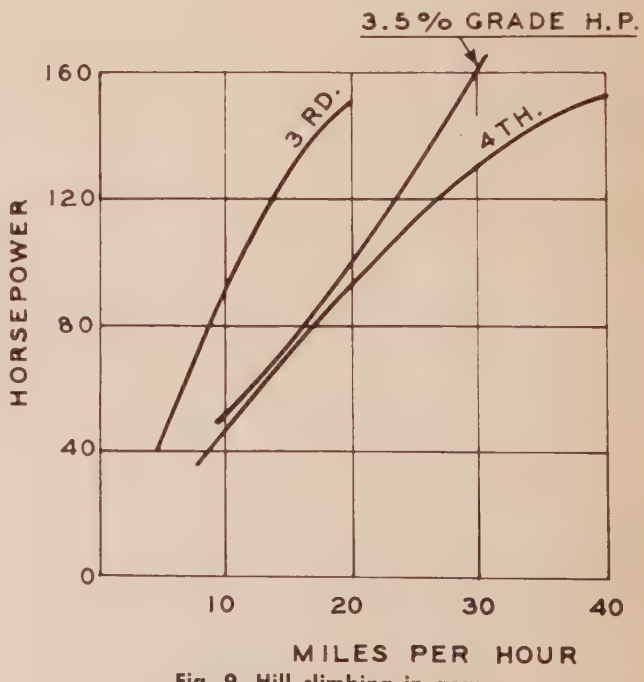

Fig. 9—Hill climbing in gear

Table 5—Shift Time—5-Speed Transmissions

Shift from	Ratio Step	Synchro Box Time, sec	Non-Synchro-Double-Clutch Time, sec
Second to Third	1.94	1.3	2.2
Third to Fourth	1.81	1.2	1.9
Fourth to Fifth	1.43	0.7	1.5
Fifth to Fourth	1.43	1.5	1.5
Fourth to Third	1.81	2.3	2.3
Third to Second	1.94	2.2	2.2

a direct loss in horsepower-hours, producing a time loss of equal percentage.

2. Driver's ability to keep the power to the axles up is affected by time, effort, frequency, and difficulty applying to the particular shift.

Driver reaction and maintenance cost tie in strongly.

We will attempt to outline roughly the number of shifts and time per shift, which would give an idea of the time lost.

Shift Frequency

Observations on various runs of truck-tractor-trailer units such as we are considering are given in Table 4.

All were run under supervised test conditions, fully loaded, with an observer.

These figures cover work in light rolling to severe hills, running in Ohio, Indiana, and the Southwest; they include no driving through major cities.

Table 5 shows the times obtained when we ran some checks on shifting with a good driver.

The synchronized transmission shows gains on the upshift because the driver does not have to wait for the engine to slow down, as in double-clutching. Effort is greater downshifting than upshifting the synchronized box since the cone clutch has to speed up the disc and gear inertia against friction (friction helps on upshift). This accounts for the longer time in downshifts.

Shift time with a 2-stick shift, close-stepped auxiliary plus main box may be generalized at 2.1 sec.

Time for a close-step single shift in auxiliary would be about the same as fourth to fifth and fifth to fourth above, 1.5 sec.

Two-speed axle shifts, either up or down, require somewhat more time than is required to accelerate or decelerate the engine to the new speed. (See Fig. 11.)

Times for combination shifts of a 2-speed axle and main box are quite close to the 5-speed transmission times given.

Distribution of shifts, from tests, are given in Table 6.

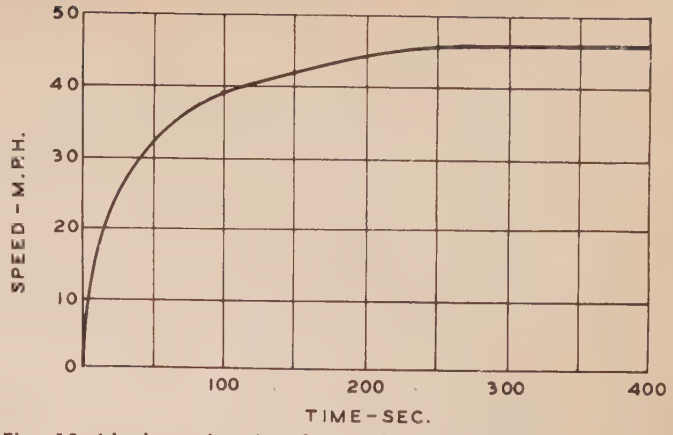

Fig. 10—Ideal acceleration for truck of 55,000-lb gcw, 150 hp net

No attempt will be made to predict the general time lost in shifting, but we might try a random calculation.

34.4 mph, includes 10% personal time.
2 shifts per mile.
2 sec per shift, engine time loss.
Total: 137 sec or 4.2% of driving time.

This is not trivial and shows a field for improvement by simplified arangement, low inertia spinning parts, clutch brake, power shift, or automatic shift.

Shift time ties in with:

A. Engine acceleration and decleration time, declutched.
B. Inertia of the clutch disc and the transmission gear train.

On double-clutched shifts "A" definitely limits the time, but "B" prescribes the accuracy with which speeds must be held. On synchronized shifts "B" sets the time.

Fig. 11 shows acceleration and decleration times for 6-cyl and 8-cyl engines of medium size. The driver must, in double-clutching, wait for the engine to reach the new speed—a definite time factor.

In all shifts the difficulty goes up roughly with the

Table 6—Distribution of Shifts

5-Speed Box—Rolling Hills

Into Gear	Fifth	Fourth	Third	Second
Total shifts, %	26	32	26	16

10-Speed Combination—Severe Hills (2-Speed Axle)

Into Gear	Tenth	Ninth	Eighth	Seventh	Sixth	Fifth	Fourth	Third
Total shifts, %	4.3	3.8	13.5	21.6	19.4	17.2	11.3	9

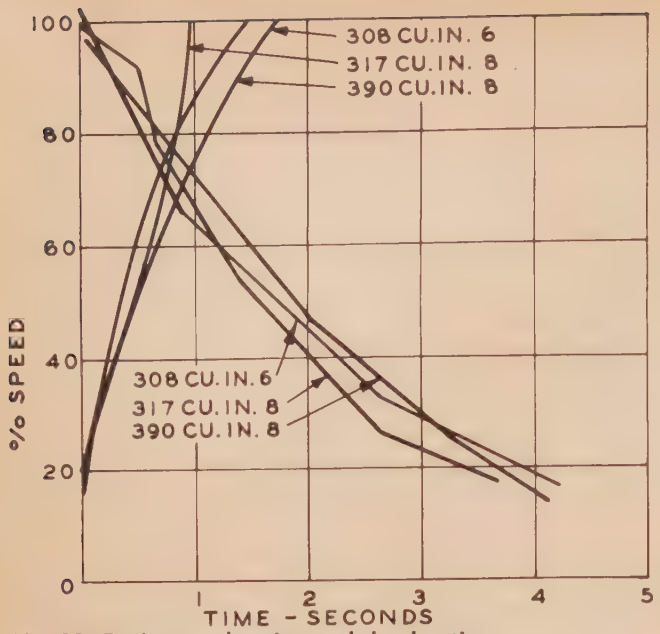

Fig. 11—Engine acceleration and deceleration
Acceleration: declutched—full throttle
Deceleration: declutched—closed throttle

308 cu in. 6—3000 rpm max
317 cu in. 8—3800 rpm max
390 cu in. 8—3250 rpm max

square of the engine speed, calling for more careful design of shift details with the new engines.

We are highly interested in the inertia of the clutch disc. Fig. 12 shows the savings that are possible when a double disc clutch replaces the usual single disc. A gain in the form of reduced pedal pressure is also made, as shown by Fig. 13.

The clutch effort is a phase of gear shifting that deserves much more attention. Continually pumping a clutch pedal with a resistance of 45 lb or more will soon fatigue the most rugged driver. We would rate an improvement here as a top project.

The 2-plate clutch also reduces the WR^2 chargeable to the flywheel, since its area is disposed on a smaller mean radius, and the flywheel diameter can be reduced. This would show up on engine acceleration and deceleration. Still on the subject of inertia, it would seem that designers should make an effort to reduce the weight of the gears which run with the clutch disc, which includes practically all of them. The inertia of these gears enters into one or two million shifts in the life of the unit and is well worth going after.

To repeat, if the clutch disc and the gear train have high inertia, we will have more effort and delay in a synchronized transmission, and get more clashing in a double-clutched job.

Table 7—Fuel Consumed on Various Runs

(55,000-lb—150 hp—flat bed)

Rate of Rise, %	Fuel, mpg	Fuel, gph
0	5.4	9.62 (legal speed)
2	2.2	10.9
4	1.3	10.8
6	0.86	11.9
8	0.62	11.3

Table 8—Fuel Consumption on Down-Grade

Rate of Fall, %	Mpg
0	5.55
2	9.52
4	11.1
6	11.4
8	11.4

Fuel

Gasoline Consumption—Uphill—Again from the Pennsylvania tests we show in Table 7 the fuel consumed when the vehicle was run on various degrees of adverse grades; these are averages, not definite, uniform grades.

The unit runs with 13 gph at wide-open throttle. It burns less on adverse grades. We have seen that, due to a finite number of gear ratios, the horsepower to the wheels is well down from maximum. A lot of the running is at part throttle.

There is certainly a loss due to constant use of the carburetor accelerator jet while making the very numerous shifts.

Some further effects, such as governor cutting in (in geared speeds) or mixture disturbance due to continuous changes, may enter in.

This would be an interesting field for study, by cycling an engine on the stand in a manner similar to its use on hills.

Gasoline Consumption—Downhill—Unfortunately, the engine keeps on burning fuel when running downhill.

Table 8, from the Pennsylvania tests, shows how our unit consumes fuel down-grade.

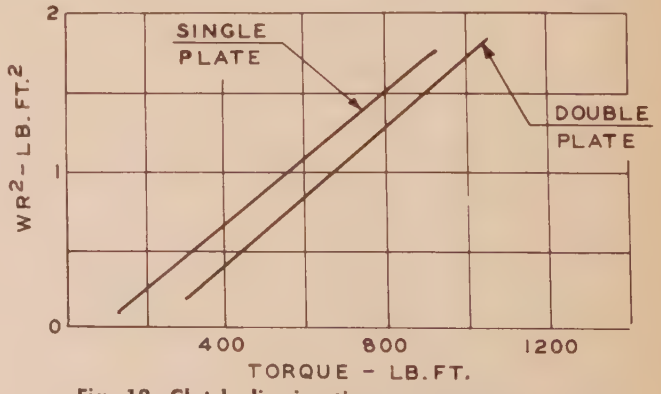

Fig. 12—Clutch disc inertia versus torque capacity

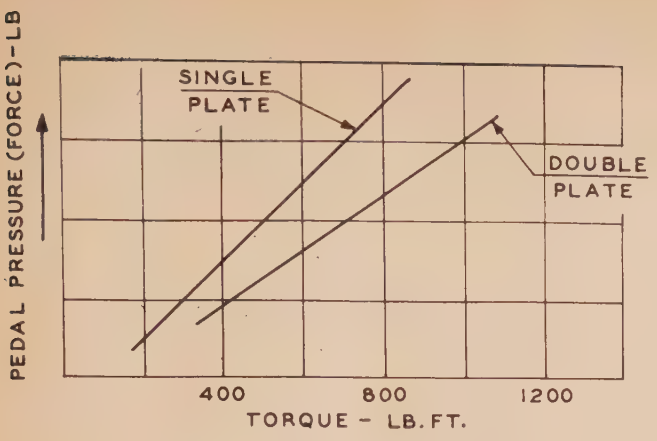

Fig. 13—Clutch torque capacity versus pedal pressure

Roughly, the fuel consumption is, on severe downgrade, half the level consumption. This is due to the use of lower gears for engine braking, producing a high rate of engine revs per mile with idle fuel feed operating at high vacuum.

Drive line engineers might do something here with a retarder which would permit running downhill in faster gear for less engine revs per mile, or even with the engine declutched. This seems to be a drive line problem.

The engine man might train the engine to use less fuel under fast coast conditions.

Overall Consumption on Hills—Table 9, showing overall fuel use in hills, is shown to complete the picture. Note the sharply increased fuel even on light grades.

Design-wise we can only work on the separate uphill and downhill phases, which have already been discussed.

One transmission factor enters: If a gear speed higher than normal is available, the driver will often run up to this speed down short hills or "knobs" to get a running start up the next one. This may save fuel as well as time.

Ton-Miles per Gallon versus Load—In our survey it may be interesting to check the effect of load on fuel cost.

Economy at 50+ mph, from the Pennsylvania tests, level running, 55,000 lb, gcw is given in Table 10.

Table 9—Gasoline Consumption on Hills

(55,000-lb—flat bed; rise and fall equal)

Rate of Rise and Fall, ft per 100 ft	Fuel, mpg	Fuel, ton-miles per gal
0	5.4	148
2	3.51	96.5
4	2.47	67.8
6	1.64	45.1
8	1.16	31.8

Table 10—Economy Results

Unit Weight, lb	Fuel, mpg	Fuel, ton-miles per gal
20,000	8.7	87
40,000	5.9	118
60,000	5.3	158
80,000	4.75	190

Fig. 14 (by A. F. Stamm of Dodge-Chrysler) shows rate of doing work, output, versus rate of doing work, input (ton-miles per hour per horsepower) for various speeds for van unit of 100 sq ft front area. Fig. 15 (also by A. F. Stamm) shows the same data re-arranged.

These data suggest that our 4-axle 55,000-lb job may some day be replaced by a heavier 5-axle unit; for economic reasons there will be continuous pressure on the states with lower load and length limits to raise them for higher gross.

New 4-lane freeways with ample turn radii could permit, for instance, the use of so-called double-bottom jobs; tractor-semi-trailer and full trailer—with five axles. Here we have a permitted gross of 72,000 lb or more, while complying with width, axle-load, and bridge formula regulations.

These curves show that the economic limits for freight haulers lie in the smaller, not the larger, sizes. Haulers on private rate-of-way, such as loggers and miners have moved in the direction of heavier, moderate-speed units to get more ton-miles per hour for less cost.

With larger units we shall require greater gear range and more steps, judging by past experience.

Gasoline Consumption with New Engines and Transmissions—We have seen how very important fuel cost is to the operator. Also, it is almost the only

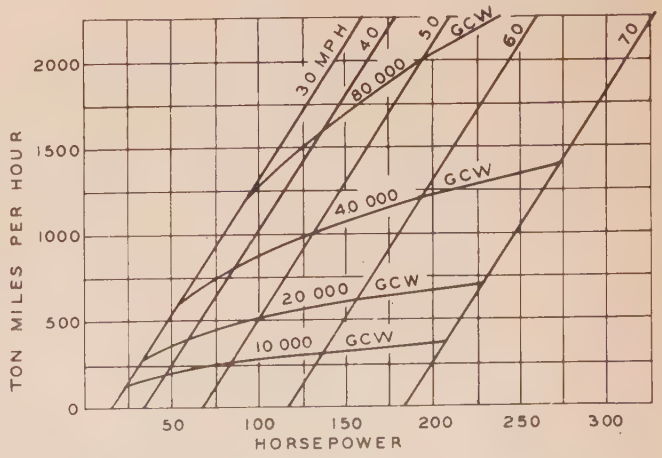

Fig. 14—Horsepower versus ton-miles per hour

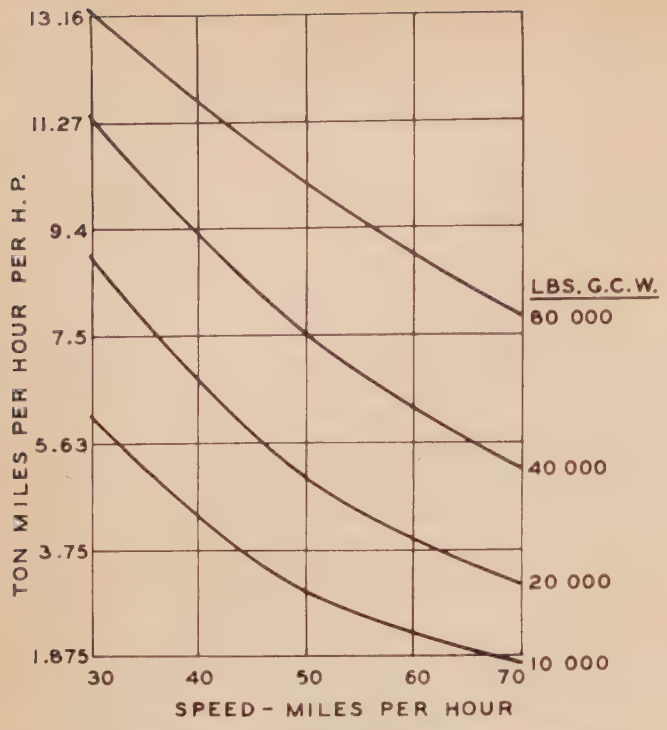

Fig. 15—Speed versus ton-miles per hour per horsepower

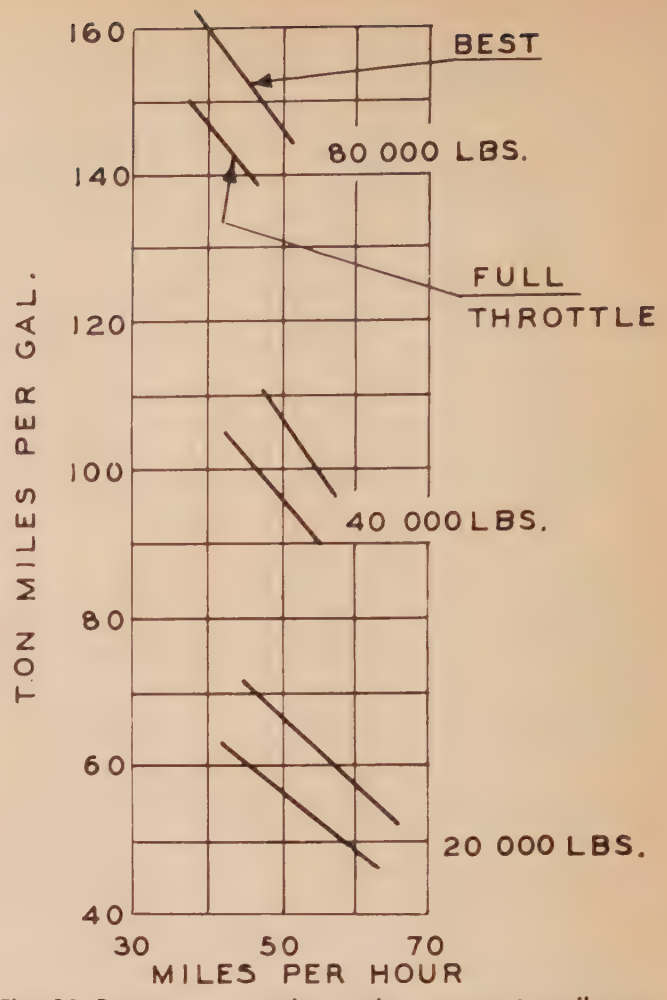

Fig. 16—Best-economy engine settings versus ton-miles per gallon

cost about which he can do anything. We have seen that 150 net hp will run our 55,000-lb rig into speeds where fuel costs begin to hurt.

Recent engine models are making available an abundance of horsepower at reasonable cost, with short engine length.

The owner cannot usually cash in on higher speeds because trip allowed time has been frozen by union region-wide agreements.

Let us now do some freewheeling, partly in the engine man's territory. Transmissions have always been designed to suit the engine, but perhaps the engine can stand some adapting.

A transmission is postulated which will hold the engine speed at full throttle above 70% (or higher) of the governed on up-and-down shifts. Part-throttle shifts can occur at lower speeds.

It will be recalled that the diesel engine became successful on freight hauling only after the engine was held within a close speed range, which required close gear ratio steps and driver training.

Acceptance of the transmission for a truck-tractor by a customer is based on profitable operation. Factors in the profit-making concept are:

(a) Reliability.
(b) Maintenance cost.
(c) Fuel cost.
(d) Driver preference.
(e) Payload ton-miles output.

Fuel and ton-miles per hour tie in rather closely: A 2-speed axle or auxiliary rig in hilly terrain will, if vigorously shifted, save by test 5–10% time; a job shifting faster and closer to the horsepower needs should do better; we could then, with better gear sets, cut down the top speed while holding the trip time.

Time saving can be swapped for a fuel saving by such changes as a faster axle. In detail these changes might be:

1. Reduction of truck maximum speed on the level:

$$\text{hp} = \frac{(KW + K_1AV^2)\ V}{375}$$

where:

K = Rolling resistance factor: 0.01
K_1 = Air resistance factor: 0.002
W = Weight
A = Frontal area
V = Speed, mph

This equation permits calculating ton-miles per horsepower-hour (see Figs. 14 and 15).

2. Reduction of engine revs per mile by axle ratio;

this shows steady improvement on manual-shift job up to the point where too much down-shifting from high gear is needed. The key to success with this idea seems to be a close step next to high with fast or automatic shift. Faster axle ratio may be alternatively stated: Set engine speed for level cruising at economy speed for engine. (Fuel per brake horsepower-hour rises at the high-speed end of curve.)

3. Set engine power curve on best economy instead of maximum power. Fig. 16 (also by A. F. Stamm) shows the possibility of using an oversize engine cut back in power, plus lower road speeds. The engine power fuel curve would run through the bottoms of the fishhooks.

4. Repeat No. 3 with a smaller intake manifold for better distribution, reducing engine peak hp, or smaller venturi.

5. Step up compression ratio to take advantage of the fact that the transmission cannot run at full load below 70% of engine governed speed.

Octane requirement increases more rapidly below 70% speed, full throttle; close steps avoid this condition. (See Fig. 17.)

6. Tie Nos. 4 and 5 together, so that more drop in manifold is balanced by more compression ratio. This reduces the valve and ring temperatures for better engine life, and better economy at the tail end of the maintenance period.

7. Leaner mixtures for better economy. With high average engine speed, missing is less probable. (See Fig. 18.)

The assumption is that misfiring is the primary evil with lean mixtures, and valve life can be handled.

8. Reduce downhill fuel losses; with present jobs the bad hills are run in a lower gear for engine braking, resulting in a long period of fast coast at high manifold vacuum. Fuel loss here may be an engine problem.

9. To some degree, the accelerator jet can be leaned out. We do not call for great drops in engine speed, with our hypothetical transmission.

10. Since full throttle at speeds below 70% of engine governed is improbable, can the spark-advance curve be sharpened for best economy above the 70% speed?

Automatic Operation

Many of the problems pointed out have a common solution—the automatic transmission. This would take care of clutch effort, shift effort, shift time, fast decisions and errors on hills, keep the engine speed in the range desired for economy and power, and permit engine design and settings for best economy.

The overall economics of the automatic are not yet clear, though there is a lot being learned.

The objections which have been brought up are:

1. Cost: Considering the prices which are paid, for instance, for automatic movements on lathes or radial drills and which save less time than the truck automatic, this should not be a barrier.

2. Maintenance: Shorter life has been confined to a few items, representing problems which can be solved.

3. Fuel economy: This represents a more serious objection. Various designs tested show 5–9% power loss at full speed, with 8% probable. At equal speeds a 5-speed hand-shift box will show a loss of only 2–2.5%. Adding 1–2% for losses in the auxiliary or 2-speed axle still leaves the automatic behind.

We expect to make money with the automatic, but the gains we can pick up are not large. We should not start with a minus percentage of any size. All the percentages we are dealing with are small, including the operator's profit.

It would be rash to forecast the early adoption of automatics for highway freight, but it would be more rash to say that the problems will not be licked, with the effort now being concentrated on them. It is to be expected that the automatic will make earlier progress

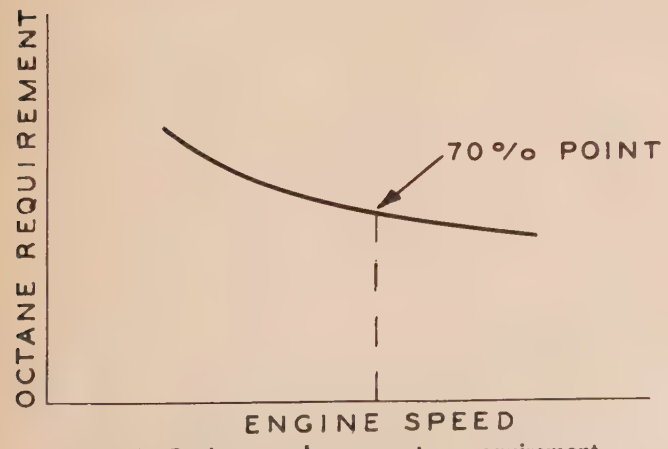

Fig. 17—Engine speed versus octane requirement

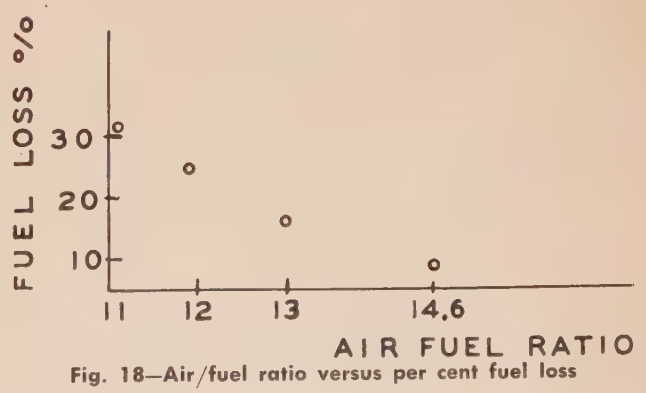

Fig. 18—Air/fuel ratio versus per cent fuel loss

in short haul, city, and off-highway uses, where shifting and clutch problems are most acute.

Safety

Present brake sizes are incapable of absorbing the heat generated on long downhill runs, without an assist from engine braking.

Brakes are as big now as space permits.

Congress, in the new road bill, has frozen truck widths at present standards so no relief can be expected from wider brakes.

Using engine friction as a brake, we have to shift down in gear ratio to a point where the engine friction torque can match the coasting torque from the axle. This means slower speeds. Higher gears throw more work on the brakes. There is a constant temptation for the drivers to cut loose to high downhill speeds, and plenty of accidents.

There is considerable thinking that the drive line should include a retarder or energy waster. This would permit the unit to descend the grade rapidly and still keep the brakes cool.

Such a retarder would have to pass the acid tests for transmissions. It must be: short, light, durable, and moderate in cost.

Unfortunately, the retarder introduces a driving complication; it applies braking torque to the drive wheels; these wheels also have to be equipped with drum brakes which will slide them.

If now the driver is using the retarder and jams on the brakes, the drive wheels slip and a jackknife accident may result.

Probably trailer brakes only should be applied while the retarder is in use, particularly on a slippery road.

Another factor to be watched is the rapid increase of torque in the fluid-type retarder, as the speed goes up. This can greatly exceed the capacity of the drive line in some types.

Retarders have had some sale in severe work such as logging and special applications; the recent upsurge of interest in the use of retarders for highway work and the approval where they have been tried point to an expansion in their use, where long severe grades exist.

The Propeller Shaft

The designer has three choices of propeller shaft speed in top gear:

1. Engine speed—direct.
2. Faster than engine—overdrive.
3. Slower than engine—underdrive.

No. 2 is appealing as a similar condition to high voltage in electric design, but with designs now available it has serious drawbacks. Forces in Cardan joints at varying angles go up with the square of speed; if vibration occurs either in a new or deteriorated shaft we have horsepower disappearing in vibration plus noise, driver discomfort, and frequent replacement of shafts.

This problem is one of the main challenges facing drive line designers.

While some propeller shafts run with fair success at speeds of 5000 rpm where the universal-joint angles are fixed, as in trolley cars, the varying angles on truck and truck tractor installations call for lower shaft speeds.

In truck sizes such as we are discussing this should be set at not much over 4000 rpm, unless ideal conditions prevail. Problems with higher speeds are balance, vibration, and noise when new, and the same problems accentuated as wear occurs.

Even at this speed if two shafts are used they should be assembled together with the center or midship support bearing when balancing, to avoid error accumulation at assembly.

Further, a bearing mounting should be used which cancels drive line vibration to a large extent.

Volume producers of 8-cyl trucks with engine speeds from 3500 to 4000 are avoiding the use of overdrive transmissions, which would speed up the propeller shaft. It is to be hoped this cautious approach continues.

Conclusion

There is very little brought out here which has not been thought of, or tried; we have attempted to sharpen the questions by bringing up the relevant data where possible. We cannot settle such controversies as: 2-speed axle versus compound gearbox; economic number of truck speeds; synchronized versus double-clutched shifts; automatic versus manual shift. Neither can we approve any of the concepts brought up without operating tests.

It is believed that manufacturers should carry on continuing road tests or test installations in line trucks to clear up some of the questions. Some answers are no doubt available now in fleet truck records, if some neutral investigator could compare them.

Only the largest truckers can afford test programs, yet each has a somewhat different problem, possibly requiring a different gear and engine combination.

In city bus work, where large buys are made, sample buses are loaned by the makers for evaluation of economy and overall speed. Perhaps more of this would help in freight hauling.

Acknowledgments

I thank the following for assistance: C. Saal, Bureau of Public Roads; A. F. Stamm, Chrysler Corp.; E. P. Lamb, Chrysler Corp.; L. T. Flynn, GMC Truck & Coach Division; B. Loeffler, Mack Mfg. Corp.; G. E. Soper, Mack Mfg. Corp.; B. Ucko, Mack Mfg. Corp.; C. A. Lindblom, International Harvester Co.; and members of the Dana staff.

PREIGNITION IN AIRCRAFT RECIPROCATING ENGINES

J. Anderson, Curtiss-Wright Corp.

This paper was presented at the SAE Annual Meeting, Detroit, Jan. 11, 1956.

THE service problem most commonly associated with preignition is the burning of holes through pistons. Other problems, such as failed exhaust valves and engine backfiring in certain types of engines, may also be involved. The known rapidity with which preignition does its damage and the suddenness usually associated with a failure, such as a hole burned through the piston, suggests an association between the two. It must be recognized, however, that conditions other than preignition, for example, severe detonation or high temperatures for prolonged periods of time may bring about similar results with preignition not occurring at all.

The primary purpose of this paper is to review the fundamental factors involved in preignition in aircraft reciprocating engines and, by so doing, provide, as much as possible, advice on its practical control in the maintenance and operation of the engine. It is not the intention to prove that all burned pistons experienced in service are caused by preignition, and in this respect we may deviate from the subject of the paper to cover what may be other important aspects of the overall service problem.

We have taken the liberty of using data from other publications in a number of instances to produce a more complete picture of the overall problem.

Outline

In order that the discussion might be followed more readily, an outline of what is to follow is presented. The differences between normal combustion, detonation, and preignition will be reviewed. Under "Effects," what these respective forms of combustion may produce will be discussed. Following this, a study of combustion-chamber areas will be made to analyze the direct causes of preignition. The subject of fuel will then be reviewed in general terms as to its possible effect on preignition and combustion-chamber difficulties. The final portion will be a brief resumé of important operation and maintenance factors.

Definitions

Fig. 1 presents pictorial definitions of normal combustion, detonation, and preignition. It is recognized that in the automotive industry recent attempts have been made to standardize on definitions, and that other terms are used to classify the abnormal combustions. Since many of these definitions are based upon audible noises, they are not practical for aircraft engines where the level of other noises is high. The terms "detonation" and "preignition," will be used because they have been generally accepted in the aircraft industry.

In the case of normal combustion the charge is ignited by the spark plug at the designed time with respect to crankshaft angle. Once ignition has been initiated, the flame will progress through the combus-

HOLES burned in pistons are usually associated with the word preignition. There are, however, other causes for piston burning, such as, detonation and excessive temperatures, and these as well as the fundamental factors causing preignition, are discussed in this paper.

The author suggests that spark-plug condition is probably the most important factor involved in preignition.

Considering the other causes of piston burning, operation, maintenance, and possibly fuel variations, all become important because each may contribute to conditions conducive to piston burning.

tion chamber at a speed which is very much affected by the fuel/air ratio. Flame speeds at best power fuel/air ratio are in the order of 80 fps, while at very lean mixtures, flame speeds decrease to as low as 45 fps.

When detonation occurs, there is usually first a normal flame which is followed by a spontaneous combustion of the remaining unburned charge in the combustion chamber. With this spontaneous combustion high-pressure vibratory waves are induced which may be detected by a vibration pickup installed on the cylinder head. This is the basic principle used for detonation detection on aircraft test engines. This description of detonation is perhaps oversimplified but it is not the intent nor object of this paper to go into the latest research findings of the phenomena of detonation.

Preignition, as the word indicates, is ignition before normal spark. When preignition occurs, there is a flame similar to a normal flame emanating from the point of preignition. The reaction, however, as will be illustrated later, is quite different and more severe. To simplify the pictures, we have not illustrated the combinations of the different types of combustion which may occur. For example, it is very likely, if conditions of spark advance are correct during preignition, a detonation may also occur. Thus, if we were to draw another picture of detonation occurring with preignition, it would be similar to the preignition illustration, except that a detonation area would be indicated at some point remote from the initial source of preignition.

Fig. 2 illustrates what happens. It shows the relationship of combustion-chamber gas pressures to crankshaft angle for the various forms of combustion. Referring to normal ignition, for optimum operating efficiency, the spark timing should be such that the complete combustion of the mixture occurs at or soon after top dead-center of the power stroke. Thus, increased spark advance is used at lean cruise conditions to compensate for the lower flame speeds. The engine is designed and tested for normal combustion temperatures and pressures. However, aluminum parts, such as a piston, when exposed to higher than normal temperatures and powers will exhibit lower hardness and strength. This loss of hardness and strength is progressive with time of exposure, and it is possible over a long period of time that structural failures may result in gas blow-by and piston burning without the occurrence of detonation or preignition.

The high vibratory pressures associated with the spontaneous combustion of detonation may exceed the normal combustion peak pressures by 100%. When conditions of detonation are approached, very infrequent indications, perhaps a few flashes per minute, occur on the scope of the detector. As conditions for detonation are made more severe, the frequency and the amplitudes of the indications are increased to what is termed heavy continuous detonation. Operation with detonation may be harmful in a short period of time, if severe, or in a longer period of time if milder. As will be shown later, the damage done by detonation in its early stages appears to be a process of erosion on aluminum areas of the combustion chamber. This condition can be progressive and eventually cause more serious difficulty. There is an increase of temperature associated with detonation which is barely discernible with occasional detonation, but if the detonation is permitted to increase in severity, the increase becomes more pronounced. Ordinarily, this increase, as measured by the cylinder-head thermocouple is not large enough to detect harmful detonation in aircraft engines. The increase of internal combustion-chamber temperatures, however, may decrease the hardness and strength of aluminum parts over a period of time. If internal temperature conditions are marginal, it may be sufficient to overheat combustion-chamber areas sufficiently to initiate preignition.

As illustrated in Fig. 2, preignition is initiated prior

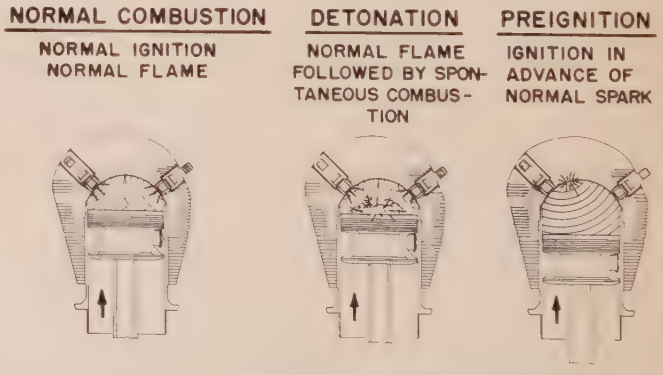

Fig. 1—Definitions of normal combustion, detonation, and preignition

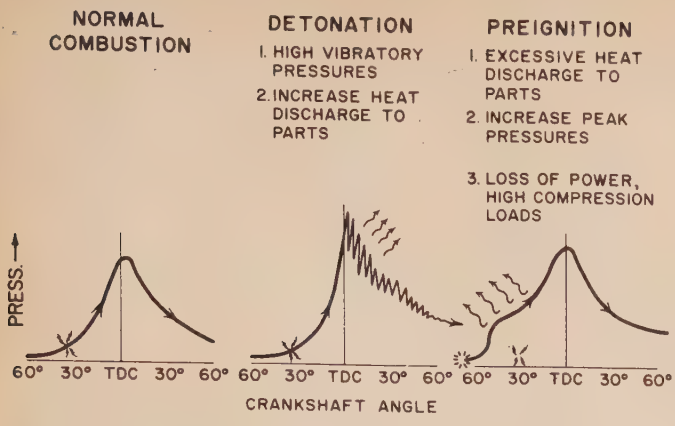

Fig. 2—Relationship of combustion-chamber gas pressures to crankshaft angle for various forms of combustion

to the normal spark. When preignition is experienced, the shape of the pressure curve btdc will depend to a large extent upon how far preignition is initiated in advance of the normal spark. During runaway preignition where the advance becomes progressively larger, there is excessive heat transfer which, if allowed to exist for longer than a very short time, for example, less than a minute, will be disastrous to the combustion-chamber parts. The high heat transfer is caused by the longer period of hot gas exposure, as well as the far above normal gas temperatures produced by the early ignition of the charge and the subsequent compression of the hot gases. As an example, a rapid rise of cylinder-head temperature from 500 to 620 F was observed when preignition was deliberately caused in a single-cylinder engine. The engine was stopped within seconds, before any piston burning occurred. Because of the excessive heat loss and the higher pressures during the compression stroke, the net power from the cylinder involved may be negative, that is to say, the work being done during the compression stroke will be greater than that which is returned during power stroke.

Summarizing thus far, high operating temperatures may reduce the hardness and strength of aluminum pistons so that structural failure and possibly burned pistons may occur. The high vibratory pressures of detonation can be harmful, but the associated increase of temperature can also soften and weaken aluminum parts, and may, under certain conditions, cause preignition. Uncontrolled preignition is primarily harmful because of the high heat dissipation associated with it.

Effects

It is not always possible to say with certainty whether a burned piston was caused by preignition, detonation, or structural causes, nor is it often possible to tell the exact mechanism of final failure. The final damage often looks similar. Where detonation is involved and the damage is limited, it is usually possible to recognize the symptoms. Fig. 3 shows various stages of detonation-damaged pistons. Note the characteristic eroded areas associated with detonation. The pistons at the right show advanced stages of damage. Based on accelerated tests were severity of detonation was controlled, it may be said that the progression of damage is a function of the severity and the time of exposure to detonation. Other factors, such as temperature of the combustion-chamber parts during detonation, also appear to have an effect on how rapid the damage will progress.

Fig. 4 shows damaged pistons believed to have been caused by preignition. It is generally felt that when a hole is burned through the dome of a piston away from the ring-belt area and not associated with a structural failure of the piston, it is most probably caused by preignition. It is entirely possible in some cases that detonation preceded the preignition that did the final damage.

The excessive heat of preignition may make a direct attack at the ring-belt area or cause seizure

Fig. 3—Various stages of detonation-damaged pistons showing eroded areas

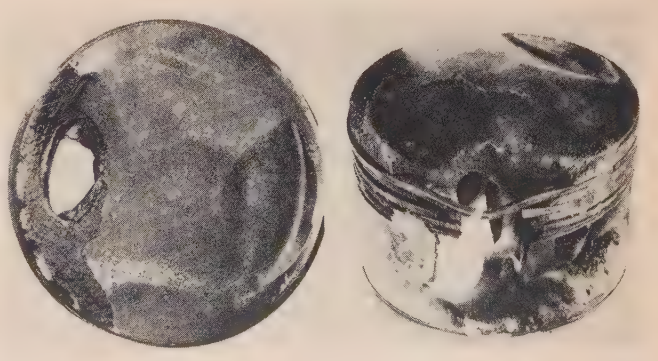

Fig. 4—Pistons believed to have been damaged by preignition

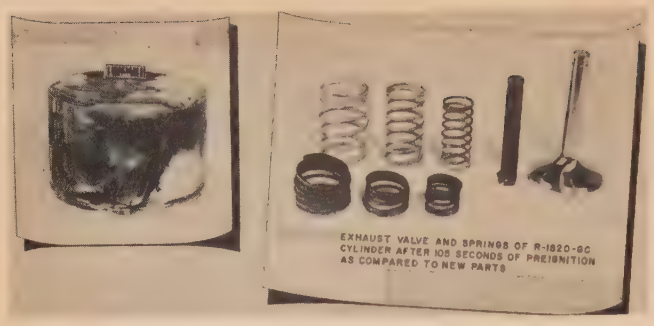

Fig. 5—Effects of controlled preignition tests on piston and exhaust valves

Fig. 6—Structural failures possibly caused by detonation which may be early stages of holes burned through pistons

resulting in a burned hole in this area. Support of this is shown at the left of Fig. 5, where the results of a controlled preignition test in a liquid cooled V-type engine are shown. The reference literature [1] reported that if the piston clearance in the cylinder were increased, the results of preignition would fail the exhaust valve first before burning a hole through the piston. Obviously, a preferred solution would be to correct the cause of preignition if possible.

The picture at the right of Fig. 5 (also taken from published literature)[2] shows the results of preignition deliberately induced in an aircooled single-cylinder test engine by a hot spark plug. In this case, the valve failed instead of the piston. This test would suggest that some exhaust valve failures experienced in service may have been caused by preignition. We do not wish to imply, however, that this is the cause of all exhaust valve failures.

Fig. 6 shows examples of structural failures which may be early stages of holes burned through the piston. They may or may not have been associated with detonation, but in any case were probably aggravated by high operating temperatures.

To make the problem of analysis more complex, there are reasons for burned holes in pistons besides those discussed. For example, ring failures can cause damage and blow-by conditions which could result in melting holes through the piston. Mechanical scoring and seizure of pistons may conceivably create sufficient temperatures to melt the aluminum and produce holes in the piston.

Direct Causes of Preignition

Preignition is an abnormal condition and it occurs when the ignition is initiated prior to the normal spark. This ignition may be the result of either abnormal spark ignition occurring at the spark plug or ignitions resulting from an abnormal hot spot. Each of the two types of abnormal ignition will be discussed.

Fig. 7 refers to an example of abnormal spark condition which was corrected by removal of a condenser from a high-tension ignition system. The condition in this instance was not directly associated with preignition damage, but it does illustrate a condition of abnormal spark ignition which may have caused damage if other conditions were marginal. The secondary condenser, originally required to shorten the period of spark and avoid electrical interference with the following spark, caused abnormal advance by increasing the high tension system capacitance so that the stored up voltage discharged itself in advance of the normal breaker point opening as the distributor finger approached the next electrode in the distributor block. The original design spark timing was 20-deg btdc advance but after the removal of the secondary condenser, 28-deg btdc advance was required to regain the average performance realized previously with the abnormal spark advance. In some cases this abnormal spark advance was recorded at 32-deg btdc and higher.

Some years ago there were several instances of burned pistons which occurred in an engine equipped with a low-tension ignition system. In one case three pistons were burned before the cause of difficulty was finally located. As shown in Fig. 8, overlaying wires in the distributor permitted chafing of the insulation and caused spark ignition in some of the cylinders in advance of the predetermined timing. A rearrangement of the wires, as shown, corrected the problem.

Automatic spark advance mechanisms controlled by engine speed generally seem to do a good job but they have been known to function abnormally in instances permitting advanced spark when not intended. On more recent engines, the advancing of the spark ignition is now done manually by the flight engineer. This permits a more consistent spark timing, and in addition provides a larger range of operating conditions where the spark advance can be permitted. (See Fig. 9.)

Before leaving this discussion of abnormal spark advance, it should be noted that important conditions

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[1] See NACA TN 1637 (June, 1948), "Effect of Preignition on Cylinder Temperatures, Pressures, Power Output, and Piston Failures," by L. C. Corrington and W. F. Fisher.

[2] See SAE *Quarterly* Transactions, Vol. 2, October, 1948, pp. 546–562: "Preignition and Its Deleterious Effects in Aircraft Engines," by A. Hundere and J. A. Bert.

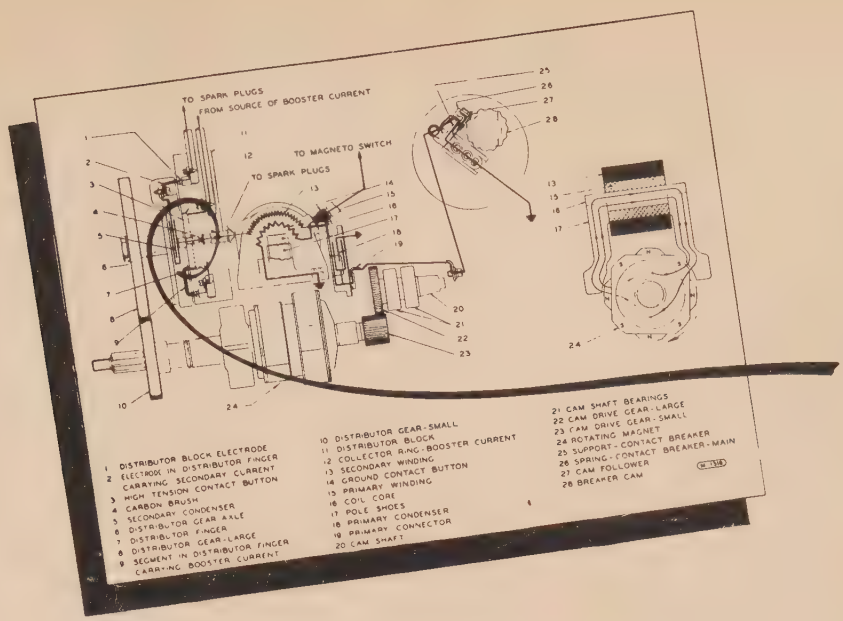


Fig. 7—Abnormal spark advance corrected by removal of secondary condenser from high-tension magneto

for satisfactory combustion-chamber performance are (1) proper setting of ignition timing during engine buildup, (2) proper functioning of the ignition system as a unit, and (3) proper use of the manual spark advance by the flight engineer.

The items considered as the hottest parts of the combustion chamber are spark plugs, exhaust valves, and combustion-chamber deposits, and these, therefore, are considered possible hot spots for preignition if abnormal conditions exist.

Single-cylinder tests at extremely severe operating conditions indicated that preignition occurred with the spark-plug insulator tip temperature of 1600-1765 F. These temperatures (Fig. 10) verify other published data which indicate that preignition will be induced at surface temperatures in the range of 1600-1900 F.

Spark plugs are given ratings which are determined by their tendency to cause preignition (Fig. 11) in a small-size liquid-cooled engine using benzine as a fuel. The length of exposed ceramic is one of the major features in determining the heat range or the rating of the spark plug. In general, the more exposure and the longer the distance heat has to travel to become dissipated, the hotter the spark plug will be. If a spark plug operates too hot it may cause preignition. On the other hand, if it operates too cold, as it may during ground idle or at low power cruise conditions in extremely cold atmospheres, the plug may foul out from carbon deposits. The spark-plug design is therefore a compromise between severe and cold engine operation.

Limited data on a full-scale engine using thermo-

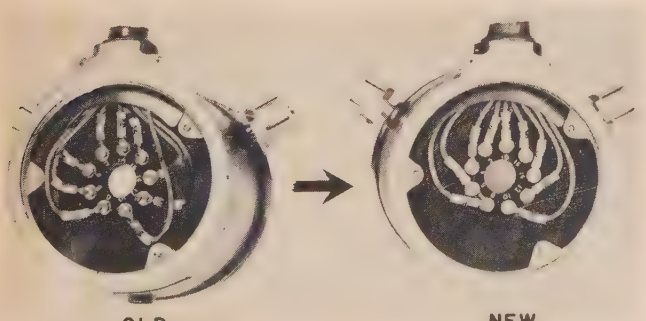


Fig. 8—Chafed wires in distributor which lead to piston burning corrected by rearrangement of wires

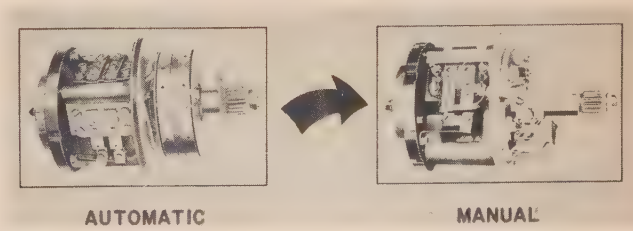


Fig. 9—Manual spark advance gives better control of spark timing

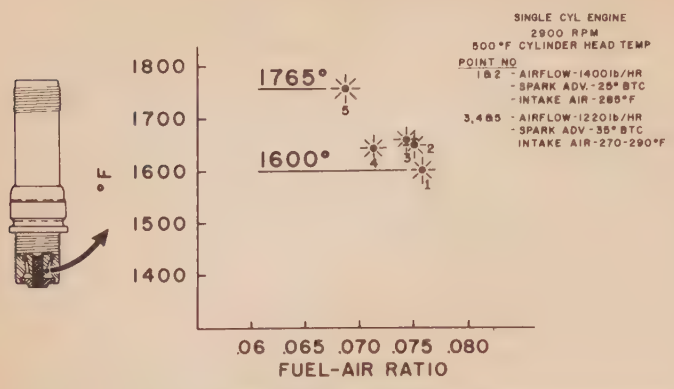


Fig. 10—Preignition with spark-plug insulator tip temperatures of 1600-1765 F

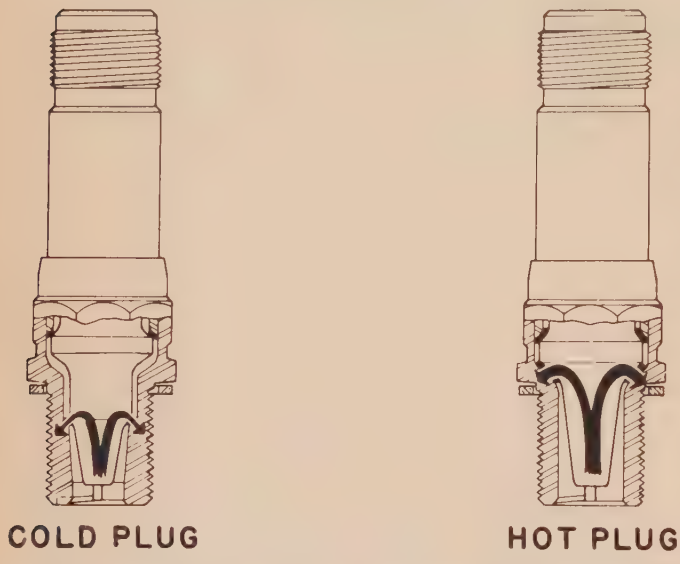


Fig. 11—Rating of spark plugs determined by tendency to cause preignition and related to length of ceramic exposed

coupled spark plugs measuring the ceramic insulator tip temperature indicate that there is a reasonable margin before temperatures critical to preignition are experienced.

There is also a spread of temperatures to be expected between similar plugs in different cylinders. This spread is most probably caused by tolerances between similar plugs and the variation of heat transfer from the plug to the cylinder head. As expected, temperatures taken on a hotter plug (lower preignition rating) gave higher temperatures as illustrated in Fig. 12, plug A. Like other combustion-chamber parts, spark-plug temperatures are sensitive to fuel/air ratio changes, and it appears that the insulator tip temperatures at cruise powers, if stoichiometric mixtures are used, may be slightly higher than those at take-off with rich mixtures. However, if manual leaning procedures below stoichiometric are observed during cruise, these temperatures will be lower than at take-off conditions.

Because of impaired heat transfer, cracked ceramics may overheat and cause preignition. The spark plug with a cracked ceramic shown in Fig. 13 was installed in a single-cylinder engine equipped with a preignition detector. To detect preignition, 90v are applied across the spark-plug gaps. This voltage is not sufficient to cause a spark. When preignition occurs, the flame will ionize the air between the spark-plug electrodes, and a flow of current, detected by an oscilloscope, will indicate preignition.

With the cracked ceramic plug, preignition was encountered at power conditions corresponding to those close to take-off. At cruise powers, preignition was not detected at 0.10 fuel/air ratio. On leaning out to 0.09 fuel/air ratio, intermittent preignition was indicated but apparently not serious enough to run away. This condition was allowed to exist several minutes. On leaning out further to 0.08 fuel/air ratio, runaway preignition did occur with characteristic extreme power loss and rapid increased cylinder-head temperatures.

Another potential for preignition is gas leakage, where combustion gases leak excessively between the insulator core and the outer shell or the center electrode of the plug, thus increasing the insulator temperatures. (See Fig. 14.) Although there have been improvements within the last few years, spark plugs must still be viewed as a possible factor in the piston burning problem.

The next possible hot spot to be discussed is the exhaust valve. In high-powered engines the exhaust valve design as shown in Fig. 15 is usually of a hollow head design so that liquid sodium may be used to transfer heat to the valve stem during the reciprocating motion of the valve. From the valve stem the heat

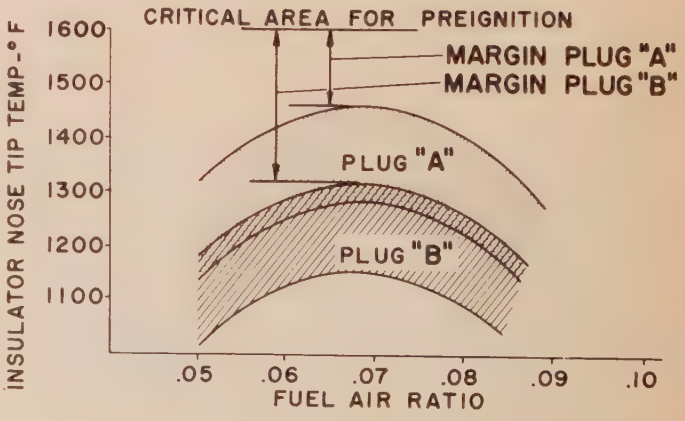


Fig. 12—Cooler spark plugs have larger margin than hotter plugs before preignition occurs

is then transferred into the valve guide and then to the cylinder head and fins. In addition, heat is also disposed via the valve seat into the cylinder head and cooling fins. Exhaust valve temperatures are relatively high because of exposure to combustion heat and particularly the heat of the exhaust stroke where the whole head of the valve is enveloped in the exhaust gases. However, unlike the secluded insulator tip of the spark plug, the dome is fairly well exposed to cooling effects of the incoming air and fuel, and there will be a tendency for the dome surface to be temporarily cooler than average valve temperatures during the compression stroke when the tendency for fuel preignition would be greatest.

Published literature [1] based on laboratory tests in other engines has indicated that the exhaust valve was a source of preignition when the dome of the valve was very badly eroded and that no preignition was experienced when nichrome coating was used to correct dome erosion. Service experience has shown a few instances of badly eroded domes where the quality of the nichrome application to the dome was inferior. In these instances, however, no preignition, at least of destructive nature, was evident.

Valve dome temperatures are in the order of 1400 F, as illustrated in Fig. 16. The lean mixtures of cruise powers may cause higher valve temperatures than those encountered at take-off powers with rich mixtures, particularly if mixtures close to stoichiometric are used. Considering that these temperatures are average and may not reflect the temporary cooler temperatures during the compression stroke, a reasonable preignition margin is indicated. It is not possible however, to rule out the exhaust valve as a source of preignition under certain abnormal conditions, but the preignition tests thus far suggest that the spark-plug temperature is probably the controlling factor.

In automotive engines, particularly with higher compression ratios, preignition induced by loose combustion-chamber deposits is a frequent occurrence. Fortunately, it appears that the preignition difficulty

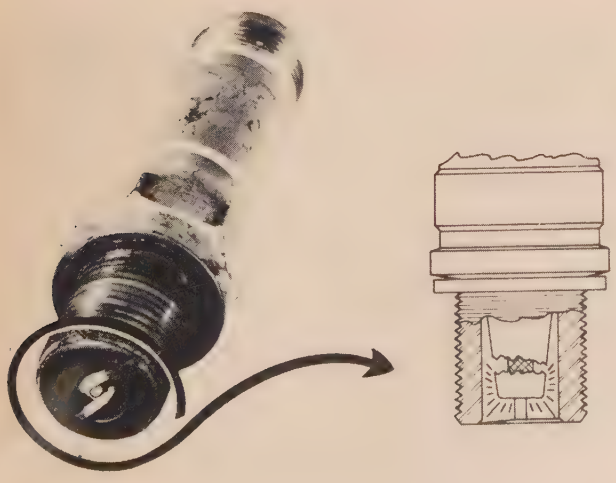


Fig. 13—Cracked ceramic may overheat and cause preignition

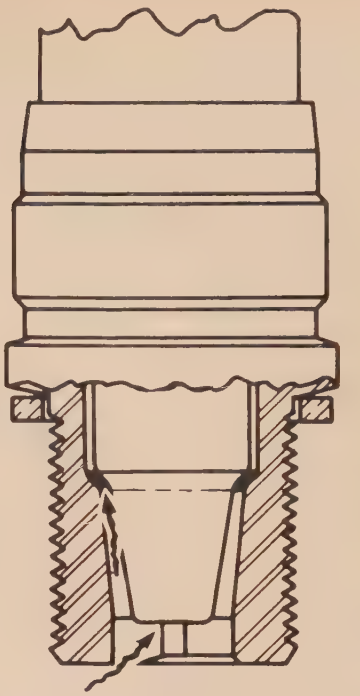


Fig. 14—Insulator core leakage may be potential cause of preignition

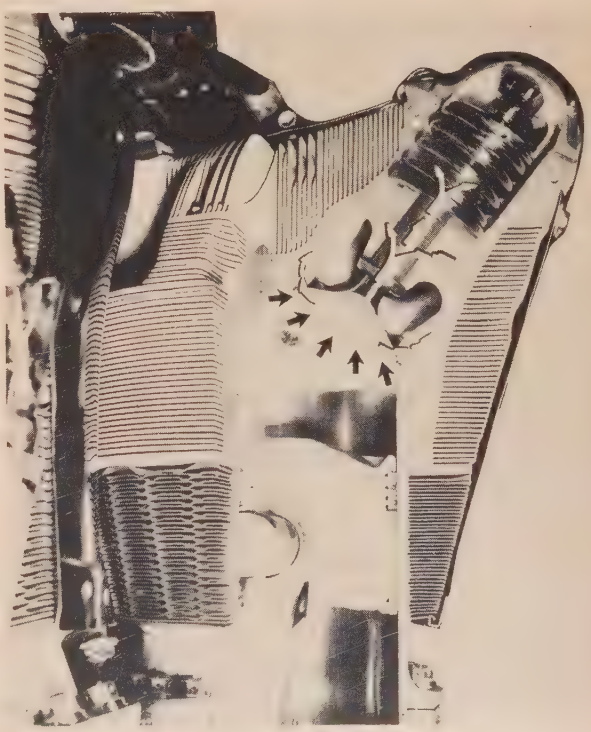


Fig. 15—Cutaway section of cylinder showing exhaust valve heat dissipation

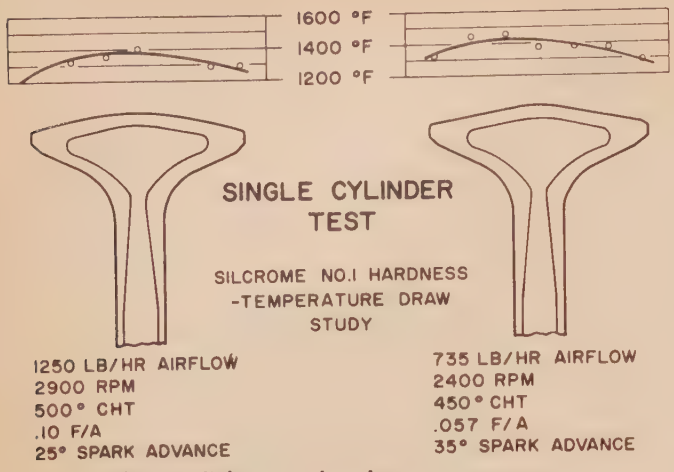


Fig. 16—Exhaust valve dome temperatures

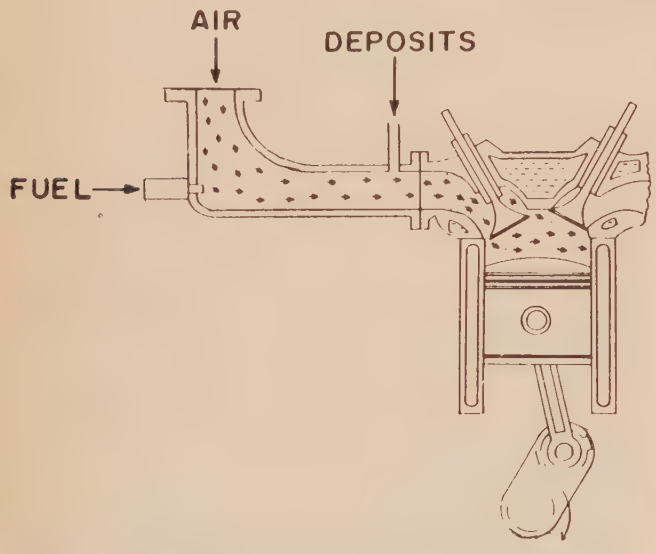
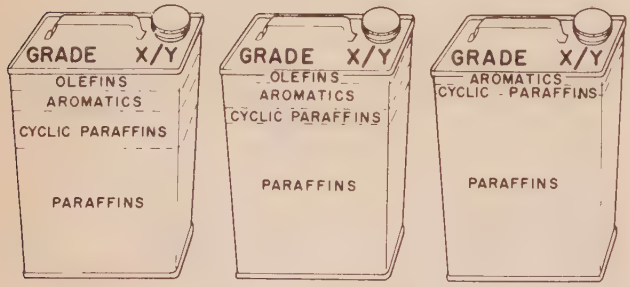


Fig. 17—Preignition in automotive engines as induced in laboratory



PARAFFINS – AT LEAST 60%
CYCLIC PARAFFINS–UP TO 30%
AROMATIC – UP TO 20%
OLEFINS – UP TO 5%

Fig. 18—Fuels meeting same grade or performance standards may be blends of different hydrocarbons in various proportions

is temporary and not of a runaway or serious destructive nature. An early procedure [3] (illustrated in Fig. 17) to induce preignition in the laboratory was established whereby combustion-chamber deposits from another engine, or any other materials as desired were injected into the induction system of the test engine.

The nature of combustion-chamber deposits in aircraft engines is generally not as heavy or as loose a formation as those found in the automotive engine, although flaky deposits are often found on the induction side of the intake valve. Combustion-chamber design, turbulence, scavenging, and temperature characteristics are undoubtedly strong factors in determining whether loose particles will cause preignition. A particle that has broken loose and caused to glow during the combustion of the charge, should not cause preignition if it is ejected from the cylinder during the following exhaust stroke.

In an attempt to induce preignition in a single-cylinder engine, the automotive test illustrated was duplicated. Various materials were used, including combustion-chamber deposits. The injections were made at various fuel/air ratios for cruise and 20% overboost powers. In no instance was preignition indicated. No preignition was evident during engine operation after long idling periods with and without the top compression ring inverted to encourage relatively large amounts of oil to be pumped into the combustion chamber. These tests indicated that the problem of preignition as caused by combustion-chamber deposits experienced in the automotive engine was not present.

Summarizing thus far, if combustion-chamber temperatures should be increased abnormally, it is probable that the spark plug would be the first to cause preignition. If the spark plug has either a cracked ceramic or an excessive gas leakage, preignition may result even under normal operating conditions. Combustion-chamber deposits do not appear to be a cause for preignition as evidenced by single-cylinder tests.

Fuels

Some comments should be made regarding fuels because it is possible that there are differences which could possibly aggravate conditions leading to combustion-chamber difficulties.

Many different types of hydrocarbons must be used in the manufacture of gasoline to meet the large demands and the stringent fuel specifications already established. Thus (as illustrated by Fig. 18) various groups of fuel meeting the same grade or performance standards may be blends of different hydrocarbons in

[3] See Proceedings of API, Vol. 33, Sec. 3, 1953, pp. 137–150: "Preignition in Automotive Engines," by J. R. Sabina, J. J. Mikita, and M. H. Campbell.

[4] See Proceedings of API, Vol. 34, Sec. 3, 1954, pp. 256–269: "Preignition Resistance of Fuels," by B. M. Sturgis, E. N. Cantwell, W. E. Morris, and D. L. Schultz.

[5] See Ethyl Corp. NAVAER–06–5–501, USAF T.O. No. 06–5–4: "Aviation Fuels and Their Effects on Engine Performance."

various proportions. The proportions would probably be within the following limits: 60% or more of paraffin, 30% or less of cyclic paraffins, 20% or less of aromatics, and 5% or less of olefins. To complicate the blending further there are many subdivisions to the hydrocarbons mentioned.

Primarily because of the different characteristics possessed by the individual hydrocarbons, two fuels meeting the same performance ratings may react differently when used at conditions different than those established for the performance-rating tests. To illustrate, Fig. 19 shows what may happen to the performance of isooctane, a reference fuel, and a so-called "very sensitive fuel" when the mixture temperature is changed from 200 to 300 F. There is a decreased detonation limit performance for both fuels as shown by the shaded areas, but with the sensitive fuel the decrease is more pronounced. Although not illustrated, if the mixture temperatures were lowered to 100 F, the so-called sensitive fuel would probably show a larger improvement in performance as compared to that for the reference fuel. This relative behavior of the fuel in respect to severity of engine conditions is called "fuel sensitivity."

The various hydrocarbons of the fuel also have different characteristics with respect to their preignition limits. It has been stated that, in general, a sensitive fuel is also more subject to preignition. In Fig. 20, an excerpt from published literature,[2,4] it is shown that there are distinct differences in preignition limited performance for various components of fuel, some of which are found in aviation gasoline. In general, the addition of tel seemed to raise the preignition level, but the hydrocarbon itself seemed to be the main factor in establishing the preignition level.

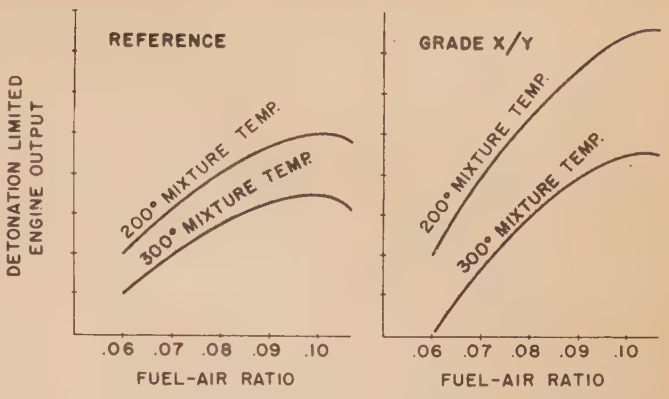
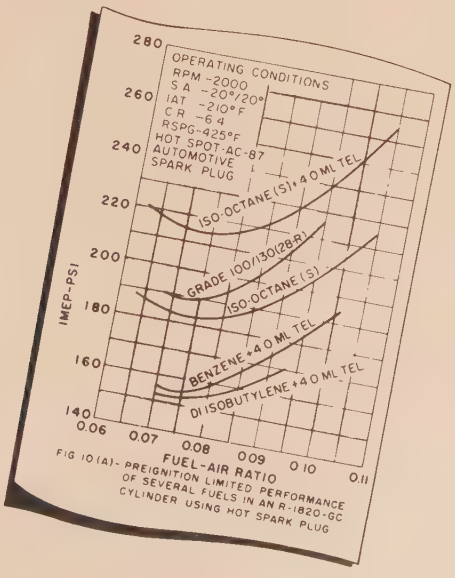


Fig. 19—Fuel performance loss with increased severity

Although fuel sensitivity and preignition limits are not specified in the present fuel specifications, there are some inherent controls already incorporated, as indicated in Fig. 21.[5] Aromatics as a class are not considered as good for preignition resistance. These are limited by the heat value of the fuel. Olefins, which are considered poor for preignition resistance, are limited by their deterioration properties. Aromatic amines, also with poor preignition tendencies, are not permitted in aviation gasolines of today.

It has been recognized for some time that there are these possible variations, or shall we say, tolerances,



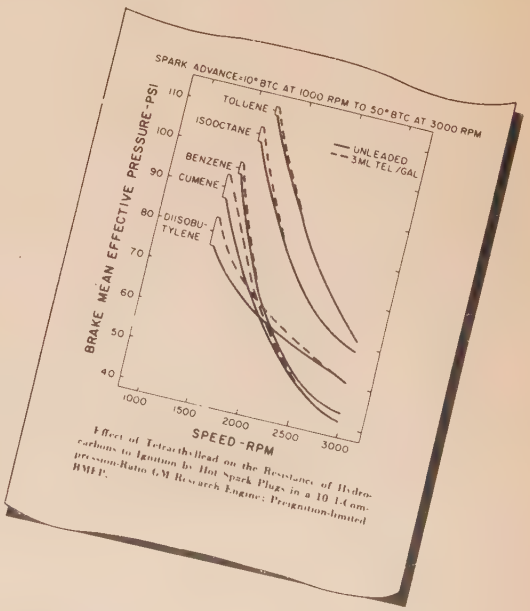


Fig. 20—Fuel components are related to preignition resistance

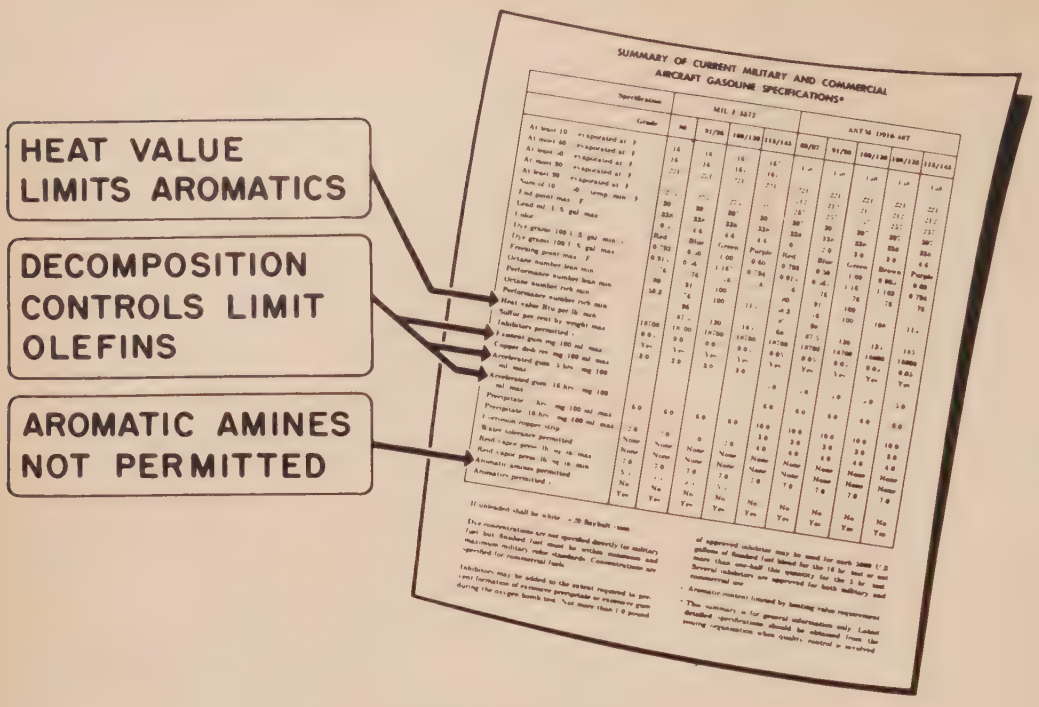


Fig. 21—Control of fuel sensitivity and preignition limits inherent in present fuel specifications

between gasolines of the same grade. There apparently has been no easy solution to control these possibilities closer than they are by the present specifications. It was generally believed that these tolerances should never or rarely cause serious difficulty. However, when frequent serious combustion-chamber difficulties are being experienced, these factors should be reviewed to be assured that the main factors contributing to the difficulty are other than the fuel.

Operation and Maintenance

Before a new engine is released to service a large number of tests have been conducted by the manufacturer to establish operating limits to protect the engine against damaging conditions, including detonation and preignition. Obviously, these limits should therefore be observed and not exceeded. Damaging results from exceeding operating limits may not necessarily become apparent immediately or even in a short time. Damage from detonation or even overheating without detonation is related to the time of exposure as well as the severity.

Overall engine operating limits alone are not sufficient if the requirements of the individual cylinder are not met. Maintenance is necessary to assure that the ignition, fuel injection, and induction systems, as well as the combustion-chamber parts themselves, are properly adjusted and in good working order.

The operating limits also are of little value if the instrumentation for their control is not reliable and maintained properly.

Table 1 lists the interrelated factors which affect normal combustion temperatures, detonation, or preignition. The corresponding operating items for their control are also listed. Each will be reviewed in respect to their relative importance.

Power conditions controlled by bmep, map, and rpm, are usually limited by basic engine structural design or detonation tendencies of the fuel used. The importance of accurate torquemeters and bmep gages should be apparent. Maintenance controls are required to prevent low power from individual cylinders at the expense of others to maintain a desired engine power. The manifold pressure limits specified provide some limited protection, but these may not be adequate to provide sufficient margin for detonation or other damaging effects if abnormal bank to bank power loading is present. The effects of bank to bank distribution will be discussed later.

Of all the factors listed, fuel/air ratio is probably the most sensitive. The higher powers where richer mixtures are used, leaning out of fuel/air ratio to the engine as a whole or to the individual cylinder, may become harmful by causing higher combustion temperatures and possibly detonation. Thus, leaky fuel-injection lines or malfunctioning fuel-injection plungers are probably most damaging at take-off or

Table 1—Detonation, Preignition Factors, and Operating Limits

Power: Bmep, map, rpm
Fuel/Air Ratio: Fuel flow
Combustion Charge Temperature: Cat, blower ratio
Combustion Chamber Temperature: Cht
Spark Advance: Manual control

climb conditions. At cruise conditions, where leaning out well beyond stoichiometric mixture is recommended, further leaning tendency will ordinarily reduce the temperatures and the tendency for detonation, provided distribution between cylinders is satisfactory.

An interesting phenomenon in this regard is that, if conditions of detonation are experienced, it may be controlled by enriching or, if power conditions permit, by leaning out further. On the individual cylinder there is a tendency for the so-called detonation range to center about stoichiometric fuel/air ratios. Thus, in general, more detonation margin as well as cooler operating temperatures are realized at cruise fuel/air ratios, leaner than stoichiometric mixtures. The importance of this must not be overlooked in manual leaning for cruise settings, particularly when other variables may be at or near maximum conditions.

Combustion-charge temperature, a very sensitive factor of detonation is affected directly by the intake pipe air temperature which, in turn, is dependent upon supercharge ratio and carburator air temperature. In high supercharge ratio, the charge temperature tends to be higher, but this is ordinarily offset by the cooler carburetor air temperatures and the decreased exhaust back pressure of higher altitudes where the high ratio is required. Heating of the carburetor air, however, is sometimes required to avoid icing of the induction system, and under these conditions in high blower the charge temperatures will be maximum. It is, therefore, very important that the instrumentation for measuring carburetor air temperatures indicate an accurate average reading, particularly if stratification of hot and cold air exists. Carburetor air temperature limits in high blower at low altitudes are more critical to detonation than at higher altitudes, because at higher altitudes there is improved exhaust gas scavenging due to decreased exhaust back pressure and a corresponding reduction in detonation tendency.

The cylinder-head temperature is a practical but not necessarily a direct indication of combustion-chamber temperatures. The internal temperatures of the combustion chamber are affected more by the internal conditions than by changes in the cooling air as may be reflected by the bayonet thermocouple. However, for a given set of conditions, changes in head cooling will have a definite effect on detonation. For example, on a single-cylinder engine operating at excessive power limits, a change in head temperature from 425 to 500 F changed a condition of no detonation to very harmful detonation. Use of incorrect or bent cylinder-head themocouples, which did not bottom against the cylinder head, have been found in service and may have been a factor in causing some cases of piston burning.

Spark advance as discussed previously, if advanced improperly, will encourage detonation and possibly preignition at the higher powers. Normally spark advance is permitted at cruise powers when auto-lean or cruise manual leaning is used.

All the factors discussed here are important to detonation or preignition margins, but in summary it may be said that the fuel flow or fuel/air ratio and carburetor heat controls are probably the most critical.

The distribution of fuel to the respective cylinders of the engine has been found to be a factor in the reduction of piston burning in service engines. This will be reviewed briefly.

The engine is equipped with two fuel injection pumps, each having nine plungers. One pump supplies fuel to the rear bank of cylinders, the other to the front bank (Fig. 22). The length of the synchronizer bar connected to control levers on each pump controls the distribution of fuel flow to each bank. Thus, making the length of the synchronizer bar shorter than the setting established will enrich the rear bank and lean the front bank. Also, unequalized flows will be experienced if the pumps are not calibrated under similar conditions, or if wear in their control mechanism takes place.

To understand the significance of unequalized fuel flow, one must be familiar with variations of power and cylinder-head temperatures that are experienced by a cylinder when leaning out the mixture at condi-

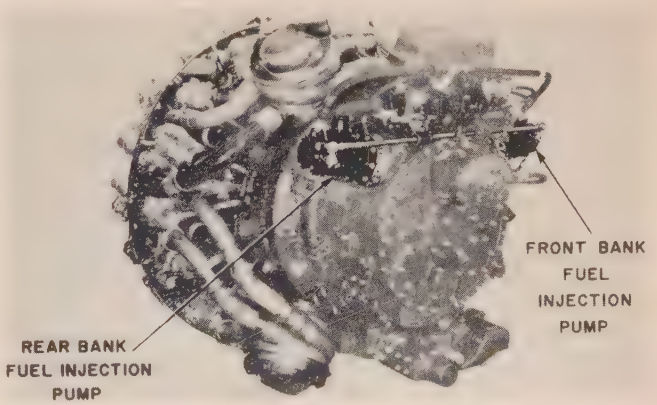


Fig. 22—Synchronizer bar which controls fuel flow to each bank

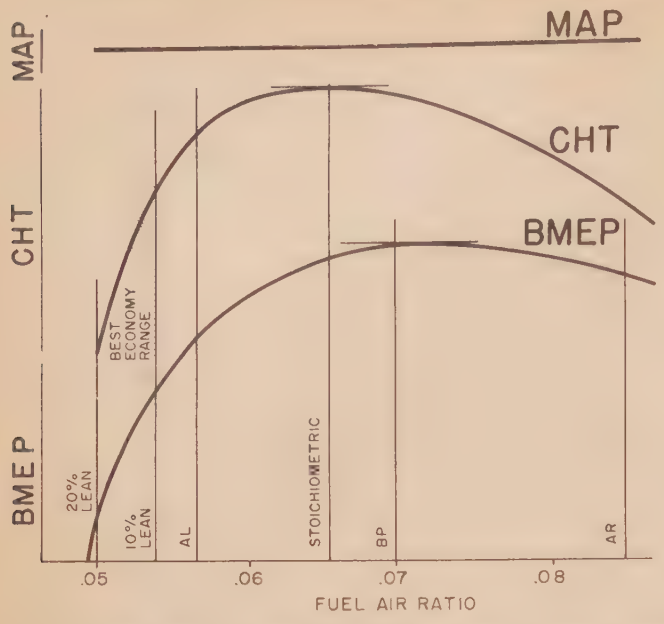


Fig. 23—Variations of bmep and cylinder-head temperatures for cruise power conditions (fixed throttle)

tions of constant speed and constant supply of air or manifold pressure. These variations are illustrated in Fig. 23 for cruise power conditions. At the rich setting of the master control the fuel/air ratio for the whole engine is above 0.08. As the mixture is leaned the power or bmep curve reaches a peak. This point is known as the best power point. As the mixture is leaned further the cylinder-head temperature continues to rise although the power will start to drop off. The temperature will peak between auto-rich and auto-lean at a point somewhere around stoichiometric mixture. On leaning further, both power and cylinder-head temperatures drop at an increasing rate. For cruise operation, a fuel/air ratio where the power drop is approximately 10% below that realized at best power conditions is sometimes recommended. The fuel/air ratio corresponding to 10 to 15% drop is the approximate region for best fuel economy.

Fig. 24 illustrates the procedure used for leaning out to the 10% lean position as the flight engineer would do it. The desired hp or bmep for cruise is set with the master control in auto rich. The engine speed is set and maintained constant by the propeller governor control. With fixed throttle, the mixture control is leaned out to best power. This increases the hp higher than desired so the throttle is retarded to bring the power back. At this point the engine is operating at best power fuel/air ratio for the desired hp since the master control maintains a constant fuel/air ratio. The mixture control is then leaned out further with the throttle and map constant until a 10% drop is indicated on the bmep gage. The throttle is then opened to increase the map and to regain the power lost. Here again the fuel/air ratio will remain approximately the same. This point is called 10% lean. Note the shape of the map curve for desired bmep and that if the engine were leaned out further to, say, 20% power drop, the manifold pressure would have to be increased considerably to re-

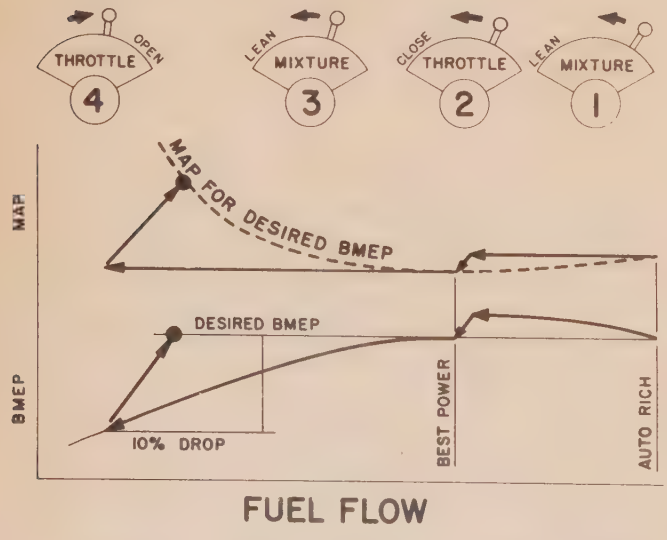


Fig. 24—Manual leaning procedure as the flight engineer would do it

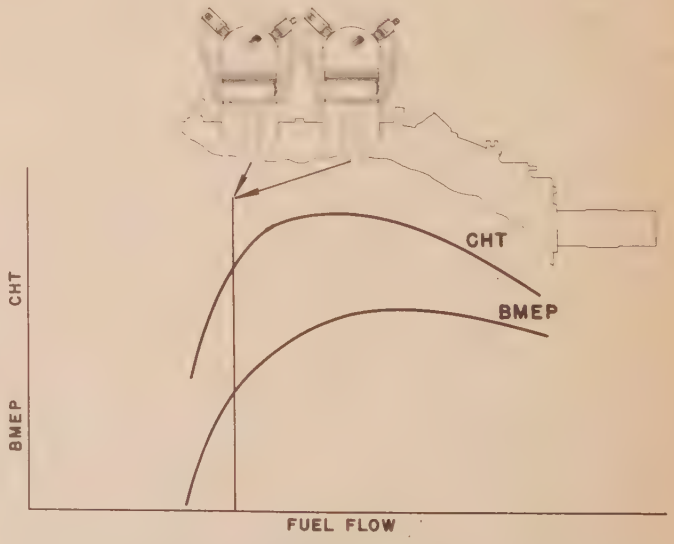


Fig. 25—Equalized fuel distribution between cylinders of two banks means each will be operating at same fuel/air ratio

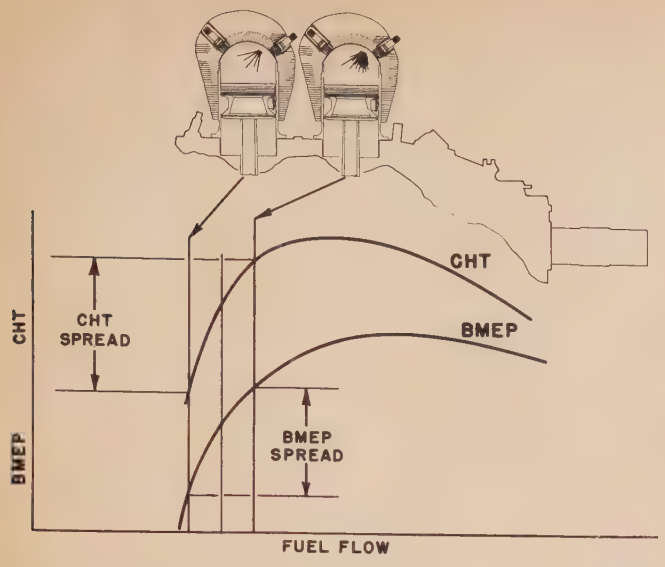


Fig. 26—Unequalizing fuel flow means richer bank runs hotter and supplies more power

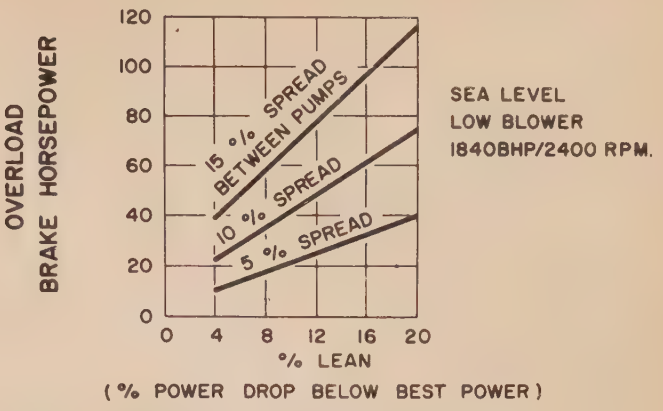


Fig. 27—Calculation of rich bank overload for various unequalized flows and per cent leaning

gain the additional power lost.

Consider now a double-row engine operating at approximately the 10% lean cruise condition. For simplicity of discussion we will assume equalized air distribution between all cylinders of the engine. With equalized fuel flow between cylinders of the two banks, each will be operating at the same fuel/air ratio, as illustrated in Fig. 25. This would mean that the power and the cylinder-head temperature would be the same for each excluding other installation variables. The fuel flow as indicated in the cockpit of the airplane would indicate twice the flow for each bank of cylinders.

Changing the synchronizer bar or unequalizing the fuel flow by any other means, causes one bank of cylinders to run richer, that is closer to stoichiometric than the other, and because of the slope of the curves it will run hotter and supply more power than the lean bank. (See Fig. 26.) All three of these factors are in the direction to cause higher combustion-chamber temperatures and possible detonation with other conditions marginal. Because of the progressive drop off of the curves, if leaning beyond 10% is done, say to 20%, with the same unequalization of fuel pumps, the difference in power and temperature between the two banks is amplified.

The overload bhp which the rich bank would sense for various conditions of per cent leaning and for various percentages of unequalized bank to bank fuel flows was calculated. These are shown in Fig. 27. Note that with a 15% fuel flow spread and at 20% leaning, the rich bank of cylinders is operating with an overload of 115 bhp. This is as though the engine were rated at 230 hp higher than it is.

After extensive work with various airlines, normal cylinder-head temperature spread limits were established between cylinders in each bank at the 10% lean point so that bank to bank fuel distribution check could be made. These are shown in Fig. 28. Installation cooling differences between the two cylinders of each row were recognized in establishing these limits. Thus, for equal distribution, the rear row cylinder would read 10 to 15 C hotter than the front row. A 25 C maximum spread with the rear head hotter was established as a limit for the rear row rich. For a limit of front row rich, the maximum spread was set at 10 C with the front head hotter.

In addition to better controlled bank to bank fuel distribution, it should be mentioned that other changes were made which may have contributed to the reduction of piston burning. Among these are included such items as minimum cowl flap openings

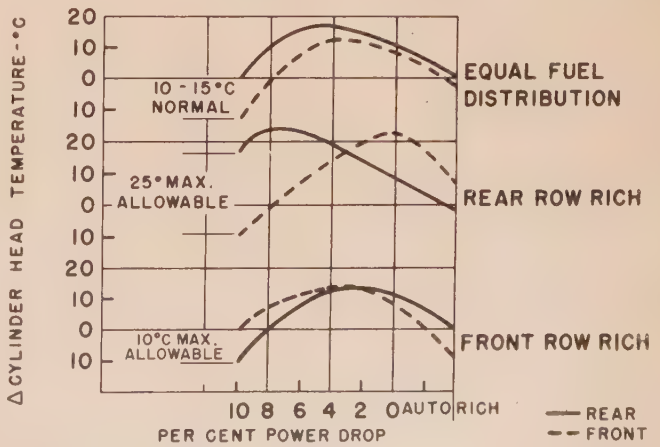


Fig. 28—Control bank distribution by temperature spread

to insure cooling in the event of incorrect cylinder-head temperature readings, reduction in spark ignition advance, correction of bmep gage calibrations, improvements in fuel injection pump calibrations and control mechanisms, and elimination of leaning in excess of 10% which aggravated bank to bank power spreads when unequalized distribution was present. Improvements have also been released by the manufacturer to improve the structural strength of the piston.

In this paper emphasis has been placed on the importance of careful controls in operation and maintenance. These are definite requirements if engine powers are to remain where they are today. Improvements have reduced the problem but the industry as a whole has not as yet reached the point where the difficulty is completely eliminated. With the continued efforts of all who are involved, including the engine manufacturer, accessory manufacturer, fuel and oil suppliers, maintenance and operations personnel, it may be possible to minimize those tolerances which, if allowed to accumulate, create the piston burning problem.

Summary

1. Piston burning (a hole through the piston) can be caused promptly (less than a minute's time) by the excessive heat associated with preignition or by operation under prolonged or severe conditions of detonation and overheating.

2. Preignition can be induced under normal operating conditions by a cracked spark-plug ceramic insulator. Insulator core leakage may also cause preignition.

3. Variations of fuel sensitivity and preignition resistance are possible factors in the piston burning problem.

4. Strict observance of established operating limits and procedures are important factors in reduction of the piston burning problem. All limits are important, but fuel flow and carburetor air temperature are probably the most critical at cruise powers for conditions of temperature and detonation.

5. Maintenance control, particularly for correct fuel and air distribution to each cylinder have been demonstrated in service to be a major factor in reduction of the piston burning problem.

6. Continued efforts by all involved are still required for a further reduction of the piston burning problem in service operation.

Additional References

1. "Relation of Preignition and Knock to Allowable Engine Temperatures," by A. Biermann and L. C. Corrington. NACA ARR 3614 (July, 1953).

2. "The Effect of Engine Variables on the Preignition-Limited Performance of Three Fuels," by D. W. Male. NACA TR 1131 (September, 1946).

3. "The High-Speed Internal-Combustion Engine," by Sir H. R. Ricardo. Pub. by Blackie, London, 1953.

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DISCUSSION

Discusses Functioning of Spark Plugs

A. Candelise

AC Spark-Plug Division, GMC

MR. ANDERSON states that the "spark-plug condition is probably the most important factor involved in preignition." A discussion of the role of the spark plug may be pertinent at this time.

The spark plug performs in an engine:

1. Under normal conditions, that is, when there is no overheating of engine components, or

2. Under abnormal conditions, when there is overheating of engine components caused by a disruption of the thermal balance.

The spark plug is not the source of preignition because it has been selected by the engine manufacturer with that very purpose in mind.

The engine manufacturer selects a plug cold enough, or with an imep rating such that the maximum temperatures encountered in the combustion chamber will not cause the ceramic of the plug to get hot enough to induce preignition.

The plug type must go through suitable qualification tests before it is specified as an approved part of the engine. Obviously, one may question whether the selection of a plug by the heat-

rating procedure, the testing conducted by the spark plug manufacturer, and the engine qualification test offer enough assurance that field conditions will prove, in all instances, that a wise selection had been made. The experience gathered from the analysis of a great number of spark plugs returned after field runs indicates that in almost all cases the used plugs have a greater margin of preignition safety than the new ones. The reason is because the combustion gas deposits tend to fill the clearance between the center electrode and ceramic, as well as the crevice between the ceramic tip and the shell, thus causing better heat dissipation from the firing end of the plug toward the cooler upper regions.

Spark-Plug Malfunctioning

One may also question whether spark-plug malfunctioning may be the origin of preignition. The malfunctioning can be classified as:

1. Failure to provide the necessary spark because of fouled conditions of the ceramic tip.

2. Failure to provide the necessary spark because of various causes which affect the structural soundness of the plug, such as, manufacturing faults, rough handling, or fatigue.

The malfunctioning caused by fouling may, in some cases, cause one of the two spark plugs of a given cylinder to become totally inoperative. A combustion chamber operating with a single source of ignition may aggravate an incipient detonation condition because of the lengthening of the combustion time. Prolonged and severe detonation, besides being dangerous in itself, may end up in preignition. The spark-plug fouling control is outside the boundary of this discussion, but it suffices to mention that the plug design, its heat range, the engine operating conditions, and the fuels used are the main variables involved. A plug may also become inoperative because of failure of some other components of the ignition system; the effect on preignition possibilities may be similar to the fouled plug condition.

Structural spark-plug failures are seldom encountered because of the high degree of workmanship embodied in the aircraft spark plug of today. However, an occasional failure, caused by manufacturing faults may be encountered. These are minor items, which, to our knowledge, are not a general cause of preignition or engine malfunctioning. An occasional failure due to rough handling is also encountered.

It can be concluded that under normal engine operating conditions an approved plug cannot be considered as a potential source of preignition with the possible exception of aggravating preignition induced by detonation when one of the two ignition sources fails to operate.

Operation under Abnormal Conditions

The first question one may ask oneself is, "What caused the abnormal conditions, especially related to preignition?" We said before that normal engine operation will not cause the plug to preignite, consequently, it must be assumed that some other engine parts or operational settings must be responsible.

When preignition sets in, a violent temperature rise of the exposed parts takes place; the plug, the valves, and the piston are the most vulnerable. The plug may overheat to the point that the electrodes may fail or the ceramic may crack or break. At the same time, the valves may suffer, and the piston may be severely damaged. Each one of these failures may occur simultaneously or singly. Our experience shows that plug failure caused by preignition may occur without any damage to engine parts, and conversely, many failures of engine parts may occur without any damage to the plug. Evidently, the condition of the parts prior to the onset of preignition is the determining factor.

We have been making and selling aircraft spark plugs since the inception of the ceramic type. We have had the opportunity of analyzing many plug failures caused by preignition throughout the years. Plugs with a good portion of the centerwire missing, with cracked ceramic, and with burned ground electrodes were analyzed for preignition characteristics and found to be in safe condition. Probably some extreme cases of jagged broken ceramics or electrodes may be conducive to preignition, but again, these very rare cases would not, in general, help in solving the reasons for preignition or piston failures.

Series of Questions

I would also like to comment on some of the author's illustrations, and ask him a few questions:

1. In regard to Fig. 2, shouldn't the *p-t* card of preignition show higher pressure than that shown on detonation? Shouldn't it also show more heat discharge to parts?

2. Referring to Fig. 8 (direct causes of preignition), we have found shorted distributor segment plates to have caused preignition in many cases, do you concur?

3. The spread in temperatures between cylinders, shown in Fig. 12, may be caused by variations in fuel-air distribution as well as other conditions mentioned. Dist. test based on this fact.

4. In Fig. 13, the ceramic is *broken,* not cracked.

5. You explained the manner in which the exhaust valve is cooled. Don't you feel that the insulator of a plug, of much less mass, is cooled in the same manner? (Refer to Fig. 15.)

6. The exhaust-valve dome temperatures shown in Fig. 16 were stated to be average, so are plug temperatures which may not directly reflect the cooler temperatures during compression stroke.

7. In your summary you say, "If combustion-chamber temperatures should be increased abnormally, it is probable that the spark plug would be the first to cause preignition." I don't believe you've shown it to be so. It may be *possible* in some cases, but not necessarily so. We have had and seen many instances of preignition where the plug definitely was not the source.

ORAL DISCUSSION

Reported by J. T. Hendren

Pan American World Airways, Inc.

W. R. Houser, AC Spark Plug Division, GMC: Could shorted segment plates be one cause of preignition?

W. W. McClintock, Pan American World Airways, Inc.: We have had cases of shorted segment plates, but they are considered isolated cases and are a function of overhaul shop technique. Have you observed any cases where a burned piston was definitely associated with a leaking plug?

Mr. Anderson: I have seen none in service, but it can be done in the lab. Overtorquing the plugs can lead to leakage, and it is important to observe torquing values.

Mr. Houser: Leakage of plugs upsets heat flow, and anything that upsets heat flow can lead to preignition.

Weld Repair of Aircraft

THE purpose of this paper is to review the equipment, techniques, and recommendations for the successful repair of magnesium aircraft castings, both sand and permanent mold. The discussion will cover both the repair of castings with service experience and also the techniques used for the salvage of production castings by welding in the foundry of our Bay City Division.

Magnesium is well known for its extreme lightness, excellent machinability and weldability as well as the high strength-to-weight ratio of its alloys. It has a sp gr of 1.74 and weighs 109 lb per cu ft or approximately 1 oz per cu in. This means that magnesium is roughly one-fifth the weight of steel and two-thirds the weight of aluminum. Pure magnesium melts at 1204 F and boils at 2007 F. Its degree of contraction from a liquid to a solid at its melting point is about 4.3% and from a liquid at its melting point to a solid at room temperature, about 9.7%. The average coefficient of thermal expansion for magnesium alloys in the range 65-750 F is about 0.000016 in. per in. per F. The total heat for fusion is approximately 89 cal per g or 160 Btu per lb.

Because of a comparatively low melting point, latent heat of fusion and specific heat, the amount of heat required to melt magnesium is relatively small. On an equal volume basis, the total heat for fusion is approximately two-thirds that for aluminum and

Table 1—Commercial Magnesium Alloys

Alloy Designation		Nominal Composition, % (Mg=Remainder)				Typical Mechanical Properties			
ASTM	Dow	Al	Zn	Mn	Temper	Tensile Strength, psi	Tensile Yield strength, psi	Compressive Yield strength, psi	Elongation, % in 2 in.
AZ63	H	6.0	3.0	0.5	-T4	40,000	12,500		14
					-T6	40,000	18,500	Essentially the	5½
AZ92	C	9.0	2.0	0.15	-T4	40,500	13,500	same as the	10
					-T6	39,000	20,000	tensile yield	2½
AZ91C	AZ91C	8.7	0.7	0.13	-T4	41,000	12,500	strength	14½
					-T6	41,000	18,500		5½
AZ81XA	AZ81XA	7.6	0.7	0.13	-T4	39,000	12,000		14

Quality Magnesium Castings

W. A. Beck, Dow Chemical Co.

This paper was presented at the SAE Annual Meeting, Detroit, January 11, 1956.

GENERAL techniques used in the weld repair of magnesium aircraft castings by both gas and arc welding methods are described here. Coverage is given to equipment, methods of surface preparation, preheating, welding procedure, post-heat-treatment.

Some typical arc weld repair jobs are described and illustrated. Included also are descriptions of weld defects encountered because of poor welding technique. Typical mechanical property data are shown.

one-fifth that for steel. The high coefficients of thermal expansion and conductivity tend to cause considerable distortion when welding magnesium unless proper precautions are taken. In this respect, magnesium is more critical than steel and behaves much like aluminum. Another important characteristic of magnesium from the welding standpoint is its rapid oxidation when heated to its melting point in air. For this reason, the arc or gas welding of magnesium requires the use of a protective shield such as an inert gas when arc welding or a flux when gas welding.

Magnesium and all of the commercial magnesium alloys produced in the form of sheet, plate, extrusions, and sand or permanent-mold castings are weldable. The commercial welding processes most widely used are inert gas metal-arc welding, gas welding, and spot welding. Other processes used include brazing and soldering, and to a lesser extent seam, flash, and pressure welding. As the title indicates, this paper will cover only the welding of castings which is accomplished by both arc welding and gas welding. However, arc welding is *the recommended* process for weld repair of magnesium *aircraft* castings and the emphasis throughout this paper will be on that particular method. Field repairs are often made by gas welding but we feel that this method should be considered for temporary or emergency use only.

We will close this introductory material with a brief look at some of the common casting alloys of magnesium and their characteristics. Table 1 lists the most commonly used magnesium alloys and their designations, compositions, and mechanical properties. This table shows only the commonly used alloys of magnesium which are the alloys of the magnesium-aluminum-zinc system. There are, of course, two new systems of alloys which were developed in recent years and are finding increasing usage. These are the magnesium-rare-earth-zirconium and the magnesium-thorium-zirconium alloys. Since the Mg-Al-Zn alloys are the ones most commonly en-

Fig. 1—View of sand cast magnesium aircraft brake carrier showing part marked for welding. Part is 13½ in. in diameter by 5½ in.

countered in the field, this discussion will be confined to them. It should be pointed out, however, that the newer alloys are quite weldable and the general techniques described here are also applicable to them. Details of recommended welding, pre- and post-heat-treatments for the Mg-rare earth and Mg-thorium alloys may be readily obtained from Dow.

Table 2 is a compilation of physical properties of the same magnesium alloys listed in Table 1.

General

Arc welding is the recommended process for weld repairing magnesium aircraft castings, although field repairs are often made by gas welding. Weld repair should be limited to the repair of defects in clean metal, such as broken pieces, mutilation due to mishandling, and corroded areas where such corroded areas consist of definite pitting. At the Dow Bay City foundry, weld repair is restricted to such clean defects as mutilations in clean metal due to riser break-off, chipping, sawing, grinding or bruising, and bubbles, sand holes, and pits. Misruns are also weld repaired if these defects occur on extremities but not if they extend through a wall section. We do not weld shrinks, blows, and porous areas since these defects are usually spongy and will tend to blow bubbles during the fusion welding operation.

In the field, an impregnated casting, if porous, can be expected to give a lot of trouble during welding. If the welding is performed in a sound metal area then no great difficulty should be encountered since there is no microporosity present which would have been filled during the impregnation process. Oily and greasy castings may also cause a lot of difficulty. If the oil penetrates a porous area, and that area is later welded, there will probably be a lot of trouble with bubbling as the gas emerges during the welding operation.

Weld repair of castings should not be attempted without consulting or gaining the approval of the manufacturer of the part. Weld repair should be done only in those areas approved by the manufacturer of the article. Fig. 1 shows a typical magnesium part which has been clearly marked by the manufacturer to indicate which areas of the casting are weldable and nonweldable. This magnesium casting is for an aircraft brake carrier. Such information, in the form of marked prints or photos, is readily obtainable from the part manufacturer.

Gas Welding

At our Bay City foundry we do not use gas welding for the repair and salvage of magnesium castings. It is our recommendation that gas weld repair be held to the direst of emergencies since it is very hard to clean the welding flux used from the casting. This is undesirable in aircraft castings, of course, and particularly so in Naval aircraft castings.

Magnesium parts which have broken or cracked usually can be sufficiently repaired by gas welding to enable the part to be placed back into use. Generally speaking, such repair should be considered only as a temporary measure until a replacement part can be obtained.

The first step in making a gas weld repair on a magnesium casting is to clean the casting thoroughly from grease, oil, dirt, chemical coatings, paint, and the like. This can be done by the use of alkaline cleaners, carbon tetrachloride, gasoline, or soap and water.

After cleaning, the immediate area in which the repair is to be made must be prepared. The edges of the hole, crack, or other type of defect must be dressed down to clean, sound metal. This may be accomplished by routing, filing, chipping, scraping, or similar means. The slope of the sides of the defect should be about 45 deg. If a defect penetrates the

Table 2—Physical Properties of Magnesium Alloys

Alloy	Approx. Solidification Range, F	Thermal Conductivity (cgs Units) at 212 F	Specific Heat per Unit Volume (68-212 F)	Approx. Thermal Diffusivity per Unit Volume	Electrical Resistivity at 68 F, microhm-cm	Density, lb/cu in.
AZ63	1130—685	0.18	0.46	0.39	13.0	0.067
AZ92	1110—770	0.16	0.45	0.36	17.0	0.066
AZ91C	1105—875	0.16			17.0	0.066
AZ81XA	1110—855	0.16			15.7	0.066

wall of the casting, a land or root face of about 1/16 in. should be left on the bottom side. The surface of the casting around the area to be repaired should also be scraped, sanded, filed, or wire brushed, to be certain that all oxide, paint, or other coatings are removed.

Data concerning welding fluxes are shown in Table 3. Welding fluxes for magnesium are supplied in dry powder form and are prepared for use by mixing two parts of the powder with one part of water. Welding fluxes for magnesium are hygroscopic and will take on moisture. Containers of flux should be kept tightly closed. Only enough flux paste for one day's use is mixed at a time and the paste too is kept in covered glass containers when not being used. Of the two fluxes shown in Table 3, Dow 450 has excellent welding characteristics with oxyacetylene gas but is more likely to cause pitting of the bead than Dow 460. If the bead is not to be ground, or if bead appearance is important Dow 460 with its lessened ease of welding should be used with oxyacetylene gas.

The first step in the welding operation is to apply a flux paste with a small brush to the edges of the area to be welded. Before starting to weld, the torch should be adjusted to a neutral or slightly reducing (carburizing) flame. The torch should be adjusted to a 1/4 in. inner cone or until such inner cone slightly "feathers" at the tip when the oxygen is lowered.

The casting to be gas welded should be preheated with the torch or in a furnace before actual welding is started. The entire casting should be brought up to a preheat temperature of about 650 F. This is generally not feasible in the field because of the castings being machined, containing inserts, and the like, so that distortion cannot be tolerated. Local preheating would be used. A mark made on the casting with ordinary blue carpenter's chalk will turn white when a temperature of approximately 600 F is reached. Tempilsticks or temperature indicating crayons may be used if available. Actual welding is started by slowly preheating a small area with a fanning action of the welding flame. As the metal begins to melt, the welding rod is dipped into the molten puddle and then withdrawn as sufficient molten metal is obtained. The rod is held in the outer flame and is dipped into the puddle intermittently, the torch being moved forward as rapidly as possible. The flame should be moved forward in a straight line, the torch being inclined at about 45 deg to the work. All welds should be made in one continuous pass, if possible, since this keeps the surrounding metal up to temperature. If insufficient heating is being maintained, as indicated by cracking, a larger tip size is indicated.

Welders working with magnesium alloys should, of course, wear gloves and the usual type of welding goggles. Blue glass gives best visibility during welding.

After welding is completed, the casting should be placed in a furnace for 1 hr at 500 F for stress relieving. If no furnace is available, a gas flame can be played over the entire casting to bring it up to a temperature of about 650 F, after which the casting should be allowed to cool slowly, away from drafts.

Table 3—Welding Fluxes for Magnesium

Dow No.	Use	Type	Package Net Weight, lb
450	Oxy-acetylene welding	Glass jars	1/4 and 1
460	All gas welding	Glass jars	1/4 and 1

When the repaired casting is cool enough to handle with bare hands, all excess flux should be removed by thorough scrubbing with a stiff bristle brush and plenty of hot water. Next, the repaired parts are immersed for 1 min in a chrome pickle solution consisting of 24 fluid oz of nitric acid and 24 oz of sodium dichromate plus water to make 1 gal of solution. After chrome pickling, the parts are washed in cold running water then boiled for 2 hr in a solution of 8 oz of sodium dichromate to 1 gal of water. Parts are carefully rinsed and dried, and are then ready for the final finishing system.

Detailed instructions for gas welding procedures are contained in Dow publications: Technical Memorandum No. 10 d, "Emergency Repair of Magnesium Parts by Gas Welding" and Bulletin No. DM 94A, "Joining Magnesium."

If standard welding rod is not available, it is possible to use strips cut from sheet, extrusions, or castings of the same alloy to be welded. Such strips should be thoroughly cleaned immediately before use with steel wool, emery cloth, or by scraping.

Fig. 2 shows a typical gas welding repair operation.

Arc Welding—Equipment

As pointed out earlier, arc welding is the recommended process for weld repairing magnesium aircraft castings. In the Dow foundry we use a P&H 30-285-amp d-c welder, Model WA200, equipped with a variable foot control called "Arctrol," which is made by the Mullenbach Mfg. Co. of Los Angeles, Calif. This is certainly not suggested as the only make or type of acceptable welder or foot control. However, in general, our foundry feels that a d-c welder will give a quieter weld puddle and so make weld repair easier. There are now on the market a-c machines called "balanced wave" machines. These are the quietest or smoothest type of a-c arc welding machines and will probably give very good results in the weld repair of magnesium castings. An a-c welder will give deeper penetration but this is not required on castings.

The d-c welder is hooked up as direct-current reverse-polarity (DCRP) as indicated in the sketches of Fig. 3. As shown in these sketches in straight-

Fig. 2—Gas welding of magnesium castings is used primarily for emergency repair

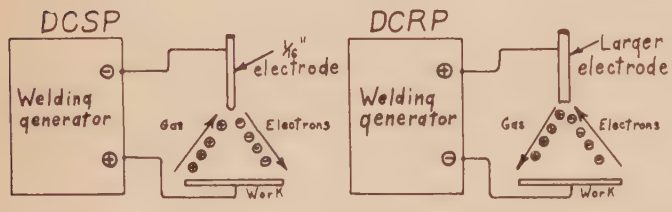


Fig. 3—D-c welding—straight polarity versus reverse polarity

polarity (DCSP) welding the electrode is negative and the work positive. Thus, the electrons flow from the electrode to the workpiece. In DCRP, the connections are just the opposite; electrons flow from the plate or workpiece to the electrode.

In straight-polarity welding, the electrons hitting the plate at high velocity exert a considerable heating effect. In reverse-polarity welding, just the opposite occurs; the electrode acquires this extra heat, which then tends to melt off the end of the electrode. Thus, for any given welding current, DCRP requires a larger diameter electrode than DCSP does. Straight-polarity welding produces a narrow, deep weld, whereas reverse-polarity current gives a wider, shallower weld. One effect of DCRP welding is the so-called plate cleaning effect which seems to occur. The probable explanation for this cleaning effect is that the electrons leaving the plate and the gas ions striking the plate tend to break up the surface oxides, scale, and dirt usually present. This is beneficial to the welding of magnesium.

Fig. 4 shows the d-c welding machine used at our magnesium foundry. The foot-controlled rheostat is shown on the floor in the right foreground. Other equipment items shown are the Heliarc torch, a gas torch for heating the work, the inert gas supply (helium), a supply of welding rods, and extra Heliarc torches.

Our welding department feels that a foot control is almost a necessity for successfully repair welding magnesium castings at high efficiency rates. Most cracks which occur in magnesium castings during weld repair operations start at the instant the arc stops. Our welder tries to get around this by two means:

1. He likes to continue welding the area until the complete weld repair is finished once he has struck his arc. He likes to continue this even if it means someone has to stand next to him and hand him one welding rod after another.

2. When he does finally bring his weld to completion he eases back on his foot-controlled amperage controller and gradually fades out the arc until it dies away. This means there is no great thermal shock when the arc is stopped.

Most castings are not all of uniform section. This means as the weld repair is made the arc may pass from thin to thick section. The thicker the section the more current required to keep the molten puddle of metal of uniform size. Here again the variable foot control comes in handy, since as the arc passes over the heavy section the current can be increased and as the arc works toward a thinner section the arc can be decreased. Generally, the operator will set the current on the machine 10 or 15 amp higher than desired. He can then start slowly using his foot control. During the welding he can vary the current to get more heat if running cold or cut down on the heat if the tungsten electrode begins to "wobble." At the

Fig. 4—D-c welding machine used at magnesium foundry

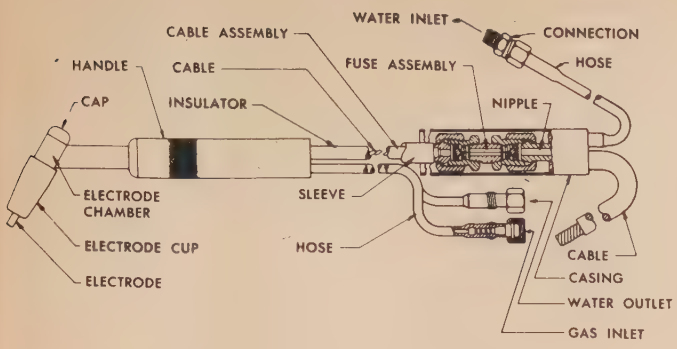


Fig. 5—Diagram of Heliarc torch

Table 4—Welding Rod to Be Used for Weld Repair

Alloy	Correct Welding Rod
H	H
C	C
AZ91C	C
AZ81XA	C
EK30A	EK30A
EZ33A	EZ33A
EK41A	EK41A

Usual rod sizes: Length—36 in.
Diameters—1/16 in., 3/32 in., 5/32 in., 3/16 in.

end of the pass, as mentioned before, he can ease off the current to avoid thermal shock. Our welder feels that this control has paid for itself many times over in castings saved from the scrap pile.

A Heliarc torch similar to that shown in Fig. 5 can be used and is available on the market. This is a water-cooled torch and is particularly designed for long, continuous welds on wrought metals. In the foundry, we use a torch built in our own shop that is not water-cooled and is therefore slightly less bulky. Such a torch would not be satisfactory for long, continuous welds since it would heat up and become uncomfortably hot. Fig. 6 shows a close-up of the Heliarc torch, the foot control, and a number of spare torches with varying electrode sizes.

In our own welding setup we have a steel surface table exactly the same height as the preheat furnace hearth sitting immediately outside the vertical lift door on the electric furnace. The steel on this table is connected to one side of the welding machine. When the casting has been sufficiently heated the door is opened, the casting is dragged out, and the welder is ready to go to work on the casting immediately. Fig. 7 shows an aircraft wheel casting as the welder prepares to remove it from the oven to the surface table.

The correct kind of welding rod for each casting alloy is shown in Table 4. This is the same as for gas welding.

In welding operations, of course, the operator should be properly protected against arc rays. Suitable cloth to cover exposed skin surfaces and a welder's helmet with the proper shade of glass to protect the eyes and face should be worn. The shade of glass depends upon the intensity of the arc, as shown in Table 5.

The operator generally uses a No. 12 shade of glass since his welding currents may run up to the range indicated.

Preparation

Details of surface preparation for arc welding of castings are the same as for gas welding repair. Cast-

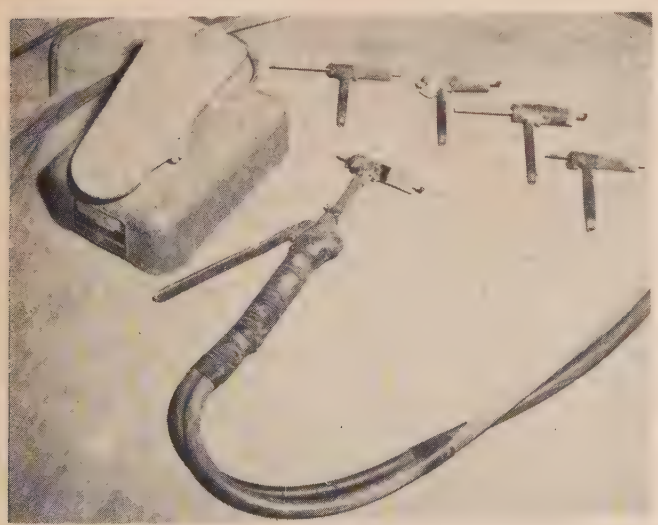


Fig. 6—Various accessory items for welding

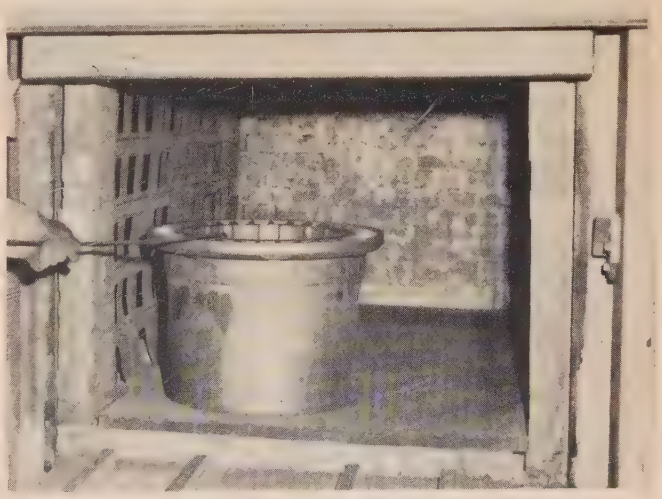


Fig. 7—Aircraft wheel casting about to be removed from preheat furnace to surface table in foreground for welding

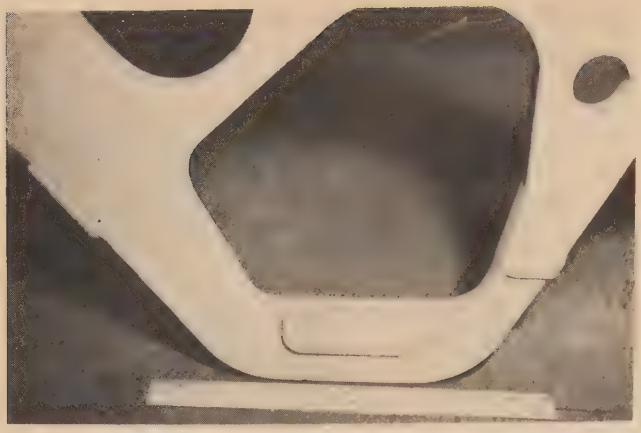


Fig. 8—Magnesium casting before weld preparation

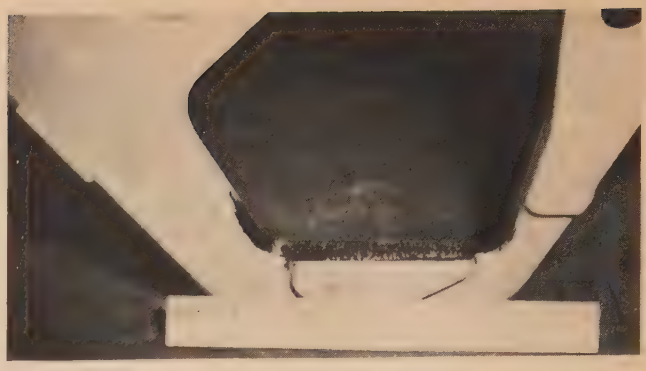


Fig. 9—Casting cleaned for welding

ings should be stripped of paint and degreased before welding. Enough preheat should always be used to be sure that the casting is dry and free of any foreign material in the pores of the castings that might vaporize and cause porous areas in the weld repair. Impregnated castings are very difficult to weld repair successfully. If a porous area in the impregnated casting has to be weld repaired, a good prebake of the casting would certainly be in order to be sure that any volatile matter in the impregnate will have vaporized off so that it will not cause a porous weld repair.

Metal should always be removed in sufficient quantity so that only clean metal is at the surface to be weld repaired. Where large holes in defective areas are to be filled in with weld metal, a backing plate may be used to avoid excessive dropdown of the weld metal. We have found that blocks of cast magnesium can be worked very readily with the hand routing tool to form contoured backup plates that will exactly fit the many complicated curves in a magnesium engine casting. However, welds of this size are not generally attempted in the Dow foundry. The bottom edge of a hole in a casting that is to be weld repaired should have a little lip on it to give something to start the arc on.

In our foundry we do not weld cracks in castings since the applicable specifications do not allow it. If cracks are welded in the field as an emergency repair measure, it is strongly recommended that adequate inspection of the weld repair be made to insure that the weld goes to the end of the crack. Inspection methods involving zyglo, dye-check, and X-ray should be utilized. A good procedure in welding cracks is to drill a hole at each end of the crack to insure that the crack does not propagate further during the welding operation.

Table 6 gives the recommended preheat practices for the commercial Mg-Al-Zn casting allows. Fig. 7 shows a typical wheel casting being removed from a preheat furnace. Most magnesium castings which are thin walled and castings having areas of considerable restraint require preheating to avoid weld cracking. Where welds are to be made in relatively unrestrained areas, such as external bosses or flanges, preheating may be unnecessary or slight local preheating of 300-500 F may suffice. Care should be taken in local preheating so as not to overheat the metal and damage its properties.

An air circulating furnace with a temperature control of plus or minus 10 F at preheat temperature is

Table 5—Relation of Shade of Glass and Intensity of Arc Rays

Glass No.	Welding Current, amp
6	Up to 30
8	30 to 75
10	75 to 200
12	200 to 400
14	Above 400

Table 6—Heat-Treatment of Castings before Welding

Alloy	Temper	Weld Preheat
H	T4	Heavy and unrestrained sections—none or local Thin and restrained sections—720 F (min—650 F)
	T6	Heavy and unrestrained sections—none or local Thin and restrained sections—720 F (min—650 F)
C	T4	Heavy and unrestrained sections—none or local Thin and restrained sections—750 F (min—650 F)
	T6	Heavy and unrestrained sections—none or local Thin and restrained sections—750 F (min—650 F)
AZ81XA	T4	Heavy and unrestrained sections—none or local Thin and restrained sections—750 F (min—650 F)
AZ91C	T4	Heavy and unrestrained sections—none or local Thin and restrained sections—750 F (min—650 F)
	T6	Heavy and unrestrained sections—none or local Thin and restrained sections—750 F (min—650 F)

used for preheating. No special atmosphere is required; however, a separate temperature limit control set at 10 F above the preheat temperature should be used as a safety measure. Solution heat-treated or solution heat-treated and aged castings can be charged to a hot furnace at preheat temperature without damage to the casting. However, as-cast (—F), castings should not be charged into a furnace above 500 F, and are heated to the preheat temperature at a rate not exceeding 150 F per hr between 500 F and the preheat temperature. This heating procedure must be followed in order to avoid incipient fusion of compound. Castings should remain at preheat temperature for a period of time sufficient to assure uniform heating throughout.

Welding Procedure

Castings can be welded in the as-cast (—F), solution heat-treated (—T4), or solution heat-treated and aged (—T6) conditions. Our foundry prefers not to weld in the as-cast condition because it is believed that more cracking will be encountered in that condition. If the casting is to be given a —T2 or stabilizing heat-treatment only then is it preferable to weld the casting in that condition. Otherwise, we believe it best to weld in the solution heat-treated (—T4) condition, if posisble. There is no difference between the —T4 and —T6 conditions as regards cracking tendency.

After castings are removed from the preheat furnace, welding should proceed immediately and many specifications require that the casting temperature not drop below about 650 F. In our foundry, however, we often find it possible to allow the casting to cool down much below this temperature without serious welding difficulties. Both helium and argon can be used as the shielding gas for a-c or d-c welding. With d-c, helium is generally preferred because of the better arc stability obtained. The usual helium flow is about 6-7 liters per min. About half the volume rate of argon is required. Welding currents of 50-140 amp are normally used. It is impossible to tell what a job will require except by experience. If a casting is preheated, about half as much current is required. On very long welds welding should begin at the center of the defect, where possible, and progress toward the outer ends. Medium size weld beads are preferred. Too little heat results in porous welds; too much heat may cause cracking or voids in the parent metal. Filling of holes is usually the most critical type of repair as far as cracking is concerned. The arc should be struck at the bottom of the hole and the welding done upwardly. The arc must not dwell too long in one area, since it makes the weld more susceptible to cracking and encourages the formation of fusion voids in the parent metal.

In welding operations, we find it important to wire brush each pass of a multiple-pass weld. Also, it is necessary that the welding rod be clean. The rod is brushed with steel wool each time before using.

As previously mentioned, it is sometimes advisable to discontinue welding when the temperature of the casting has dropped below 650 F. A casting weighing 50-100 lb, usually can be welded within a period of about 4-5 min after removal from the furnace before its temperature falls below 600 F. When the casting temperature falls below 600 F, the casting usually must be returned to the furnace for reheating before completing the repair. Many castings can be cooled in still air at room temperature after welding without danger of cracking. If of especially intricate design, the casting may require slow cooling in a previously warmed soaking box or pit.

Fig. 8 shows a magnesium frame casting in H alloy which is solution heated, treated, and aged. On this casting it was found necessary to build up a pad ⅛ in. thick on the inside upper surface of the bottom loop shown. Fig. 9 shows the casting cleaned up for welding. This cleanup work was accomplished with a rotary file. The cleaning is simplified in this case, of course, since the casting is relatively new with only a chrome pickled surface to remove. Fig. 10 shows the welder in position for the actual welding operation. Only local preheating with a torch was used on this job. A $\frac{3}{16}$ in. thick pad was built up for later grinding down to the finish thickness of ⅛ in. The pad was built up by putting on a weld bead of almost $\frac{3}{16}$ in. in a single pass alongside each outside edge, and then filling across between these outside ridges in a series of passes. Fig. 11 illustrates the wire brushing operation accomplished by hand after each pass. Fig. 12

Fig. 10—Welding tools in position on partially complete weld

Fig. 11—Weld is wire brushed after each pass

shows the finished welding job, and Fig. 13 shows the welded pad after grinding to the desired rough dimension.

In this welding operation, the operator tried $\frac{3}{16}$-in. weld rod on three castings but ran into cracking trouble. He then switched to $\frac{5}{32}$-in. rod and had no further difficulties. The current setting was 165 amp. The welder used a No. 12 filter glass in his helmet.

Fig. 14 shows another typical repair job. The circled defect on a cast magnesium C alloy gearbox (in the —T6 heat-treat condition) is a chipped-off boss corner. Fig. 15 shows the same defect cleaned up for welding, and Fig. 16 shows the casting immediately after the weld repair and before grinding.

Figs. 17 and 18 show a typical aircraft wheel casting—in H alloy in the —T4 heat-treat condition —before and after weld repair of a defect. The type of defect shown is a very clean defect which can be easily tooled out for welding. The parent metal is very sound.

Post Weld Treatment

Excess metal is removed from welds with a routing tool. Welds are then examined for soundness. Visual inspection with a magnifying glass is satisfactory for detecting small cracks in or around the welds after etching the area with a 10% acetic acid solution. Normal radiographic methods with a 1-3% sensitivity are satisfactory for detecting most critical defects such as incomplete penetration, porosity, and cracks. On welded castings we X-ray all welds using fine-grain film. We also supplement this on occasion with etching techniques. Fluorescent penetrant and dye-check are sometimes used. However, radiography is the preferred method.

Table 7—Post Weld Heat-Treatment of Castings

Alloy	Temper	Post Weld Treatment
H	T4	Heavy and unrestrained sections—12 hr—730 F (Optional —½ hr at 730 F) Thin and restrained sections—As above
	T6	Heavy and unrestrained sections—12 hr—730 F (Optional —½ hr at 730 F plus 5 hr at 425 F) Thin and restrained sections—As above
C	T4	Heavy and unrestrained sections—½ hr at 770 F Thin and restrained sections—½ hr at 770 F
	T6	Heavy and unrestrained sections—½ hr at 770 F plus 4 hr at 500 F Thin and restrained sections—As above
AZ81XA	T4	Heavy and unrestrained sections—½ hr at 780 F Thin and restrained sections—½ hr at 780 F
AZ91C	T4	Heavy and unrestrained sections—½ hr at 780 F Thin and restrained sections—½ hr at 780 F
	T6	Heavy and unrestrained sections—½ hr at 780 F plus 16 hr at 335 F Thin and restrained sections—As above

Note: Temperatures shown are maximum, not nominal ± 5 F as usually shown.

Castings weld repaired in the foundry are especially marked to indicate that they have been welded. This is done by circling the casting with a white paint and also circling the weld itself. Castings are stamped with a serial number for weld identification, and the weld area is identified by stamping the casting adjacent to the weld with three concentric circles with a number identifying the welder in the inner circle. This is probably not too important in the field but some sort of identification is desirable.

Repaired castings are generally heat-treated again after welding. The recommended post weld treatments are shown in Table 7. The heat-treatments depend upon the condition of the castings before welding and the condition desired after welding. Because of the small grain size and fine dispersion of compound in the weld metal, a short heat-treatment time of 30 min is sufficient to obtain complete solution of compound

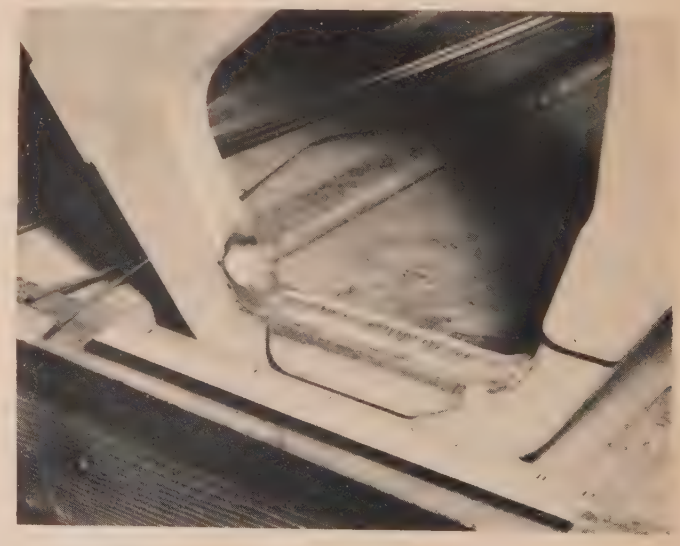


Fig. 12—Casting with completed pad built up by welding

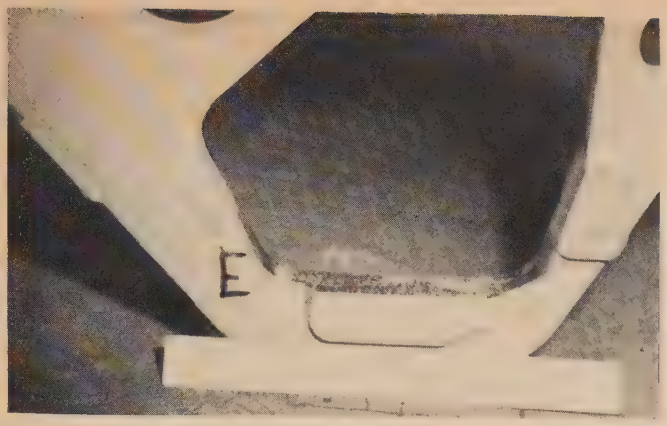


Fig. 13—Casting with welded pad ground to rough final dimensions

Fig. 14—Magnesium gearbox with defect to be welded

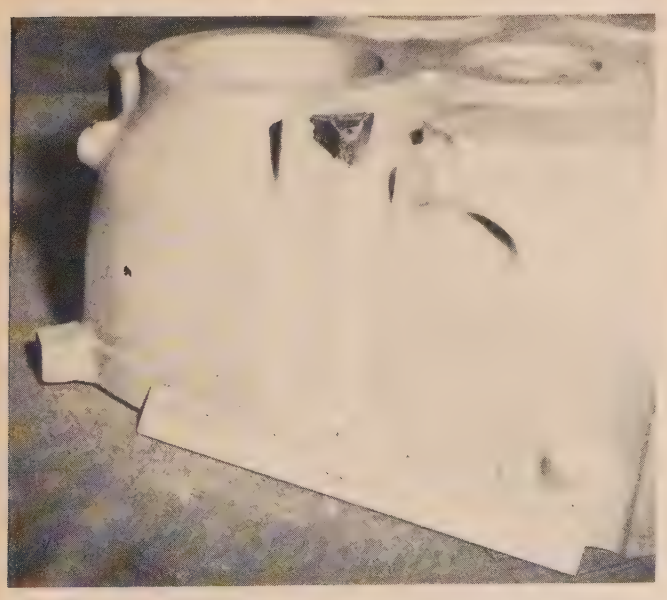


Fig. 15—C alloy gearbox casting with defect cleaned for welding

Fig. 16—Aircraft gearbox following weld repair—bead not ground

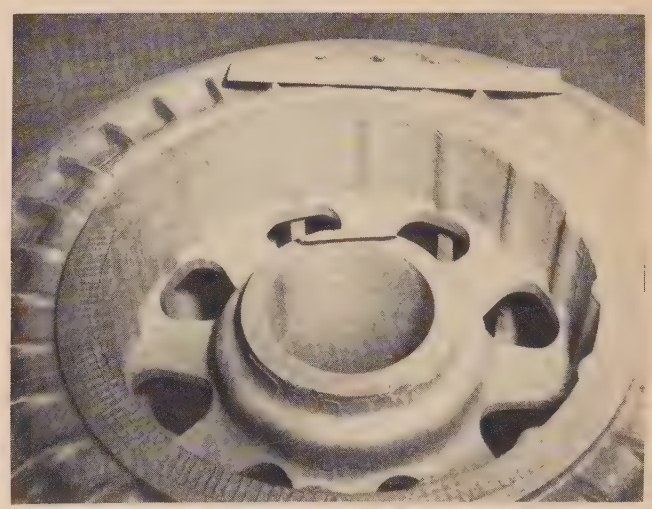


Fig. 17—Aircraft wheel showing defect to be welded

Fig. 18—Aircraft wheel with defect welded and ground down

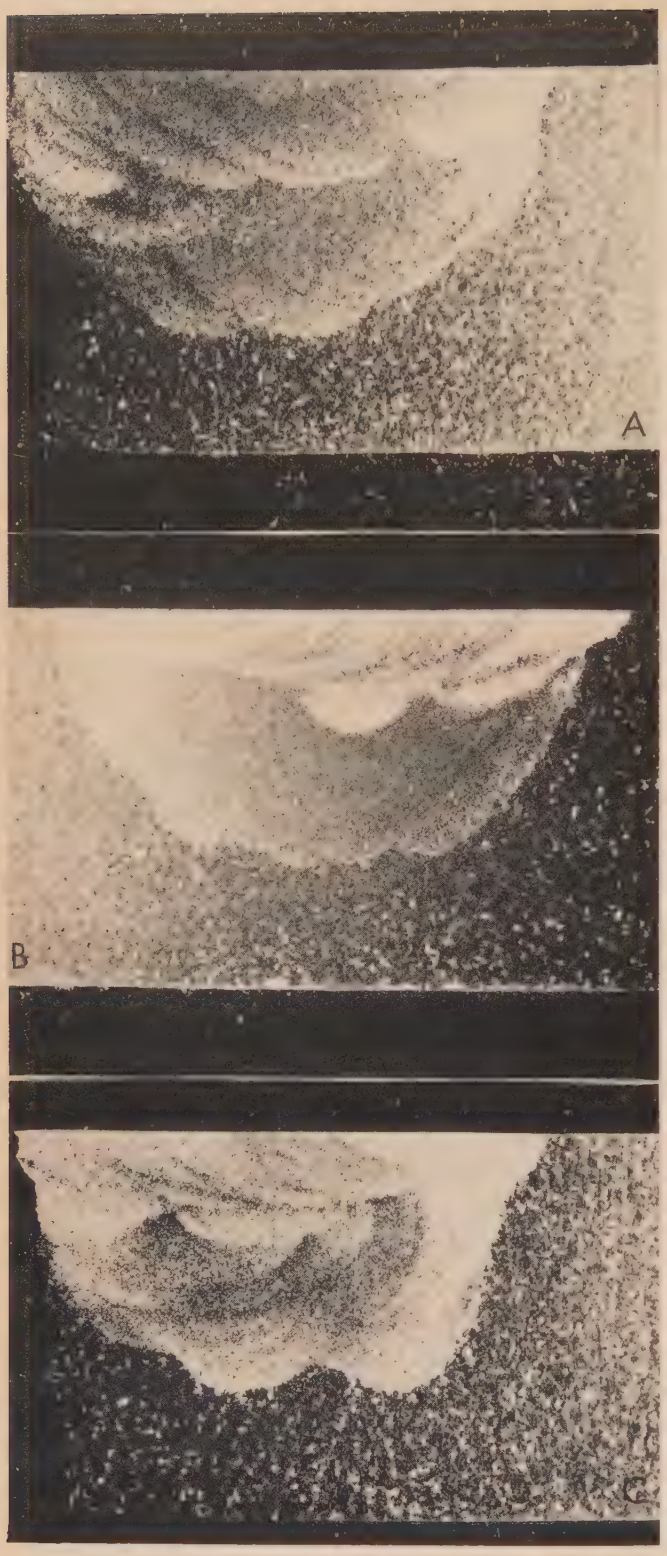


Fig. 19—Macrographs of welded test bars (reduced from macrographs taken at 5X magnification)

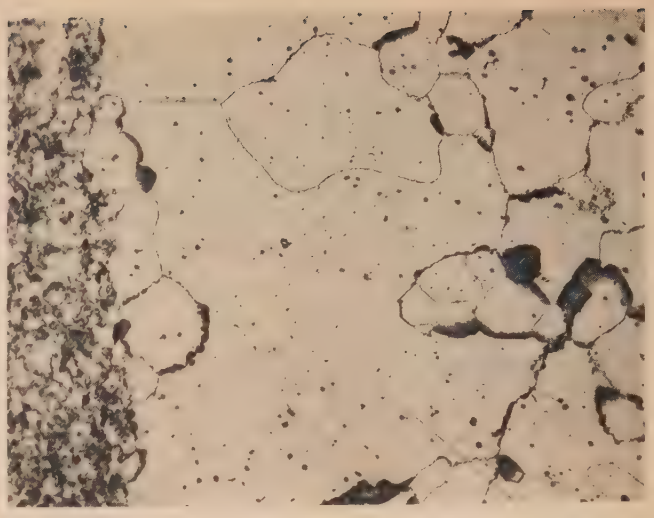


Fig. 20—Photomicrograph of C alloy welded in the as-cast condition

in the weld. Only the minimum heat-treatment time necessary for complete solution should be used for welded Dowmetal C and AZ91C castings, because of the susceptibility of the weld metal in these alloys toward germination or excessive grain growth. The high-temperature solution heat-treatments for C alloy, AZ91C, and AZ81XA should be given with the use of a 1% SO_2 atmosphere in the furnace. A minimum of 0.5% is satisfactory for Mg-Al-Zn alloys.

If complete solution heat-treatment is not desired, welded castings should be stress relieved. An hour soak at 500 F is sufficient to accomplish the stress relief of all the alloys shown.

In the foundry, the only surface treatment given to welded castings is the normal chrome pickle already described under gas welding. Welded castings can be given the same chemical treatments as non-welded castings. The chrome pickle may be applied as a brush-on treatment for cleaned surfaces after welding if facilities are not available for immersion of the part. For maximum surface protection and paint adhesion, a dichromate treatment (Dow No. 7) may be applied. Details on these treatments are given in Table 8.

Very good weld efficiencies are obtainable on arc welded magnesium alloys. When the welding operation is properly performed, the weld metal is dense and fine grained and there is a minimum heating effect on the adjacent metal. The usual weld efficiency obtained in sound metal is around 90%. Tests on the strength of helium arc welded butt joints in cast Dowmetal H alloy and C alloy have shown weld efficiencies of 90-100%, where the efficiency is based on the strength of the original metal in the same condition of heat-treat.

Fig. 19 shows macrographs of test bars of AZ63A-T4 (H alloy) welded with AZ92A (C alloy) rod. These test specimens were preheated to 720 F before welding and given the usual —T4 heat-treatment following the welding. The welding was done on radiographically sound ½-in. thick panels which were

Table 8—Chemical Surface Treatments for Magnesium

Type of Treatment	Composition of Solution	Method of Application	Uses
Sulfuric acid pickling	Concentrated H_2SO_4—4 fluid oz; water to make 1 gal	10-15 sec immersion or a period sufficient to remove 0.002 in. per surface	Used on rough Mg sand casting to remove effects of sand, shot, or grit blasting
Chrome-pickle (MIL-M-3171A, Type I)—Dow 1	Sodium dichromate 1.5 lb; nitric acid, concentrated 1.0-1.5 pt; sodium, potassium, or ammonium acid fluoride—2 oz; water to make—1 gal; temperature—70-90 F	½-2 min immersion plus 5 sec in air, followed by cold—and hot—water rinse and air or forced dry, max 250 F. When brushed on, allow 1 min before rinse	Used as paint base and surface protection. Used on castings for protection during shipment
Dichromate (MIL-M-3171A, Type III)—Dow 7	No. 1 bath: hydrofluroic acid (as 60% HF)—24 fluid oz; water to make 1 gal; temperature—70-90 F No. 2 bath: sodium dichromate—16 to 24 oz; Ca or Mg fluroide—⅓ oz; water to make—1 gal; temperature (boiling)—212 F	5 min immersion in HF acid followed by rinsing in cold running water. Immerse 30 min in boiling dichromate solution followed by cold—and hot—water rinse and air dry	Used for maximum surface protection and paint adhesion. No effect on dimensions of magnesium parts

grooved to half the thickness. Three passes were required to fill the groove. Welding was done from the center to the outer edges to avoid cracking. Test bars were then machined from the panels. The first bar (Fig. 19A) broke in the heat affected zone, the second one (Fig. 19B) broke in the parent metal, and the third bar (Fig. 19C) broke in a defect in the heat affected zone adjacent to the weld. The photos show the marked difference between the relatively fine-grain size of the weld metal and the normal parent metal. Test properties obtained on these bars were as given in Table 9.

Fig. 20 shows a photomicrograph of C alloy welded in the as-cast condition and then solution heat-treated and slightly aged. Relatively large grains can be seen in the metal adjacent to the weld, contrasting with the very fine-grain structure of the weld metal. When welding C alloy and AZ91C alloy, it is desirable to weld in the —T4 or —T6 condition, and then heat-treat for only ½ hr at 770 or 780 F, as previously indicated in Table 7.

If good welding practice is not followed, there are three particular conditions which may be encountered in metal adjacent to the weld which could lower the physical properties at this point: (1) cracking, (2) incipient fusion, (3) grain growth.

Cracks generally originate in the metal adjacent to welds and extend into the weld metal. They are caused by stresses in the metal from heating and cooling of the metal when welded. Cracks follow grain boundaries and will follow weaknesses like incipient fusion and microshrinkage voids if these are present.

Item (2), above, occurs when the low melting constituent in the cast metal melts. It may disappear and leave a hole, and this is sometimes encountered in solution heat-treated castings of C and H alloy. In welding C or H alloy, the metal adjacent to the weld may contain these holes caused by the oxidation of the low-melting Mg-Al-Zn phase. Some of the first weld beads may contain many of these holes caused by the heat from the later passes in multiple-pass welds.

Grain growth may be a factor in lowering the strength of the metal adjacent to the weld. Grain growth is caused by stresses encountered during welding along with fine grain and absence of compound. It also may occur during the solution heat-treatment following welding.

The above three conditions may be avoided by (1) good welding practice to avoid thermal shock (and using alloys of a wide melting range), (2) avoidance of prolonged heating in one spot during welding and multiple passes, where possible, and (3) careful heat-treatment following recommended practice.

In closing, magnesium alloy castings are used extensively for aircraft engines, wheels, and airframe parts. These castings are often salvageable by welding. Good welding practice, as briefly outlined in the foregoing paragraphs, is a requirement for sound welds. Some of the current military specifications applicable to the welding of magnesium are listed in Table 10.

Table 9—Effect of Welding on Alloys

Fig.	Ultimate Tensile Strength, psi	Elongation, %
19A	32,000	7
19B	34,600	6.5
19C	28,000	4.5

Table 10—Specifications Applicable to Magnesium Alloy Welding

Specification	Title
MIL-W-18326 (AER)	Welding of Magnesium Alloys; Process for
MIL-T-5021A (ASG)	Tests; Aircraft Welding Operators' Certification
MIL-F-6943	Flux, Magnesium Alloy Welding
MIL-R-6944 (1)	Rod, Welding, Magnesium Alloy
MIL-H-6857	Heat-Treatment of Magnesium Alloy Castings; Process for
MIL-I-6865a (ASG)	Inspection, Radiographic
MIL-I-6866a (ASG)	Inspection, Fluorescent Penetrant

Full-Scale Field Service Tests

Report of the CFR-DFD Group on Full-Scale Field

IN 1945 an increasing shortage was predicted in certain areas of diesel fuel oil of the grade railroads were accustomed to burn in locomotive diesel engines. Most railroads' specifications at that time limited their supply of fuel to a relatively narrow range of fuel within the No. 2-D classification. It was also predicted that changes would occur in No. 2-D diesel fuel characteristics during the next decade and that increased supplies of fuel for railroads would be available if they could use No. 2-D fuels of higher sulfur content, higher end point, and lower cetane number. Therefore, it was arranged to investigate this possibility by means of field service tests with different types of equipment under various operating conditions.

Comparisons were made with the narrow range of No. 2-D railroad fuels which the railroads had been accustomed to burn and which averaged 0.34% sulfur, 50 cetane number, and 631 F end point.[1] Results indicate that No. 2-D fuels having 1% sulfur, approximately 40 cetane number, and 700 F end point can be used without interfering with operations but that the engine conditions were not as good in respect to deposits, wear, and effect on lubricating oil and filters. It also appeared that fuel properties other than those under test may adversely affect locomotive operation and maintenance. These tests did not include any fuels outside the present ASTM No. 2-D fuel oil classification but were intended to explore the performance of fuels which had characteristics at or near the limits of the classification. These tests seem to indicate the advisability of exploring the possibility of using fuels of still wider ranges in characteristics.

This paper contains a summary of the more significant data contained in the detailed 381-page report of this project.

The significance of these tests can be best appreciated by a review of conditions which caused them to be made and the conflicting views which existed prior to their beginning.

History and Background

The Rapid Growth—Locomotive diesel development was slow, then very rapid. The first American

[1] Average for control fuels used in the field tests.

# of Railroad Diesel Fuels

Service Tests of Railroad Diesel Fuels of CRC

R. W. Seniff, Leader of the Group
F. A. Robbins, Leader of the Analysis Panel

This paper was presented at the SAE Annual Meeting, Detroit, Jan. 11, 1956.

THIS paper describes the background, technology, and recognizes the economies concerning a wider range of fuel quality for the railroad diesel locomotive.

Comparisons are made between normal railroad fuel and a wider range of fuel in eight one-year tests in three types of locomotive equipment.

The test fuels caused increases in engine deposits, wear, and in contamination of filters and lubricating oil. In five of the tests the test fuels did not interfere with locomotive availability. In three tests changes in materials or methods of maintenance were necessary for continuance of operations.

It appears that fuels of the types tested can be used without interfering with locomotive ability if proper precautions are taken.

diesel locomotive was a 300-hp switcher on the Central Railroad of New Jersey in 1924 (Fig. 1), but ten years were to elapse before locomotives powered by diesel engines were perfected to the point where they gained acceptance (Fig. 2). The march of dieselization by American railroads did not begin until 1934 when the first 600-hp streamlined articulated light-weight passenger train called "The Zephyr" went into service on the Burlington Railroad. This type of equipment quickly became popular. It was closely followed by the first separate mainline diesel locomotive which went into revenue service on the Baltimore and Ohio Railroad in 1935 (Fig. 3). Its engines developed 1800 hp. The same year two similar units were coupled together to make a 3600-hp passenger locomotive to handle standard trains on the Santa Fe Railroad.

In 1939 the first mainline freight locomotive appeared on the Santa Fe (Fig. 4). Its engines produced 5400 hp by coupling four 1350-hp units together to comprise one locomotive. The diesel locomotive had come of age. By 1941 diesel locomotives were coming to many railroads in a variety of colors which were

Fig. 1—First railroad diesel locomotive, 1924, 300 hp

Fig. 2—First diesel which gained railroad and public acceptance, 1934, 600 hp

Fig. 3—First separate mainline diesel, 1935, 1800 hp

Fig. 4—First mainline diesel freight locomotive, 1939, 5400 hp

Fig. 5—4000-hp passenger diesel locomotives of five roads, Washington, D. C., 1941

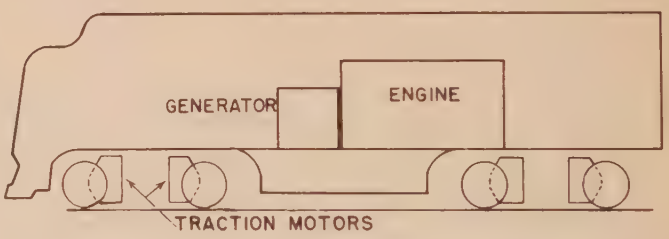


Fig. 6—Typical arrangement of a diesel locomotive unit used on American railroads

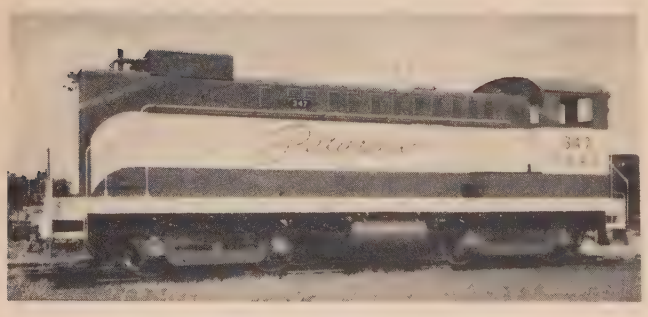


Fig. 7—Modern switcher, transfer-type locomotive used in the CRC tests

Fig. 8—Passenger diesel used in the CRC tests

flashier even than the automobile of today (Fig. 5). In 1949 the Monon and the Gulf, Mobile, and Ohio railroads became the first Class I American railroads to achieve complete dieselization. Today American railroads are about 85% dieselized, and it appears that the trend will continue toward nearly complete dieselization if the economics of diesel locomotives continue to be favorable to it. The typical diesel locomotive on American railroads has electric transmission and the general arrangement is indicated by Fig. 6. American roads now operate 25,325 locomotive units totaling 35.4 million horsepower, plus a significant number of smaller diesels in work equipment.

Technology Also Expanded—During this period of rapid dieselization, many changes took place in the locomotives. Some were visible changes, but the most important and far-reaching ones were invisible. Some idea of the degree of invisible changes may be obtained by comparing the outward appearance of the earlier locomotives (Figs. 1, 2, and 3) with Figs. 7, 8, and 9, which are typical of locomotives which were used for the fuel tests. Some of the changes which are not visible to the eye were improved design and manufacture, increased horsepower and bmep of the engines, and improvement in performance as we dispelled our ignorance concerning design, materials, and methods of construction, maintenance practices, and fuel and lubricant requirements. For instance, the extent of the changes which have occurred in some fuels are as great as those which occurred in the appearance of the locomotives.

There is a complicated series of delicately balanced interlocking relationships between design and materials, methods of construction and maintenance, and the nature of the lubricants and fuels. Changing one of these things may affect another which appears only remotely related to it. This situation can be used to advantage in the light of adequate knowledge, or it can be a "booby trap" in the path of progress. Perception of the significance of these interrelationships began during the early stages of railroad dieselization and influenced the actions of both the users and suppliers of the engines, lubricants, and fuels.

Why Did Railroads Want a High-Grade Fuel?—In the beginning of railroad dieselizaiton it was anticipated that locomotive diesels would successfully burn a wide range of fuel oil. Petroleum fuels are classified according to their physical and chemical properties into a number of grades which indicate their suitability for various purposes. Suitability is determined by experience. The relationships between burner fuels and diesel fuels are indicated in Table 1. The three classes of diesel fuels are selected from the correspondingly numbered classes of burner fuels by taking those fuels having the special properties which make them particularly suitable for diesel use. In other words, any class of diesel fuel is a good burner fuel of the same classification number. On the other hand, not all burner fuels of a given class number make good diesel fuel of the same class number. To differentiate between diesel and burner fuels a "D" is placed after the class number to indicate that it is a burner fuel having properties which make it suitable for diesel use. The various classes of fuel oil are produced by fractional distillation. The heavy portion which is left after distillation is referred to as residual fuel oil. By special processes even the heavy portion can be changed to lighter fuel and distillate leaving only solid petroleum coke. The various classes of fuel oil can also be produced by mixing two or more suitable classes to make the one which is wanted.

Table 1—Relationships Between Burner Fuels and Diesel Fuels

Grade of Fuel Oil ASTM D 396-48T	Grade of Diesel Fuel Oil ASTM D 975-53T
No. 1 — A distillate oil intended for vaporizing pot-type burners and other burners requiring this grade of fuel	No. 1-D — A volatile distillate fuel oil for engines in service requiring frequent speed and load changes
No. 2 — A distillate oil for general-purpose domestic heating for use in burners not requiring No. 1 fuel oil	No. 2-D — A distillate fuel oil of low volatility for engines in industrial and heavy mobile service
No. 4 — An oil for burner installations not equipped with preheating facilities	No. 4-D — A fuel oil for low- and medium-speed engines
No. 5 — A residual-type oil for burner installations equipped with preheating facilities	
No. 6 — An oil for use in burners equipped with preheaters permitting a high-viscosity fuel	

Because of its abundant supply and ready availability throughout the country No. 2 fuel was selected for diesel locomotives in the mid 1930's. But soon after dieselization started to expand in 1935, and while using No. 2 fuels, certain conditions developed in the engines which were suspected of being associated with the properties of the fuels. Analysis of No. 2 fuel is shown in Table 2. Some of these conditions were stoppage of fuel filters and strainers as indicated by Fig. 10, corrosion of parts of the fuel transfer system and injectors as indicated by Figs. 11 and 12, and valve and exhaust-passage deposits as indicated by Figs. 13 and 14. Also suspected as being associated with fuel quality were smoke, excessive products of combustion in the crankcase oil, bearing corrosion, ring and liner wear, and stuck and broken rings.

Of course, not all of these things were caused by fuel; but the best way to find out appeared to be by elimination, as far as consistent, of the fuel variables and allowing the engines to say whether they liked the fuel. Railroad specifications were lax in those days; and railroads, having not yet recognized the importance of these variables, paid little or no attention to the quality control of the fuels they purchased. Since fuel quality varied widely, it appeared logical to expect that a better and more uniform grade of fuel would be beneficial. Accordingly, most railroads

Fig. 9—Freight locomotive of the type used in the CRC tests

Fig. 10—Plugging of fuel-oil strainer adjacent to injector

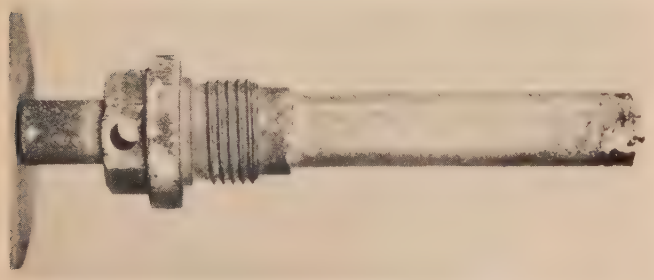


Fig. 11—Corroded strainer

tightened up on their fuel specifications by demanding the "cream" of the No. 2 fuel oil production in the form of a very clean, stable, straight-run No. 2-D fuel having a maximum of 0.5% sulfur, a comparatively low end point, and a minimum cetane number of 50 (see Table 2). Actually the fuels supplied were in general agreement within those limits, to the extent that most railroad experience up to 1949 was with transparently clear, dry, perfectly stable No. 2-D fuel having an average sulfur content of possibly 0.25%, 630 F end point, and higher than 50 cetane number. Very little, if any, trouble occurred in the engines which could be blamed on such fuel. This left the engine builders free to develop good engines and good maintenance practices unhampered by complicated fuel quality factors. This in turn resulted in a high level of engine performance, reliability, and very favorable economics of maintenance and operation.

Supply versus Demand—As previously mentioned, once equipment was perfected, the rate of dieselization by railroads was rapid. Fig. 15 indicates the installed locomotive diesel horsepower by years. Of course, the consumption of the special grade of fuel oil railroads were asking for also increased as indicated by Fig. 16. Consumption of distillate burner fuel from which the railroad No. 2 diesel fuel is selected also increased, as indicated by Fig. 16, principally because of its increased use in home oil burners for household heating. In addition to this, exhaustion of old oil fields and the opening of new ones introduced in some cases different kinds of crudes which yielded a product of different characteristics which failed to meet traditional railroad specifications for the select portion of No. 2-D fuels. During the war and especially thereafter there was a rapid change in refining methods from straight run to catalytic cracking in order to produce more gasoline. This not only increased the competition of gasoline for the No. 2 fuel part of the crude but also

Fig. 12—Corroded fuel line from locomotive

Fig. 13—Valve deposits caused by fuel oil which resulted in stuck valves

Fig. 14—Exhaust-passage deposits

produced a No. 2-D fuel oil of characteristics different from those with which railroads were familiar. Thus the traditional railroad grade of diesel fuel oil, which was a narrow range of fuel selected from the No. 2-D classification and which in turn was selected from the No. 2 classification, became short in supply in some areas.

Table 2—Properties and Their Limits Used in Stating Requirements for No. 2 Fuel Oils

	Fuel Classification and Specification		
	ASTM Classification (1955)		A Railroad Diesel
Tests (ASTM)	No. 2 Burner	No. 2 Diesel	Specification (1946)
Min Flash Point, F	100 or legal	125	150
Pour Point, F	0-20	10 Below Ambient	0
Max W and S, volume %	0.10	0.10	0.05
Max Carbon Residue, %	0.35	0.35	0.35
Max Ash, weight %	—	0.02	—
Max 90% End Point, F	675	675	640
Viscosity, SUS at 100 F, min	—	32	—
max	40	45	32/45
Max Sulfur, weight %	By agreement	1.0	0.5
Max Corrosion	Pass	No. 3	No. 3
Min Cetane Number	—	40	50

Recognition of the Problem—In 1945 the Western Petroleum Refiners Association called attention to the changing conditions and suggested that the future supply of fuel for railroads could be increased if they

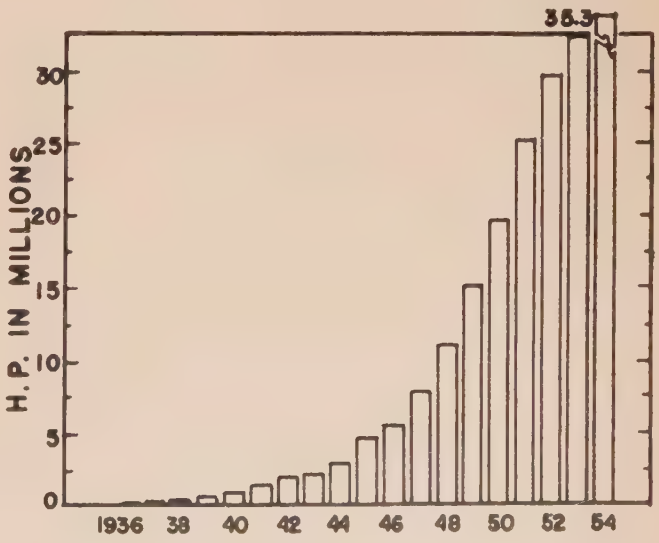


Fig. 15—Installed railroad diesel locomotive horsepower

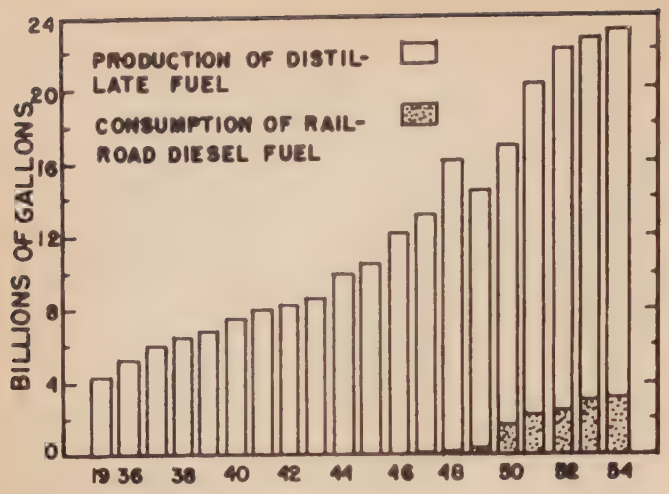


Fig. 16—Production of distillate and consumption of railroad diesel fuel

Fig. 17—Data from one field test (left) condensed by the test team into the report at the right

could find it consistent to broaden the range of fuels which they could successfully use and thereby relax the restrictive railroad specification. Joined by the American Petroleum Institute they requested the Coordinating Research Council, Inc., which is jointly sustained by American Petroleum Institute and the Society of Automotive Engineers, to undertake a study of fuels for railroad locomotive diesel engines.

Preliminary Survey

The program got under way in 1946 when the CRC established a group to survey current and future diesel fuel properties and to predict the possible trends these might take in the next ten years. A survey was also made covering the locomotive manufacturers' and railroads' specifications for diesel fuels.

The survey covering diesel fuel characteristics indicated that future diesel fuels in greatest supply would have higher sulfur content, lower cetane number, and higher final boiling point. Specifically, they predicted the levels which these properties might reach were:

$1.0 \pm 0.2\%$ sulfur content
40 ± 2 cetane number
$700 +$ F end point

After studying these surveys, the Diesel Fuels Division of the Coordinating Fuel and Equipment Research Committee of the CRC recommended that a program be undertaken to investigate, by means of field service tests, the effect of variations in fuel properties on railroad locomotive operations. The CRC then invited a group of representatives from the petroleum, railroad engine building, and railroad companies to discuss the possibilities of such tests.

Resistance to Change—The views of the users at that time and also those of a smaller group, which met later to discuss the details of the proposed tests, are particularly significant; and it is necessary to review them and to understand the reasons for them in order to grasp today the significance of the CRC Full-Scale Field Service Tests of Railroad Diesel Fuels.

Remember that was nine years ago, and railroads were enjoying comparative freedom from previous troubles which were caused by fuel quality, and they still vividly remembered some of the early distressing experiences with poor-quality fuels. To explore a wider range of fuel quality looked like a backward step and an unwarranted invitation to trouble. Nor were they alone in their doubts. When the small group consisting of a few well-known authorities on fuel oil and engines and one or two railroad men met to discuss details for the first service test, mention was made of the possibility of combining the three variables, 1.0% sulfur, 40 cetane, and 700 F end point in one test fuel to expedite the work. The question was asked, what do you think it would do to the engines? Not a person would hazard a guess. When we went to railroads (and we contacted many of them) to try to arrange for tests, we had a very difficult selling job. So difficult in fact that two years were to elapse before agreement could be obtained for the first test. And it was obtained only after the engine builder agreed to assume the responsibility for possible damage to the engines. Such was the state of

[2] CRC-292: "Report on Full-Scale Field Service Tests of Railroad Diesel Fuels." Available from SAE Special Publications Department. Price: $5 to members, $10 to nonmembers.

[3] The term "power assemblies" refers to cylinder liners, pistons, rings, piston pins, rods, bearings, and so forth, which make up the parts assembled to produce one cylinder assembly.

mind up to 1950 when the first tests were completed.

In May, 1949, when agreement was obtained for the first test the Diesel Fuels Division established a separate group designated as the Group on Full-Scale Field Service Tests of Railroad Diesel Fuels of the Diesel Fuels Division of the Coordinating Fuel and Equipment Research Committee to carry out the previously defined objective of the program.

Method of Testing

Uniform procedures for testing and reporting were formulated by the Group on Full-Scale Field Service Tests of Railroad Diesel Fuels. Each test was made by a test team composed of a technical man from the railroad, the oil company which supplied the test fuel, and the engine builder. This team made periodic joint inspections of the engines during the test, and members rode engines at regular intervals in regular service to evaluate service performance. All members of the test team joined in inspection, rating and measurement of test parts, evaluation of the data, and preparation of the summary report of their test. The paper work involved in one test is shown at the left in Fig. 17, and the summary report into which it was condensed is at the right.

The Analysis Panel of the CFR-DFD Group on Full-Scale Field Service Tests of Railroad Diesel Fuels at the end of the field portion of the program then reviewed all test-team data shown at the left in Fig. 18 and condensed them into the 381-page final report [2] shown at the right.

Sufficient duplicate tests were made to verify the results. In each test two or more similar locomotive units were used for approximately one year. The units were equally divided between test fuel, prepared as close as practical to the levels predicted in the fuel survey, and control fuel, which was the fuel normally used by the railroad. In each test, the test and control fuel units were assigned to similar service and run coupled together in most instances. Special effort was exercised to eliminate statistical bias and to keep the test fuel the only significant variable in the test. Special precautions were taken to prevent mixing of test and control fuels. Since the locomotives were not new at the beginning of the tests, at least two precisely measured power assemblies [3] were installed in each engine except in one case, where an additional unit was observed on both test and control fuel for overall effects only. These measured assemblies were removed at the conclusion of the test and were measured and inspected by the test team to determine significant differences.

Records were kept during the tests noting changes of all engine parts and accessories that may have affected test results. In addition, periodic inspections were made to observe conditions that would show fuel effects and to assure control on test operation.

Periodic samples of both the control and the test

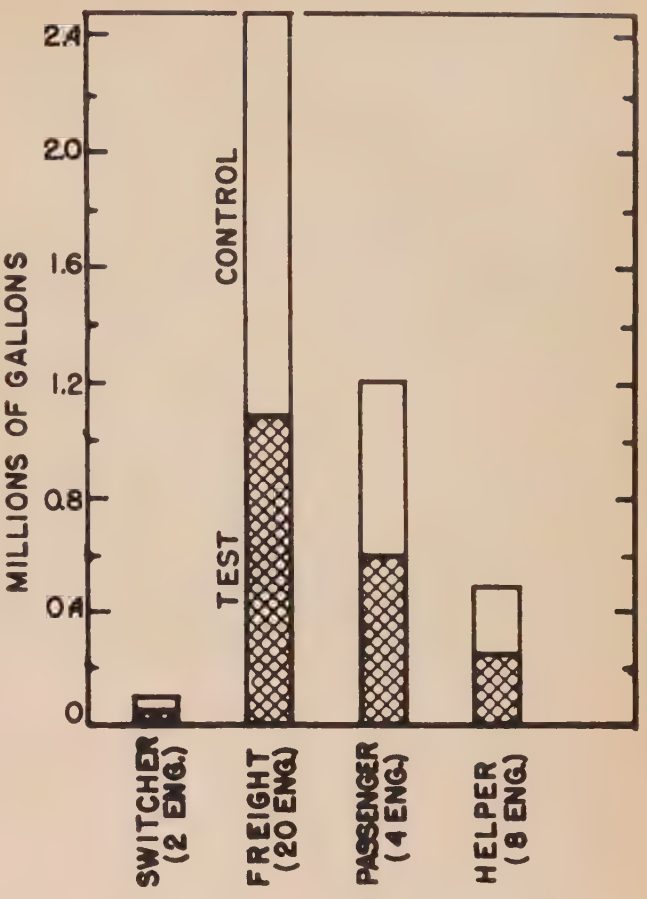


Fig. 19—Diesel fuel consumed during test

Fig. 18—Data from test teams reviewed by the analysis panel and condensed into 381-page report at the right

Fig. 20—Railroad test locations and terrain

fuel were analyzed. These data were then combined to give an average representing the fuel properties. The full-scale tests described in this report include data on average fuel consumption based on various parameters, for example, unit month and gross ton-miles. These data have been included to give an indication of gross engine utilization, but they are not to be construed to reflect engine efficiency as affected by fuel type. Engine efficiency data can only be developed under conditions of constant load and speed. Such tests were beyond the scope of this project, which was to determine the effect of certain fuel types on operation and maintenance of locomotive diesel engines under normal railroad operating conditions. Periodic samples were taken of used lubricating oils. These were analyzed by predetermined test methods to establish the effect on oil properties.

Table 3—Description of Diesel Engines Used in the Tests

Type	Stroke Cycle	Cylinders No.	Cylinders Bore	Cylinders Stroke	Rpm	Compression Ratio	Bhp
Switcher[a]	4	6	12.75	15.50	625	11.42	1125
Freight[a]	4	12	9.00	10.50	1000	13	1600
Passenger	2	12	8.50	10.00	800	16	1100
Freight	2	16	8.50	10.00	800	16	1650
Freight[a]	4	12	9.00	10.50	1000	13	1500
Pusher[a]	4	8	12.75	15.50	625	13.45	1400
Freight	2	16	8.50	10.00	800	16	1440
Freight	2	16	8.50	10.00	800	16	1650

[a]Supercharged.

Description of Tests

The first of the tests was started in May, 1949, and the last one was completed in June, 1955. The eight tests covered by the program involved a total of 34 diesel engines of three makes and were equivalent to approximately 38 engine test years. Seventeen of these engines were operated on test fuels and seventeen on control fuels. The test program covered approximately 2½ million unit miles of combined high-speed passenger and average-duty to heavy-duty freight operation and 8000 hr of switching service. Approximately 2½ million gal of test fuel and an equivalent amount of control fuel were consumed during the tests. Fig. 19 showing consumption of fuel in switcher, passenger, and freight locomotive tests gives a general idea of the relative amount of testing allotted to each type of service.

Engines—Table 3 indicates the nature of engines tested. The engines are typical of those used in the locomotives of American railroads.

Operating Conditions—Table 4 gives a description of the diesel locomotives employed in the study, type of service, and conditions under which the equipment was operated. Fig. 20 indicates the locations and terrain where the test locomotives were operated.

Fuel Properties—All test and control fuels fall within the ASTM D 975 classification for No. 2-D

Table 4—Test Locomotives and Operating Data

Railroad	1	2	3	4	5	6	7	8
Locomotive Builders Code	JA	KA	LA	LB	KB	JB	LC	LD
No. of Locomotives Involved	2	2	1	2	2	2	2	2
No. of Units Involved	2	4	2	4	2	4	4	6
Type of Service	Average Switching	Moderate freight	Moderate high-speed passenger	Heavy freight sanding under power	Average-duty freight sanding under power	Road-freight pusher	Medium-duty freight	Heavy freight sanding under power
Terrain	Flat	Gently rolling	Flat	Mountainous	Gently rolling	Mountainous	Hilly	Mountainous
Locomotive Horsepower per Unit	1000	1600	2000[a]	1500	1500	1250	1350	1500
No. of Engines, Test	1	2	2	2	1	4	2	3
Control	1	2	2	2	1	4	2	3
Total	2	4	4	4	2	8	4	6
No. of Measured Power Assemblies on Test, Test	3	4	4	4	2	8	4	4
Control	3	4	4	4	2	8	4	4
Total	6	8	8	8	4	16	8	8
Duration of Test, months	12	11	12.1	6.3	17.0	11.6	12.8	13.5
Average Total Test Miles per Unit	—	64,000	204,000	45,000	71,500	43,000	149,000	104,000
Total Gal Test Fuel Used	55,000	254,000	361,800	216,000	128,500	310,000	596,000	601,000
Average Miles per Unit Month	—	5,450	17,000	7,100	4,200	3,700	11,700	7,700
Average Gal per Unit Month	4,600	10,900	24,800	17,100	8,150	12,400	23,400	14,800
Hours of Idle per Day	9.5	9.0	11	16	2[b]	12	9	17.5
Hours of Load per Day	14.5	15.0	13	8	7[b]	12	15	6.5
Ambient Temperature, max, F	100	98	100	90	100	99	100	100
min, F	10	−20	−10	−5	20	10	−38	−5

[a]Two engines per unit.
[b]Locomotive shut down 15 hr per day.

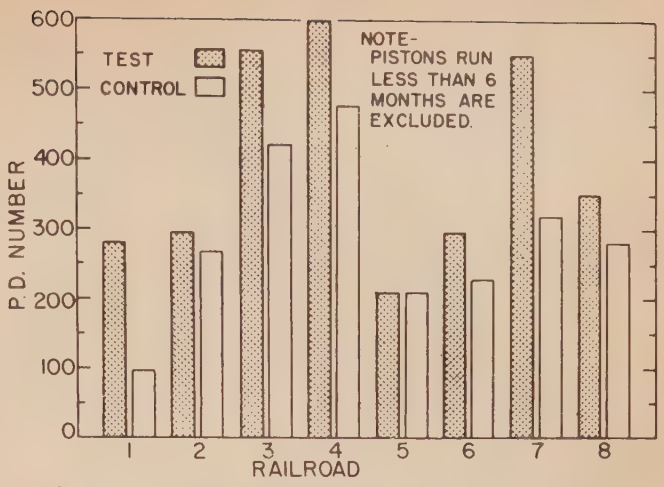


Fig. 21—Average piston-skirt deposits (P. D. number)

fuel. The control fuels were those regularly used by the railroad in the location where the tests were made. The significant differences in the properties of the test and control fuels which were used in each test are shown in Table 5.

Lubricating Oil—The lubricating oils used in all tests were those used by the railroad in the location where the test was made. In one case oil reclaimed from mixed drainage was used exclusively. All the oils were equivalent to or higher than Army 2-104B quality and were typical railroad diesel oils.

U.S. Bureau of Mines Analysis of Fuels—Supplemental analytical data were obtained on all test and control fuels by the U.S. Bureau of Mines, Petroleum Experimental Station at Bartlesville, Okla. The data included the distribution of sulfur and type of hydrocarbons throughout the boiling range of each fuel. The behavior of the fuels was also studied by them in the "constant-volume bomb," and the Bureau has been kind enough to offer that group of data as additional information.

Small-Engine Tests—Two independent laboratories offered to make tests of certain of the fuels in 2- and 4-stroke-cycle small-scale engines. One laboratory used the same lubricating oils that were used in the railroad engines. The other used their standard straight mineral test oil. These data were presented to the Group on Full-Scale Field Service Tests of Railroad Diesel Fuels and are included in CRC Report No. 292 as information.

Future Work—The field test phase of the program is now completed, but studies of the relationship

Table 5—Significant Variables in the No. 2-D Fuels Used in the Tests

	Sulfur Content, %	Cetane No.	End Point, F
Railroad 1			
Test	1.00	42	695
Control[a]	0.155	47[b]	632
Railroad 2			
Test	0.95	48	686
Control[a]	0.45	49	640
Railroad 3			
Test	0.98	45	625
Control[a]	0.31	53	616
Railroad 4			
Test	1.15	43	649
Control[a]	0.30	51	635
Railroad 5			
Test	0.90	39	644
Control[a]	0.51	45	611
Railroad 6			
Test	1.0	42	647
Control[a]	0.42	49	642
Railroad 7			
Test	1.10	44	687
Control[a]	0.21	54	634
Railroad 8			
Test	1.00	42	697
Control[a]	0.29	49	636
Test Fuel Target	1.00	40	700

[a]Fuel normally used by railroad.
[b]Calculated cetane number.

Table 6—Test-Fuel Units Showing Greater Deposits or Wear than Control-Fuel Units

Piston-Skirt Lacquering	16
Cylinder-Liner Bore Wear	12
Piston-Ring Wear	11
Cylinder-Liner Bore Deposits	9
Cylinder-Liner Port Deposits	9
Wristpin-Bushing Deposits	9
Rod Eye Wear	9
Valve Deposits	9
Injector Deposits	8
Ring-Groove Wear	8
Ring-Groove Deposits	5
Crankcase Deposits	5

which may exist between various data continue. A special group of selected experts in CRC has recently been assigned the task of examining the Bureau of Mines, the small-scale engine, and the railroad engine data for correlation.

General Results

The following general results have been established:

1. All eight full-scale railroad field tests involving 17 test-fuel and 17 control-fuel engines produced data which could be interpreted with respect to the purpose of this project.

2. The use of test fuels did not affect locomotive performance or suitability for its assigned purpose. However, in order to keep test locomotives operating in certain cases, corrective measures had to be taken to overcome filter clogging caused by the high cloud and pour points of the fuel or injector sticking caused by products of fuel instability which were deposited on injector plungers and barrels.

3. The test fuel caused greater wear or deposits in

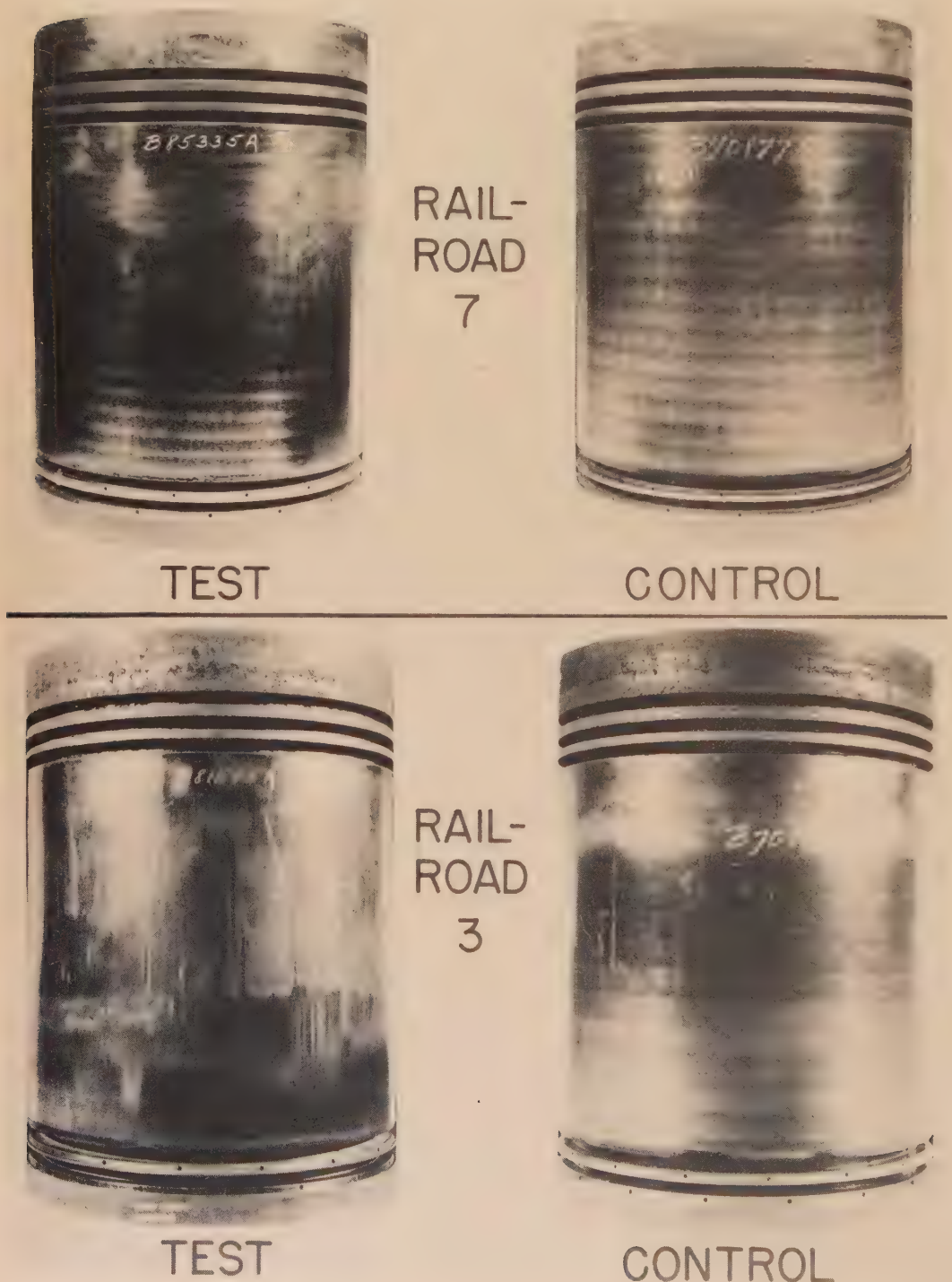
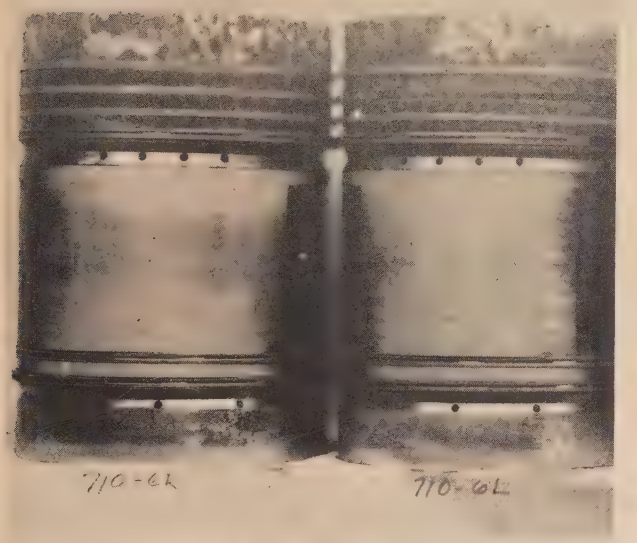


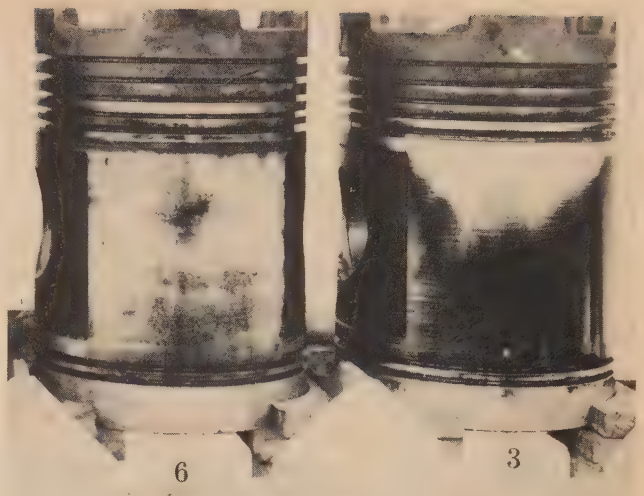
Fig. 22—Piston deposit conditions

TEST



CONTROL

Fig. 23—Piston deposit conditions, railroad 5



TEST

CONTROL

Fig. 24—Piston deposit conditions, railroad 6

seven out of the eight tests. Table 6 indicates the number of locomotive units out of the 17 units on test fuel which had greater deposits or wear than the companion control-fuel units.

It appears significant that no clear-cut beneficial effect of test fuel was noted in any case except a minor instance in one test where the fuel strainers and filters were cleaner with the test fuel than the control fuel. Reexamination of the fuel-oil analysis disclosed that this test fuel had a lower gum content than the control fuel used in this test.

4. The small-engine investigations conducted on several of the test fuels and control-type fuels ex-

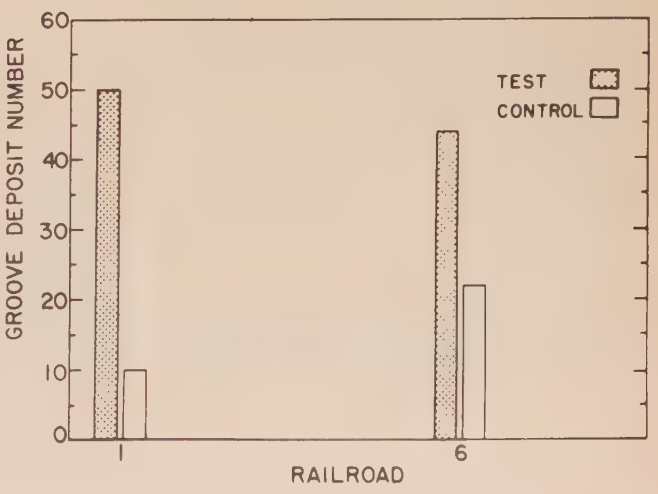


Fig. 25—Average top-ring-groove deposits

hibited trends similar to those observed in the full-scale service tests with respect to deposits and wear.

Detailed Effects of Test Fuels

Deposits—Piston-skirt deposits were greater in all but one of the tests as shown by Fig. 21. Photographs, Figs. 22 through 24, illustrate actual deposit conditions.

Ring-groove deposits were greater in only two tests as indicated by Fig. 25. No significant differences in ring-groove deposits were observed in the other six tests.

Cylinder-bore deposits were greater in the ring travel area in all tests on 2-stroke-cycle engines. No significant differences were observed in this area in the 4-stroke-cycle engines. Fig. 26 illustrates the situation.

Connecting-rod-eye bushing (wristpin bushing) deposits were greater in all 2-stroke-cycle engines using test fuel (see Fig. 27).

Intake-port deposits were greater in the 2-stroke-cycle engines (Fig. 28).

Intake-system deposits of 4-stroke-cycle engines were greater in six out of the eight engines. Fig. 29

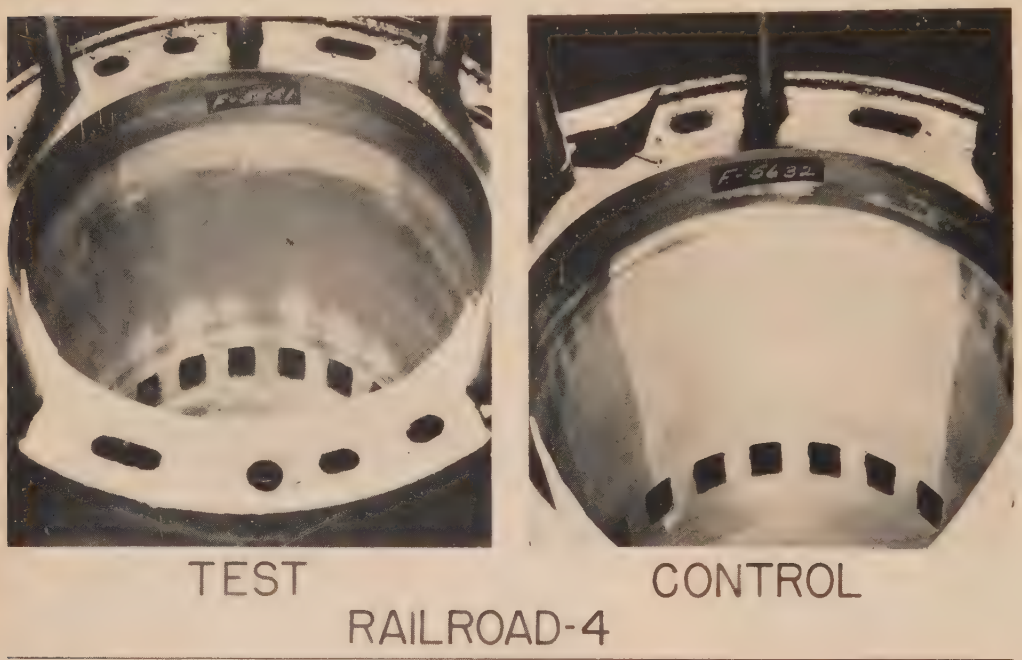
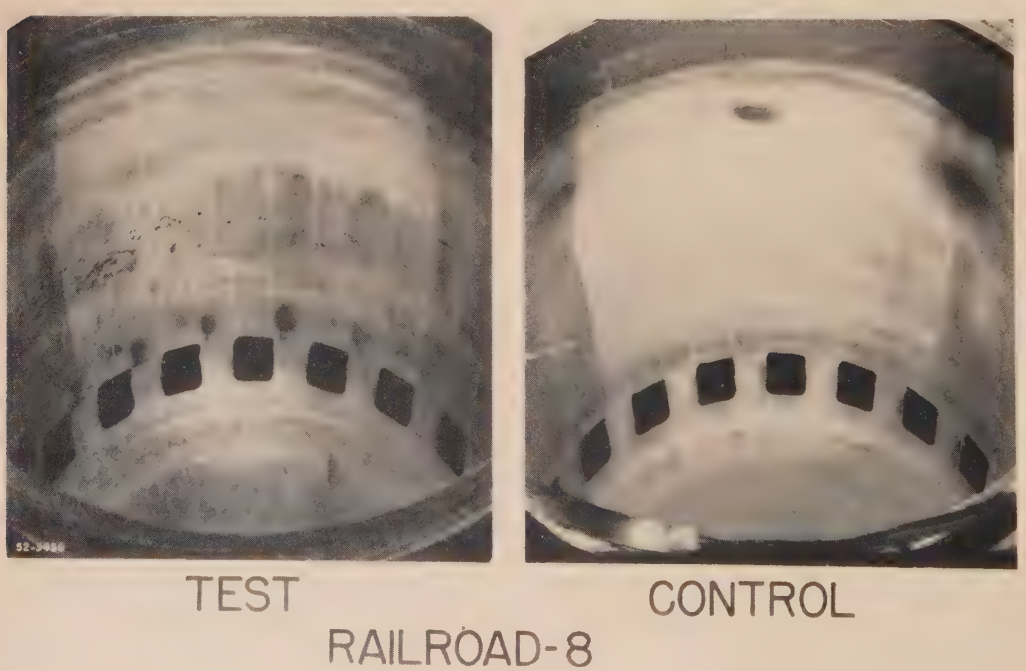


Fig. 26—Cylinder-liner-bore deposits



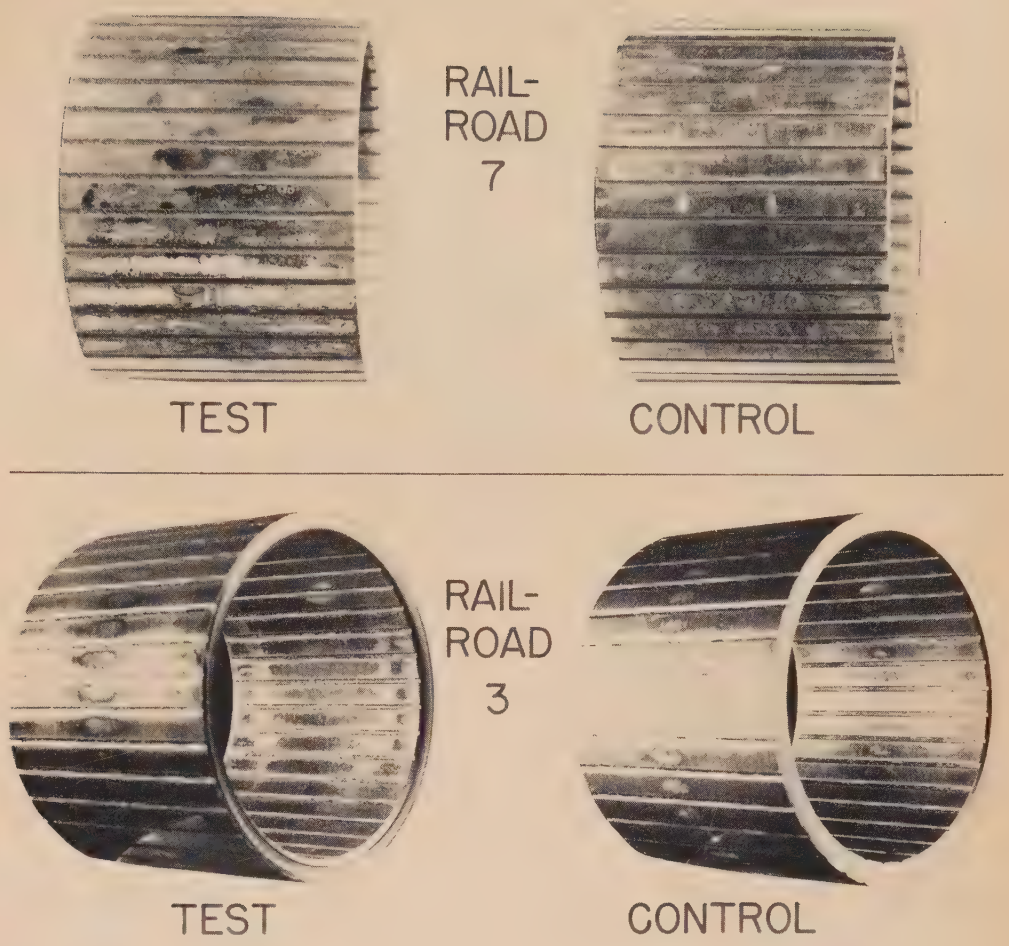


Fig. 27—Wristpin bushing deposits

indicates the difference in the intake valves with control and test fuels on railroad 1. The difference was less on railroad 6. The air intake passages of the cylinder heads run with test fuel on railroads 1 and 5 were coated with from $\frac{1}{16}$- to $\frac{3}{16}$-in. thicker deposits than those run with control fuel. No difference was visible in that area on railroads 2 and 6.

Exhaust-valve deposits were slightly greater only on railroads 1, 4, and 7.

Exhaust-chamber deposits were not significantly different in any test.

Crankcase and top-deck deposits were greater on railroads 4 and 8.

Injection-pump deposits on railroads 3, 4, and 8 are illustrated by Fig. 30. On railroad 4 injector sticking became severe after 2½ months and interfered with railroad operations to the extent that the test had to be temporarily discontinued. This sticking was caused by a dark lacquer-like deposit at the top of the injector plungers where lubricating oil and fuel oil came into contact with one another. These deposits caused the plunger to bind in the injector bushing and remain in the down position. The test was resumed after taking corrective measures.

Nozzle-valve deposits were greater on railroad 1 as illustrated by Fig. 31. Sticking of the valves necessitated changing them after six months' operation.

Fuel-filter deposits were significantly increased on railroads 1 and 8 which used the same test fuel in

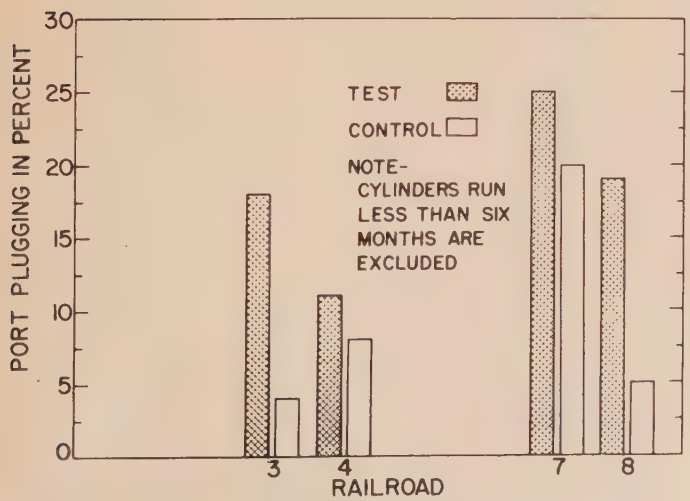


Fig. 28—Average intake-port plugging in cylinder liners

Fig. 29—Intake- and exhaust-valve deposits, railroad 1

locomotive equipment of different manufacture. Fig. 32 illustrates the maximum differences which were observed. Fuel-filter deposits of wax during winter months from test fuels of higher than normal railroad fuel cloud and pour points stopped test locomotives on railroads 7 and 8. On railroad 8 it was necessary to install heating facilities for unloading and storing the test fuel and fuel heaters on the locomotives in order to continue operations.

Lubricating-oil-filter deposit differences were not determined in the earlier tests. However, in test on railroad 8 they were included in the inspection reports. In that test the filter deposits were heavier on the filters from the test-fuel engines as indicated by Fig. 33, and the threads of the cotton waste in these filters were noticeably weakened as if they had under-

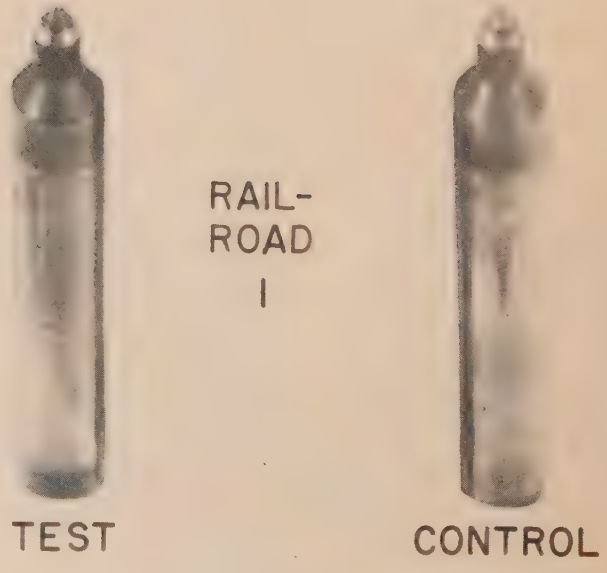


Fig. 31—Nozzle-valve deposits

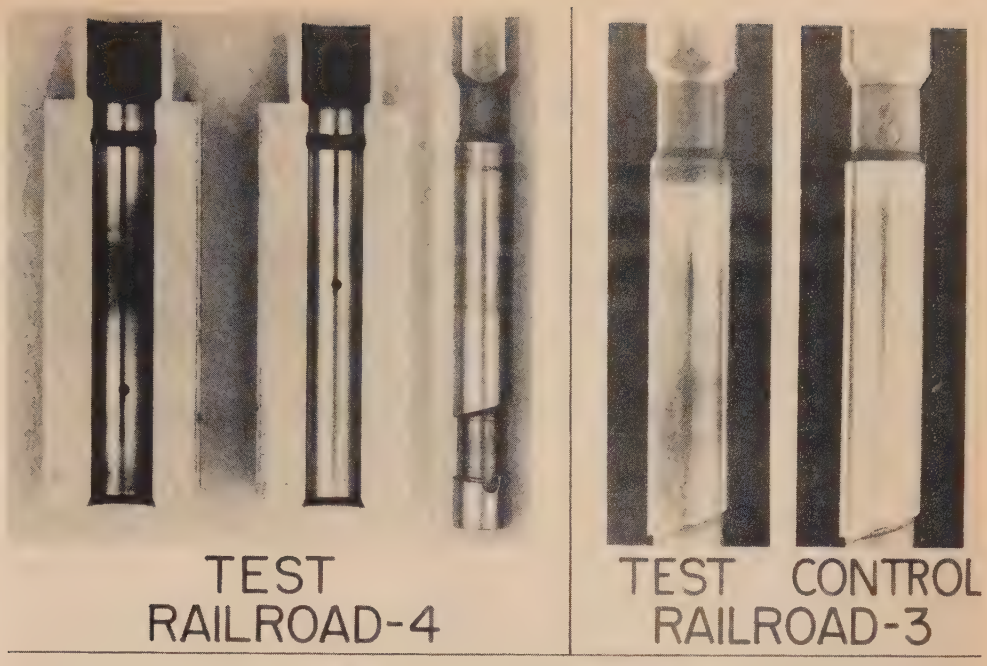
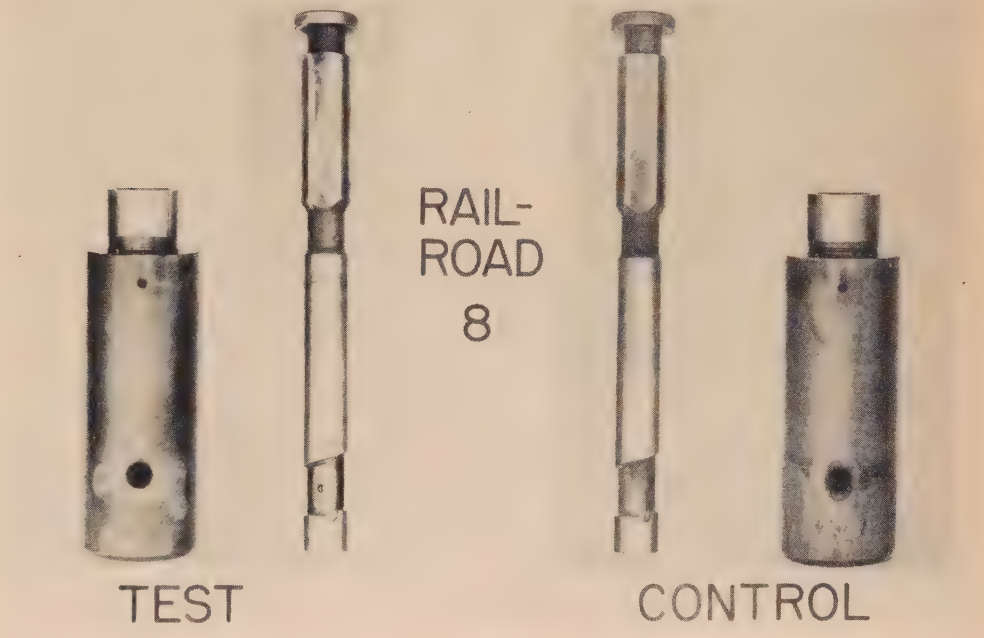


Fig. 30—Injector plunger and bushing deposit conditions



gone some form of chemical attack.

Effect on Lubricating Oil—In seven out of the eight tests there was no outstanding clear-cut difference between the oil in the test and control engines which could be detected by ordinary laboratory inspection methods. In some cases effect of water and fuel-oil dilution in the crankcase oil interfered with reaching clearcut conclusions. However, it was reported by each test team that the oil in the control engines remained cleaner than in the test engines. Blotter spot, dark field microscope, and electron microscope inspections were made in some tests. There appeared to be some correlation between these

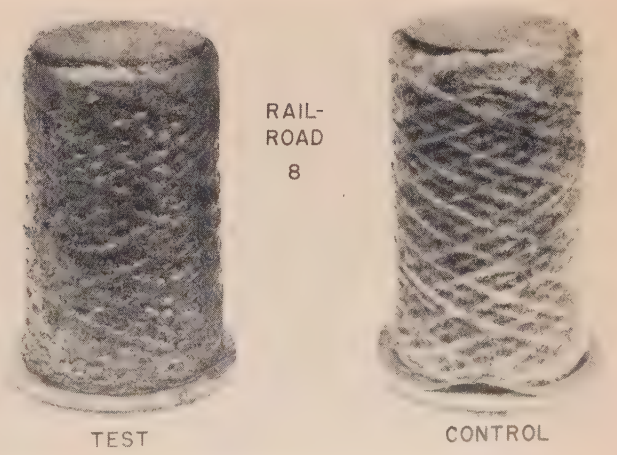


Fig. 32—Fuel-oil-filter deposits

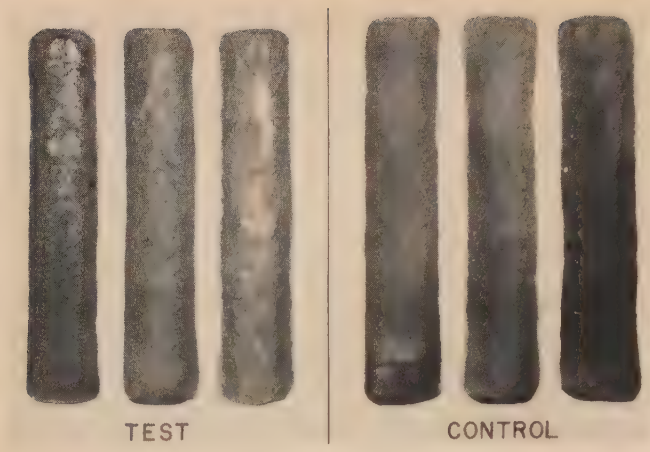


Fig. 33—Lubricating-oil-filter deposits

tests concerning degree of dispersion of particulates in the lubricating oil.

On railroad 8 there was a consistent and significant difference between the oil in test and control units, both in appearance and laboratory analysis and in the effect on lubricating-oil filters. The lubricating oil in the test-fuel engines in this test consistently developed the following differences when compared with the oil in the control engines:

Total Acid Number	Slight increase
Total Base Number	Slight decrease
pH	Moderate decrease
Viscosity	Slight increase
Carbon Residue	Slight increase
Ash	Slight decrease
Pentane Insolubles	Large increase
Benzene Insolubles	Large increase

The reason for this marked difference between test 8 and the other seven tests is not clear. However, conditions which might have some bearing on it are: (1) the test fuel was nearer the test target quality of 1.0% sulfur, 40 cetane number, and 700 F end point, and (2) the locomotives were in heavy-duty freight service.

Smoke densities were not measured, but frequent visual observations at terminals and during over-the-road tests under all operating conditions failed to indicate any significant differences at the exhaust stacks except on railroad 5 where visible smoke density was slightly greater in the test-fuel engine at full load.

Wear—increased wear occurred in 6 of the 8 tests on 12 out of the 17 engines.

Cylinder liners showed the increased wear in 12 out of 17 engines as indicated by Fig. 34. Chrome-plated liners (railroads 1, 2, and 5) had substantially less wear than iron liners.

Ring-wear differences are shown by Fig. 35. Top-ring wear only is shown because it is greatest and therefore the controlling factor.

Ring-groove wear increased in eight test units as indicated by Fig. 36.

Pistons had no other significant wear.

Connecting-rod-eye wear increased in all 2-stroke-cycle engines using test fuel as indicated by Fig. 37. This was accompanied by increased deposits on the floating-rod-eye bushings as previously indicated by Fig. 27.

General Conclusions

The following conclusions can be drawn on the basis of the tests:

1. The overall approach to full-scale testing employed in this project appears to be a valid means of determining the effect of variation in fuel properties on railroad diesel engines.

2. Considering only the sulfur content, cetane

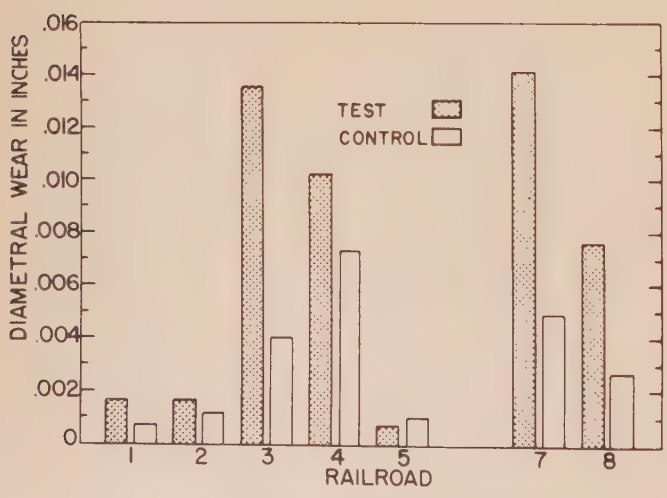


Fig. 34—Average annual top-cylinder wear

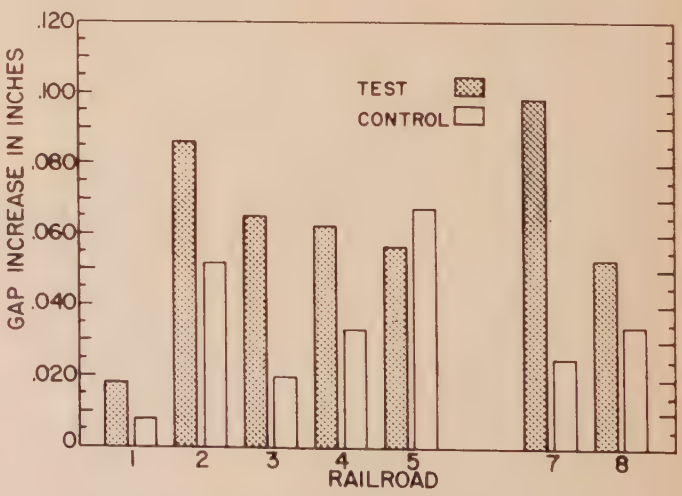


Fig. 35—Average annual top-ring-gap increase

number, and end point as physical properties of the test fuels, it was possible to operate the locomotive diesel engines under study without sacrificing locomotive availability.

3. In considering application of fuels of the type under test in this program, it should be noted that fuel properties other than sulfur content, cetane number, and end point may adversely affect locomotive operation or maintenance. Instability and high cloud and pour points are examples of such properties which were encountered in these tests.

4. Due to wear or deposits, increased engine maintenance such as more frequent change-out of power assemblies may result when using fuels of the type under test. This trend depends on the nature of the test fuel used, the nature of the locomotive operation, and maintenance practices, and it was observed in 16 out of the 17 test-fuel engines.

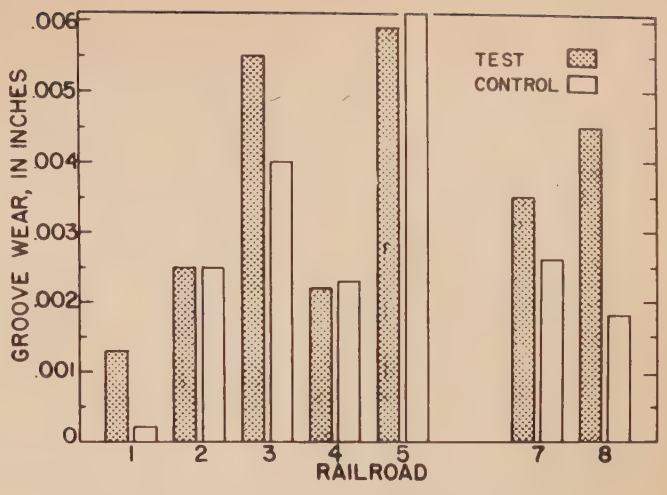


Fig. 36—Average annual top-ring-groove wear

Comments

The practical application of the results of the first tests began in 1950 soon after their completion when some railroads relaxed their fuel oil specifications. This immediately eased the fuel procurement situation for them in certain areas by permitting the use of local fuels and eliminating the necessity of hauling higher-grade fuels into such areas from greater distances. (Attention is called to the scope of the tests which limited them to the large locomotive engines. Persons interested in using fuels of the test types in small diesel engines such as those employed in railroad work equipment should make their own determinations of the effect of such fuels on such engines.) Other practical applications which were started at an early date were experiments to improve the tolerance of engines for a wider range of fuel quality, experiments with lubricating oils to give added protection to the engines under the influence of a wider range of fuels, and increased experimentation with fuel-oil additives. Railroads became increasingly interested in cooperative fuel testing. They also began to undertake tests of their own involving a still wider range of fuel qualities than those under test by CRC.

Many things were learned during the CRC test program which were not within its predefined scope and objective but which in themselves may be as important as the achievement of the test objectives. For instance, these tests have served to indicate that there are still other factors such as lubrication, maintenance, materials, methods, and design which may be as significant as the fuel characteristics which were under test. This resulted in further attention to such factors by individual railroad, equipment, and petroleum companies. Also important was the knowledge gained of the techniques of large-scale field service testing and the necessity for uniform methods of collecting and reporting data. Valuable experience was gained in the relationship between fuel oil, lubricating oil, and engines. The important effect of air filtration on abrasive wear, though already known, was more clearly discerned. Although entirely outside the scope and objective of the test program, these things constitute useful information which is being put to practical use.

Closure

Perhaps the most valuable and enduring thing many of the participants gained from the work in the test group is an understanding and appreciation of the value of coordinated cooperative research. Twenty-two different companies actively participated throughout the test program, even at the risk of embarrassment to themselves, with the hope of learning, not selfishly, but for the general good. For by increasing the range of fuels which can be successfully used as a source of energy for transportation in peace and emergency the American people become the real

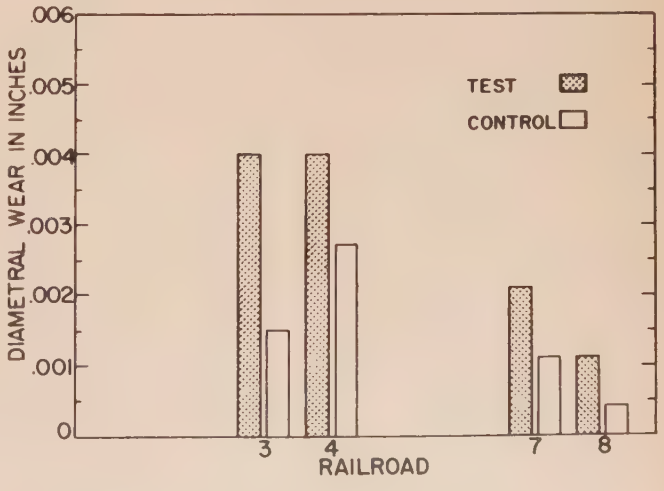


Fig. 37—Average annual connecting-rod-eye wear

beneficiaries. One not having had an experience of this kind simply could not appreciate it. Outstanding was its ability in bringing to bear on any specialized phase of the work the best technical talent in that field of specialization, not only from the cooperating industries and the military and government scientific agencies, but also from industries, universities, and other sources which were not directly involved in the tests. Refreshing indeed was the interest shown in the work by individuals at all levels in science and industry and the willingness—one might say eagerness—of the technical men with the various backgrounds of experience to assist in the exchange of needed information when called upon. Amazing is certainly the word to describe the persistence of the individuals who made up the test teams and the Analysis Panel in seeking the truth and their patience under pressure in striving for agreement upon it. Without the formal and carefully organized and coordinated research plan of the CRC, this extensive series of experiments would never have been successfully conducted, nor through any other channels could the results and conclusions be so universally acceptable.

One wonders at the impelling motives which made the men of the test group voluntarily give, with high morale, so much in time, effort, and genuine labor for this project. They were simply invited to participate—if they wanted to—and if their companies approved. Is this evidence of a growing spirit of team work and cooperation for the common good at the technological level in industry? If this is true, then well-planned, coordinated, cooperative research will become an increasingly potent tool in the general search for better things.

But in spite of all this, this whole undertaking would have been a trying ordeal except for the feeling of comradeship which in a subtle manner permeated the sentiments of the individuals who associated at one time or another, and in one way or another, in this work. Many sincere and lasting friendships originated from mutual respect which developed during this undertaking between engineers and scientists of varied interests who were strangers or only speaking acquaintances before.

The significance of this extends far beyond the personal satisfactions which are gained from it. Personal friendships have opened up whole new channels for communication of ideas. This has a catalytic effect in stimulating creative imagination and is an inspiration to increased endeavor.

DISCUSSION

Discusses Some Benefits Of Cooperative Research

—T. B. Rendel
Shell Oil Co.

AS a preliminary to the discussion on this CRC report, I should like to take this opportunity as Chairman of the CFR Committee to acknowledge the debt which the committee owes to the two gentlemen, Messrs. Seniff and Robbins, who are listed as authors of this paper and to all members of the Railroad Group of the Diesel Fuels Division for carrying out a magnificently organized and well-executed field research project on the effects of composition of diesel fuel on the performance of large medium-speed engines in railroad service and for a well-prepared and delightfully presented report on their findings. If the results have not proved all that the sponsors had originally hoped, it has shown in an outstanding manner what cooperation can do in the way of organizing and carrying out an extensive series of field tests and how to collect large amounts of important data relating to the performance of fuels in engines in various types of field service. The committee as a whole owes a debt of gratitude to its Diesel Fuels Division and its chairman and working groups and participating organizations for this work, and I am glad to acknowledge it and say, "Thank you very much."

I would also like to take this opportunity of stressing the importance of the authors' closing remarks and to point out that this report by no means completes our work on railroad diesel fuels, both cooperatively and individually in our own laboratories.

A trap into which laboratory research workers often fall is that of developing techniques to aggravate or accentuate certain conditions to give quick results. This is understandable, since the normal life of the product or the equipment is often very large and the experimenter cannot afford the time and the money to study the conditions actually occurring under actual service conditions. Carefully carried out field tests, therefore, are an absolute essential to the development of improved fuels and oils and engines. Over the past 30 years the CFR Committee has proved an admirable organization for working out techniques which have proved of inestimable benefit to the subsequent development work both by the fuel or oil producer and the engine manufacturer. Although these programs take time and cost money, they should not be criticized on that account. Remember despite all the latest advance in obstetrical techniques and all the new equipment available in hospitals, it still takes nine months.

But I would point out to you that there is no greater benefit to be obtained from cooperative research than by actually cooperating with it. The ability to sit down with your opposite number in either the engine, fuel, or consumer industry and discuss your problems and methods of solving them is incalculable. Bear this in mind, therefore, when you consider the next steps which we will have to take to develop improved engines with fuels best

suited to their digestive systems and oils to alleviate "ulcers" caused by bad digestion.

All the work reported to you in this report, plus a great deal of additional laboratory testing, is now under study by a Correlation Panel appointed to consider the interrelationship of the various factors noted. The work of this panel should be followed with considerable interest, and I for one confidently expect that some really worthwhile research programs and laboratory techniques will be forthcoming. Such programs well deserve the support of all companies interested in improved engine performance, and the laboratory methods must be tried out if improved products are to be developed. Watch for them! Encourage them! Criticize them! Try them! Not just because they are another piece of research work for the CFR Committee, but because they will assist you in your problem of getting the most out of your powerplant in the transportation industry.

Would Like a Uniform Specification To Simplify Railroad's Fuel Needs

—A. H. Fox
Standard Oil Co. of Indiana

WHEN the CRC program was started, practically all of the diesel fuel used by United States railroads was straight-run virgin distillates. During the past few years, however, there has been a pronounced trend toward specifying of railroad diesel fuels of lower cetane number. For my company, the railroad demand for lower cetane now represents more than 85% of total fuel shipped. Permitting higher proportions of cracked distillates in the fuel blend has helped the petroleum industry to increase the supply and has improved the flexibility of refinery operations.

But how has this shift affected the railroads? Most of them using such lower cetane fuels seem to have little information about how fuel properties affect maintenance or engine life. The CRC project, being appropriately concerned with locomotive availability, provides little direct economic data in answer to this question. Such information is difficult to obtain and is further complicated because as locomotives get older, maintenance expenses increase. Nevertheless, before starting further field tests on fuels inferior to those investigated in the CRC program such factors should be thoroughly appraised. We should therefore like to suggest to the authors and the railroad industry that a study be undertaken to appraise the economic aspects of railroad locomotive operations as related to fuel property factors.

In grading railroad diesel fuels, the practice is becoming common to classify fuels according to cetane number. Thus, just to mention a few, we have fuels with cetane ratings of 50+, 45, 40, 35, and below. This multiplicity of fuels has not been a blessing to the petroleum industry. Most companies do not or cannot manufacture all these grades; few, if any, have storage and distribution facilities to supply such a variety of fuels. Special requirements of this railroad and that for their particular pet additives adds to a situation that has become almost chaotic. Because this job has become almost impossible for one refinery, large or small, the time seems right to suggest a uniform specification which would simplify the railroads' fuel requirements. Certainly the 50 cetane number fuel could be eliminated.

From 1946 to 1953 railroad demand for residual fuels declined 80%. During this period, the demand for railroad distillates (or diesel fuel) increased fivefold. In terms of total liquid petroleum fuels (that is, distillates plus residual) railroad requirements have declined 20%. Meanwhile, the yield of residual fuel as a per cent of crude run has declined from 25% in 1946 to 18% in 1953. Thus, the trend is toward the production of decreasing amounts of residual fuels. If on further consideration it is found that residual fuels are preferred for diesel locomotive service, a readjustment in refinery practices would be necessary to supply the requirements.

The CRC project, covering nine years of planning, scheduling, and execution of tests, has produced highly significant data useful to the railroads particularly and to industry generally. We believe the project is one of the most ambitious cooperative programs ever undertaken by CRC and industry. The persistent devotion of those directly associated with the work and the financial assistance given by SAE, API, WPRA, AAR, and other organizations attest the worthiness of the project. We sincerely hope that the information obtained will assist the railroad industry in making a thorough economic analysis of the data and will stimulate positive action in simplifying railway diesel fuel specifications.

Gives Data from West Coast Survey of Sulfur Effects

—J. L. Broughten
Union Oil Co. of California

TO the railroad industry and the petroleum companies involved in the development of fuels and lubricants for railroad service, the CRC report on full-scale field service tests of railroad diesel fuels represents an outstanding achievement. The authors, the CRC committee group, and the test teams are to be highly commended on the splendid job of organizing the tests and the compilation of the mountainous amount of data into this comprehensive and informative report. This report will long serve as a reference for future diesel locomotive fuel development and is particularly timely in view of the great interest currently being shown by the railroads in economy-grade diesel fuels.

As railroad dieselization approaches completion, it appears that increasing the range of petroleum fuels that can be satisfactorily used in diesel locomotives has now assumed the most important role in reducing railroad operating costs. A broader range of acceptable locomotive diesel fuels will also have a profound effect on the national transportation picture, particularly in the event of a national emergency as well as in normal times. In order to satisfy the tremendously increased demand for locomotive diesel fuel and in conjunction with the search for lower-cost oils, materials not previously considered as satisfactory diesel fuels are being investigated and in some cases are used on a wide scale. The approach made by the CRC program to this problem was very logical because it had been found that the use of less desirable diesel fuels from the standpoint of sulfur content and boiling range has produced undesirable wear and deposit conditions in all types of diesel engines to various degrees. Fig. A shows the effect of fuel sulfur content on cylinder-liner wear in two different types of diesel engines under similar operating conditions. The CRC report presents an excellent comparative picture of the effect of fuel sulfur content on engine wear and deposits in three of the four major types of locomotive diesel engines. It must be noted, as mentioned in the report, that slightly different operating conditions, the use of chromium liners, lubricating oil, and other factors have a definite influence on the rate of cylinder-liner wear so that under identical conditions the difference in the cylinder-liner wear rate of the three types of engines might not be of the same magnitude.

Because fuel sulfur content has always been an inherent problem with West Coast diesel fuels, I would like to present some comparative results obtained on a Western railroad with West Coast locomotive diesel fuels ranging from 0.84% to 3.5% sulfur.

Fig. B shows the average cylinder wear in the CRC test units in comparison with the average cylinder-liner wear of 14 liners after one year's service with a West Coast fuel containing 0.84% sulfur. All of the cylinder-liner wear data shown in this figure were obtained in locomotive type L engines. The locomotives using the 0.84% sulfur West Coast fuel were operated in what would be classified as medium-duty freight service which would be similar to the test conditions of the locomotives operated by railroad 7. It is of interest to note that the average cylinder-liner wear with the 0.84% sulfur fuel is considerably lower than the wear obtained with each of the 1.0% sulfur CRC test fuels and is even slightly lower than the cylinder-liner wear occurring with the low-sulfur CRC control fuels. Top-compression-ring wear with the 0.84% sulfur fuel was in between the top-compression-ring wear values of the 1.0% sulfur CRC test fuels and the low-sulfur control fuels in each case.

In reviewing the test conditions it appears that there are two principal differences between the CRC tests and the tests with the 0.84% sulfur fuel. Undoubtedly there are other minor differences. One is the fact that the average ambient temperatures in the 0.84% sulfur fuel test were higher than the temperatures in the CRC tests, particularly during the winter period. It is believed that this would result in a generally higher engine coolant tem-

perature particularly at idle and under low engine loads in the West Coast test. This factor might easily account for a lower sulfur corrosive wear under idle and low engine load conditions with 0.84% sulfur fuel.

The other point of difference between the CRC tests and the test with the West Coast fuel is the lubricating oil and particularly the alkalinity of the lubricating oil. The extent to which additives in locomotive lubricating oils suppress sulfur corrosive action in railroad type diesel engines is not to our knowledge as yet clearly defined. However, certain types of lubricating oil additives are known to have a pronounced effect on the control of sulfur corrosion in some types of diesel engines. Since the formation of sulfur corrosive compounds in all types of diesel engines is basically similar, it is reasonable to assume that lubricating-oil additives would have some effect in suppressing corrosive wear in locomotive diesel engines.

We are well aware of the pronounced effect that airborne abrasives such as dirt and fine particles from locomotive sanding operations have on cylinder-liner wear. It is believed that the improvement of locomotive air filtration is one of the prime problems confronting United States railroads and that only under optimum air filtration conditions will it be possible to determine the full effect of fuel characteristics, lubricating oil, and other factors on engine deposits and part life.

Regarding locomotive fuel characteristics such as cetane number, boiling range, and sulfur content, we believe that on a comparative basis diesel fuel sulfur content is of prime importance with respect to engine wear and deposit formation and is duly highlighted in the CRC report.

Figs. C and D show locomotive diesel engine pistons after approximately six months' service with distillate locomotive diesel fuels of similar sulfur content. The piston shown in Fig. C was used with a fuel having a boiling range very similar to the CRC test fuels of under 700 F end point. The piston in Fig. D was used with a higher-boiling-range distillate fuel having an end point approximately 735 F. The cetane number of these fuels was 45 and 35 respectively. It is noted that there is very little difference between the deposits on these two pistons. Incidentally, the cylinder-liner wear and top-compression-ring wear with these two fuels were very similar.

In the search for lower-cost locomotive diesel fuels, we are currently cooperating with a major Western railroad in investigating the possible use of light residual fuel for locomotive service. Due to the poor combustion of the light residual fuel under idling and low engine load conditions a dual fuel system is used which incorporates a fuel heating system. Regular distillate locomotive fuel is used at idle and under low engine load conditions, and the dual fuel system automatically switches the engine over to the light residual fuel as optimum engine conditions are reached for the combustion of the heavier fuel. At this time sufficient data have not been obtained to formulate a complete picture as to the effect that the high-sulfur (3.5%) light residual fuel will have on engine wear and deposit formation. However, engine measurements obtained after three months' service under dual fuel operation with the high-sulfur light residual fuel indicate that the cylinder-liner wear with the use of the 3.5% sulfur residual fuel as compared to the lower-sulfur distillate fuel does not show increased wear proportional to the increased sulfur content.

Inspection of the light residual engine parts in the early stages of the test show that the ring-belt-area, piston, and liner deposits are slightly heavier as compared to the deposits obtained with distillate fuels, but there is no indication that these deposits will impair engine operation.

In the course of future development of locomotive diesel fuels and lubricating oils for railroad service, the well organized procedures used in conducting this CRC test program will serve as a pattern for future work along these lines.

Dual Fuel Systems May Provide Best Fuel Economy

—P. V. Garin
Southern Pacific Co.

I HAVE read with particular interest this excellent paper summarizing the work done to date by the CRC on the development of a wider-range fuel quality for railroad diesel locomotives. The reason for our particular interest in this subject is the extensive research program which the Southern Pacific Co., in conjunction with several major West Coast oil companies, has been conducting for more than two years into the utilization of fuels of less critical properties than traditionally used in locomotive

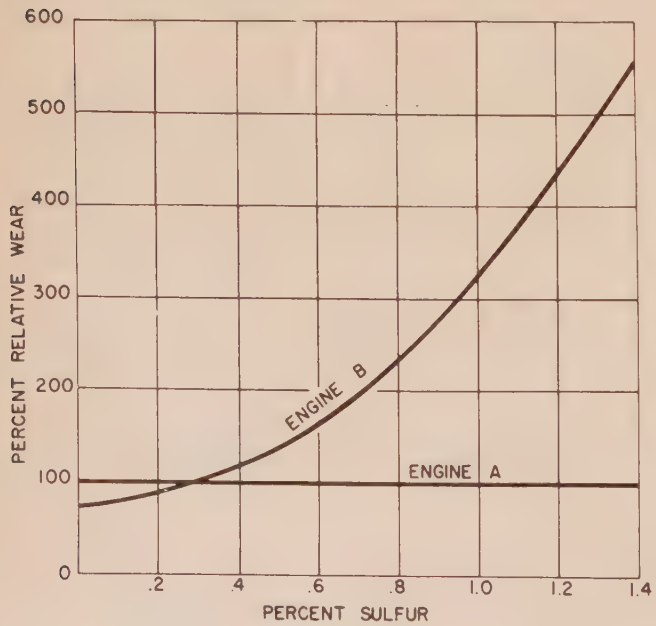


Fig. A—Effect of fuel sulfur content on cylinder-liner wear

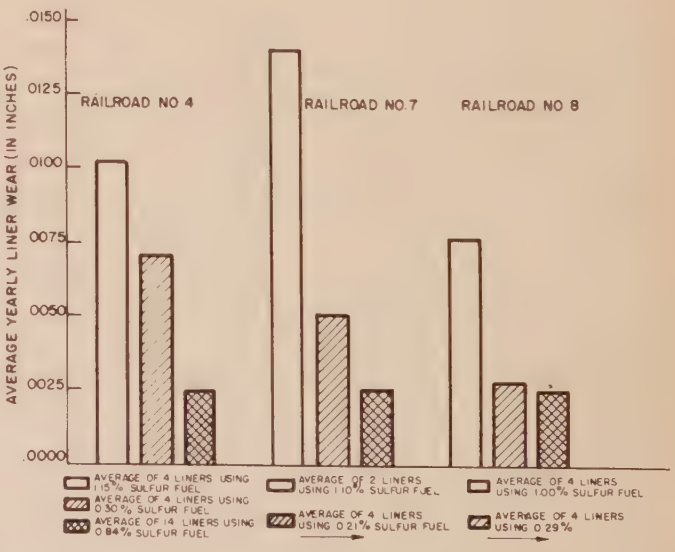


Fig. B—Comparison of average cylinder wear in CRC test units with that in 14 liners after one year's service with 0.84% sulfur content fuel

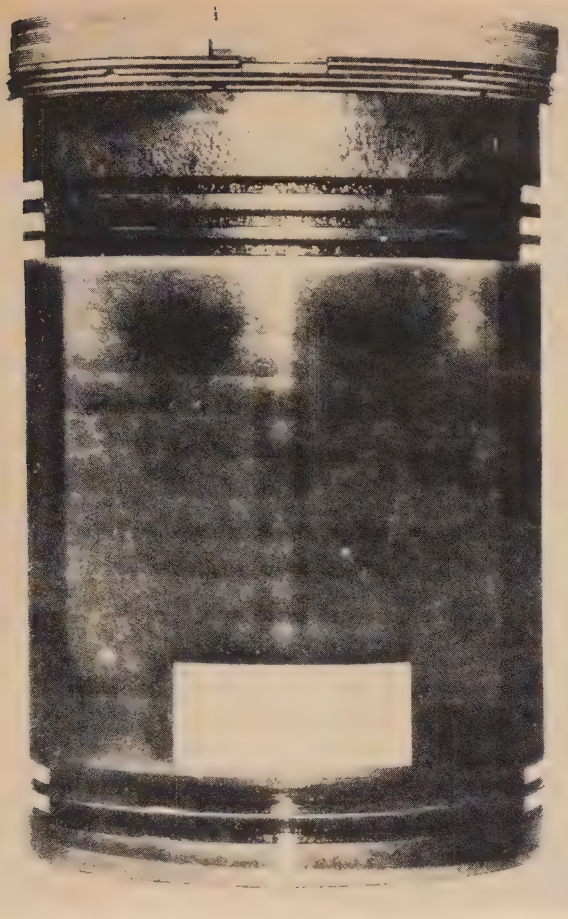


Fig. C—Piston deposits after service with diesel fuel of under 700 F end point

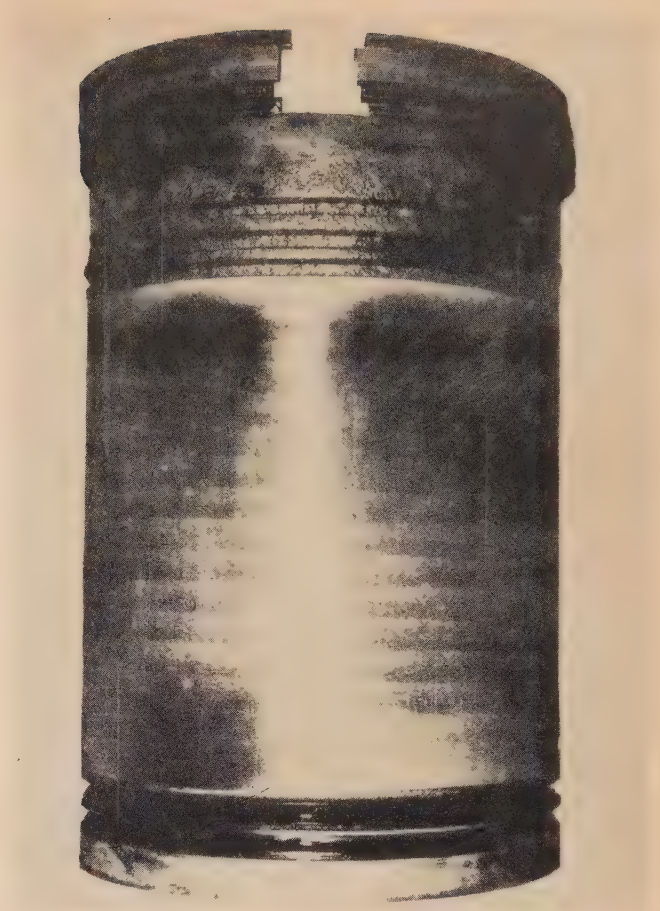


Fig. D—Piston deposits after service with diesel fuel of about 735 F end point

diesel engines. The oil companies on the West Coast have been in a particularly favorable position to cooperate in this investigation, which is based on the utilization of West Coast fuel products.

I can appreciate the amount of effort that has gone into the survey made by the CRC group and feel that they are to be commended on the excellent work they have done. However, I believe that it would be to the mutual interest of all concerned if this investigation were continued and extended to fuels of even less critical properties than those covered by the report.

Southern Pacific Co. has used, even before the adoption of economy-grade distillates, some straight-run fuels with normal characteristics about as shown in the CRC report and experienced no difficulties in their utilization. We realize that the straight-run fuels in other parts of the country have somewhat better normal characteristics than certain stocks in our area.

We adopted the general use of economy-grade distillates containing cracked stocks early in 1954 and have been using them ever since. Our current specification for economy distillates permits a minimum of 34 cetane and a maximum of 1.00% sulfur with end point of about 700 F.

We are currently equipping a number of locomotives with automatic dual fuel systems to utilize distillate fuel for idling and at low outputs, and residual blended fuel at high outputs. This dual fuel system permits the use of a suitable fuel for idling and low-output range and the utilization of a much less critical fuel which can be accommodated at high outputs. The residual blended fuel we are now using has a viscosity of 300 SUS at 100 F, gravity of 17 deg API, and sulfur content of 2.5%.

As a matter of interest, I shall summarize some of the findings of the Southern Pacific diesel fuel research program to date:

1. Experimental diesel fuel research offers an excellent opportunity for railroads to reduce operating expense of diesel locomotives and for the oil companies to develop fuels which will result in production economies through better utilization of refinery output and at the same time provide the answer to ever-increasing demands for distillates in the normal diesel fuel range. By broadening the range of acceptable fuels for diesel locomotives, the fuel supply situation in the event of a national emergency can be greatly alleviated, as the railroads represent a major consumer of middle distillate products.

2. Locomotive diesel engines will digest, particularly at high outputs, fuels of much less restrictive characteristics than generally accepted by railroads, oil companies, or engine manufacturers.

3. When considering the use of diesel fuels of less restrictive quality, changes in one or more of the following properties can be anticipated:

(a) Lower cetane number.
(b) Higher sulfur content.
(c) Higher viscosity.
(d) Higher pour point.
(e) Higher distillation temperatures.
(f) Lower gravity.
(g) Darker color.
(h) Higher residue, soluble and insoluble.
(i) Greater instability.

4. Distillate fuel with double the normal viscosity at 100 F and with high pour point can be successfully used in Electro-Motive Model 567 engines, provided necessary fuel heating facilities are installed. Residual blended fuels with ten times the normal viscosity at 100 F and high sulfur content can be suc-

cessfully used in these same engines at higher outputs, provided automatic dual fuel systems are installed together with necessary fuel heating facilities.

5. Heat exchanger utilizing heat from cooling water can be satisfactorily used to raise temperature of high-viscosity diesel fuels to about 160 F in order to obtain proper injection spray pattern. Electric immersion heaters or exhaust heat exchangers can be used to obtain higher fuel temperatures.

6. High sulfur content of approximately 0.90% has not caused abnormal wear in EMD engines during period of test. The 2.50% sulfur content residual blended fuel has not caused excessive wear when used at high outputs only with dual fuel system, based on experience to date.

7. Lower cetane rating of 35 in all-cracked fuel has not caused maintenance or operating difficulties or parts failure. Some cold-weather starting problems and idling difficulties can be experienced with low cetane fuels unless proper precautions are taken or necessary engine modifications are made.

8. Lower-gravity distillate fuels having higher volumetric Btu or heat content can be expected to afford an additional economy based on increased output per unit volume of fuel. Residual-type fuels with even lower gravity can be expected to increase this advantage. Horsepower output of engines should be adjusted to compensate for the difference in heat content of fuel used.

9. Storage and filtration problems with distillate fuels containing cracked stocks can be expected to increase, particularly with mixtures of certain fuels. Cracked fuels from different sources of supply should be segregated in storage wherever possible.

10. Filtration problems with cracked fuels can be alleviated or overcome to a large extent by the use of higher-capacity fuel filters on locomotives and increased attention to filter maintenance.

11. Filtration of cracked fuels into and out of storage tanks will also relieve the load on filter system on locomotives.

12. Selected dispersant additives can also reduce storage and fitration problems with cracked fuels.

13. Certain metallic-type fuel-oil additives can cause excessive spark emission whereas organic or ashless additives do not.

14. The use of mastic-type or similar fuel tank insulation sprayed on outside surface of locomotive tank is effective in retaining heat in fuel contained in tank.

15. Examination of engines operating with cracked fuel indicates that unit with higher-additive lubricating oil was in cleaner condition than unit with compounded lubricating oil of normal additive level. With residual-type high-sulfur fuels, lubricating oils with high alkaline reserve appear desirable, and more frequent oil drains may be indicated to offset deterioration of lubricating oil condition.

16. Residual-content fuels can be expected to cause objectionable problems at idling and low engine output, such as fouling of engine. The low fuel consumption and resultant saving in fuel cost under these conditions do not justify these potential difficulties. Therefore, the use of dual fuel system utilizing heavy residual fuel or a blend thereof for maximum output and fuel rate and light distillate fuel for idling and low output shows promise of ultimate maximum fuel economy. Such a dual fuel arrangement should include two separate fuel systems with automatic fuel selector system actuated by throttle position or engine output.

17. The automatic dual fuel system permits the use of higher-sulfur fuels by minimizing detrimental effects of this constituent to the engine and lubricating oil that would occur at idling or low outputs when cylinder temperatures and pressures are relatively low, permitting condensation.

18. The alternate use of distillate and residual-type fuels with the automatic dual fuel system is beneficial in purging fuel system periodically and reducing buildup of deposits on injector spray tips.

In conclusion, let me again commend the authors and recommend their investigation be extended to even less critical fuels. We realize that everyone would prefer to use the highest grade fuel available in his diesel engines, but we must weigh the relative overall economics against the type of fuel we use. If we can find fuels that can be utilized satisfactorily and are less critical in nature and which do not aggravate our engine operating and maintenance problems abnormally, an overall net economic balance in favor of the use of such less critical fuels can be realized. This will result in mutual benefits to all concerned. The successful continuation of this program calls for cooperation by all parties, including research, operating, and maintenance forces of the railroad, engine builders, and oil suppliers. The value of cooperative research and team work at the technological level in industry, as stressed in the CRC paper, is the key to resolving the problem of utilization of less critical fuels in diesel locomotives.

Warns Against Use of Lower-Grade Fuels in Small Diesels

—Wayne Lasky
Gulf, Mobile, and Ohio Railroad Co.

THIS successful approach to an important problem by an organized cooperative group consisting of representatives from three different industries—oil refiners, engine builders, and railroads—indicates the possibilities of additional cooperative efforts of this type on other problems.

Messrs. Seniff and Robbins have performed an excellent job in preparing a most interesting report from the voluminous data collected during the six years of tests.

We feel fortunate in having been invited to participate in this test work. I, personally, will cherish for some time my associations with the members of the various test teams serving on the project.

Our railroad has been able to experience diesel fuel savings and overall operation economy by taking advantage of the information developed in these tests.

We feel, however, that we should inject a word of caution to those planning to adopt the use of a fuel having some of the characteristics of the test fuels used in the field test.

We have two items we wish to discuss briefly:

1. The field tests were confined to the large diesel engines commonly used in road and switching locomotives. Our recent experience, however, indicates that some models of small diesel engines may not satisfactorily, or economically, burn fuel of the test grade or "lower grade." We are referring to engines in the range of 70 to 110 cu in. piston displacement. These engines are in service on many railroads on railcars, motorcoaches, or maintenance of way equipment. We also note that some recent lightweight trains, powered with a conventional large diesel engine, have small auxiliary diesel engines in the unit to supply electric lighting, heating, and the like for the train.

It has been our experience that some models of these small engines, when operated on fuels near the 40 cetane classification, get into serious difficulty in a relatively short period. The engines develop excessive carbonaceous and sludge deposits. Difficulty in starting is experienced, and the "stacks" are extremely "smoky." Line-of-road failures can be expected, and maintenance becomes necessarily frequent and costly.

We have experienced the above conditions on some of the small engines operated on our railroad. We found that all of the undesirable conditions were eliminated upon reverting to the use of a 50 cetane fuel. It is interesting to note that on these engines the rate of fuel consumption, in terms of gallons, was 20% lower when using 50 cetane fuel than when the 40 cetane fuel was used, although the 40 cetane fuel contains more Btu per gallon.

We therefore found it economically necessary to stock two different grades of fuel at all terminal fueling stations where the smaller diesel engines are serviced. On some railroads, this could result in the need of a considerable amount of extra fueling facilities, resulting in a substantial investment.

2. We also wish to call to your attention that only one model of switching locomotive was used in the field tests. This was probably the result of the generally accepted opinion that road locomotives are more critical to the type of fuel used.

We have also had indications that some models of diesel switching locomotives do not satisfactorily burn the "lower-grade" fuels under certain and various conditions. Excessive carbonaceous and sludge deposits, as well as objectionable smoking, have been noted with the lower cetane fuels, especially in the lower range of ambient temperatures in winter operation and at times of prolonged idling. This could result in excessive maintenance costs, as well as difficulty in meeting smoke ordinance requirements in some areas.

It is quite possible that these objectionable conditions can be corrected by slight changes in design of some of the diesel locomotive equipment. We hope that the engine builders involved will recognize these problems and help the railroads in correcting the conditions.

Our words of caution can be summed up by stating that the full-scale field service tests have furnished us with a vast amount of extremely valuable data and information, but that it may be necessary for each railroad to survey closely the equipment being used and the special conditions existing on their railroad before the wholesale adoption of fuels of the quality tested during the full-scale field tests are adopted.

It is not our intention for these remarks, in any manner, to reflect adversely on the CRC tests or the work accomplished, and we hope they will be accepted in the constructive manner in which they are intended.

Hopes for More Rapid Method of Fuel Evaluation

—L. S. Crane
Southern Railway System

THE authors and the entire group who worked on this full-scale field service test of railway diesel fuels are to be congratulated for the excellence of this paper.

Perhaps the greatest contribution made by the Group on Full-Scale Field Service Tests of Railway Diesel Fuels has been to develop a uniform procedure for testing and reporting the results obtained. Railroads exist for the purpose of transporting people and things from one place to another. In order to produce this service they utilize locomotives and cars, and since it is a service which they are selling, reliability is of primary importance and economical operation cannot be achieved at the sacrifice of reliability.

It is for this reason that those responsible for specifying fuel requirements of railway diesel engines have been reluctant to use fuels other than those that have proved satisfactory over a long-term service period.

During the first ten years of dieselization on American railroads, locomotive maintenance officers were faced with the primary problem of simply keeping the engines running to provide the aforementioned reliability of service. This was occasioned by the fact that it was necessary to perfect many individual components in the engine in order that they would withstand the day-to-day rigors of railway operation. During the early days of diesel operation those characteristics designating a satisfactory railway diesel fuel were not clearly recognized, and as has been noted in the paper, many operating difficulties and actual road service failures were caused by unsatisfactory diesel fuels. Through painful experience it was learned that if fuels were purchased with high cetane values, low sulfur content, low end point, and low pour point, operating difficulties attributed to fuel could be minimized and maintenance attention could be focused on mechanical difficulties.

Eventually, however, malfunctioning of mechanical parts was overcome to such an extent that the engines achieved a high degree of reliability. Attention was then refocused on economic savings that might be realized by the ability to burn a wider range of diesel fuels. Beginning about 1946, many companies, including my own, began to experiment with diesel fuels having higher end points, higher sulfur contents, and low cetane values. Only the major railroads were equipped with suitable research facilities that would enable them to undertake a program of this type and a majority of the smaller railroads were reluctant to embrace use of the wider-range diesel fuels.

It is for this reason that the CRC program was undertaken, and the results obtained from this program have made a very real contribution to the knowledge on the subject of railroad diesel fuel requirements.

At the same time, the test techniques developed by the group were so well planned that they have been adopted almost universally as a standard test method even by those railroads who were conducting research on railway diesel fuel requirements prior to the inception of the CRC program. One of the most interesting features of this full-scale field service test in my opinion is the small-engine tests which have been conducted by two laboratories in an effort to establish some correlation between small-scale tests and the full-scale field service tests. Mr. Seniff has noted in a previous paper that a minimum of 4000 hr operation and an optimum of 8000 hr operation is required for adequate evaluation of a specific fuel to full-scale field service conditions. I am personally inclined to favor the 8000-hr period which represents an average of 18 months in a field service test. This is an excessively long period of time to wait before an answer to a problem can be obtained. The length of the time interval required produces many complications in maintaining continuity of the test under field service conditions. It is to be hoped that further analysis of small-scale engine test data or further research work along this line will lead to the development of a more rapid method of evaluating specific fuels.

Although the speaker agrees with the general conclusions outlined in the report, caution should be exercised by any railroad before they embrace wholesale consumption of fuels similar to the target fuel described in this report. Our own experience has indicated that diesel fuels with sulfur contents in the range of 0.75 to 0.95% can seriously affect the wear pattern on certain types of railway diesel engines where standard power assembly configurations are used. We have also encountered within recent years severe operating difficulties due to instability and/or lack of compatability between diesel fuels whose properties when determined by standard analytical bench tests appeared satisfactory. It is quite possible that our present laboratory bench tests do not adequately evaluate certain properties of a diesel fuel or combination of diesel fuels, and some of these properties may seriously affect satisfactory operation of the engine. Further investigative work along this line is seriously needed by our industry.

Many Engine Variables Are Closely Related

—J. W. Vollentine
Caterpillar Tractor Co.

WE would like to select only one point and emphasize the relationship between fuel, lubricant, and engine load. It was mentioned that in some cases deposits and wear were not increased with the test fuel. In our experience, increases in both deposits and wear would be expected only if the lubricating oil quality and the engine load were at a level which would cause the increased fuel severity to be noticeable in the results. The variations in engine load between switchers, passenger and freight service were mentioned and it is realized that the effect of lubricating oils was not a part of this investigation. It is, however, quite difficult to discuss the effect of fuel on engine deposits and corrosive wear without also considering these other two variables. Our results have indicated that it is possible for the lubricating oil quality to be sufficiently high for the engine load imposed, that increased fuel severity would not cause additional deposits or wear. On the other hand, it is possible to show no effect from a more severe fuel when the same lubricating oil is used, but the engine load is decreased.

In our experience, the three variables, as well as crankcase capacity and oil change periods, are very closely related and it is usually necessary to consider all simultaneously in evaluating engine test results.

Technical Findings from

A CONSIDERABLE amount of critically needed technical data on the human engineering and related mechanical engineering factors for certain types of collisions have been secured. This was facilitated by the use of anthropometric dummy subjects and where feasible, human subjects.

In order to provide exacting realism for the crash tests initially contemplated, it was decided that field investigations of automobile accidents should be conducted at the scene of the accident. With the cooperation of the Los Angeles Police Department a car equipped for receiving police radio broadcasts was used by the authors to visit many accident scenes. An accident investigation form had been devised for the purpose of obtaining as much technical information relative to the crash as possible. Photographs were taken as necessary, and case files were developed of typical collision accidents.

Perhaps the most glaring inadequacy of automobile accident investigations even when conducted by engineers, is the present lack of means for determining with reasonable accuracy the car velocity immediately prior to impact. Motorists' statements following a collision are frequently slanted for self-protection, and these motorists are often in at least a partial state of shock so that their responses are not entirely reliable. Statements by witnesses frequently are conflicting and suggest that they are confused even though they were not involved in the accident. *Many collision events occur too quickly for even a group of trained observers to perceive them.* This statement has been validated in another study.[1] Only high-speed cameras can provide such information on a time basis expanded sufficiently to allow the human mind to perceive the events. The determination of a pre-impact velocity based on skid marks, the extent of deformation of a car, and the other physical conditions peculiar to its impact have many elusive aspects. There are some experts who will provide velocity estimates but, unfortunately, none of them have had the opportunity to calibrate their estimates with known standards, such as controlled crashes.

The problem of devising a controlled experimental automobile crash in which realism had not

[1] "Automobile Crash Effects," by D. M. Severy. Presented at California State Governor's Safety Conference, Engineering Division, October, 1954.

Automobile Impact Studies

D. M. Severy and
J. H. Mathewson, University of California

This paper was presented at the SAE Annual Meeting, Detroit, January 9, 1956.

been sacrificed was next considered. If it were to be a 2-vehicle crash, variables such as the following would have to be considered: (1) the percentage lateral offset of impact, (2) the angle of impact, (3) the velocities of impact, (4) the masses of the colliding vehicles, (5) the assurance that the vehicles would collide in view of the high-speed cameras, (6) the problem associated with instrumentation and data recording for two moving vehicles, and (7) the problem of finding suitable test vehicles, identical with those used in other tests at different impact velocities, so that their structural properties correspond. After due consideration of these problems it was decided that initial tests would involve a single automobile to be crashed into a fixed barrier. This would eliminate the variables of the opposing vehicle and the problem of knowing where these vehicles would crash.

In addition to the barrier collisions, a series of

AN engineering-oriented summary of the more significant technical findings derived from 12 automobile collision experiments conducted at impact speeds between 7 and 55 mph is presented here.

Use of both human subjects and anthropometric dummy subjects facilitated procurement of critically needed data on human engineering aspects of collision injury minimization. An evaluation is made of four conditions of motorist restraint in terms of the force system applied to the motorist.

Deceleration patterns, frame deformations, automobile impact analyses, and similar engineering data are given for several impact conditions. The instrumentation techniques for the collection of data in automobile collisions are briefly presented.

Table 1—Automobile Collision Instrumentation

MECHANICAL	ELECTRICAL	PHOTOGRAPHIC	PHYSIOLOGICAL
Car frame and body position indexes for providing permanent deformation data	Strang gages for measuring safety belt loadings, etc.	High-speed motion picture cameras for micromotion analysis	Anthropometric dummy subjects to provide data on the kinematic behavior of the human form during collision; to provide an approximate indication of the trauma associated with a particular collision situation
Mechanical accelerometers provide acceleration data independent from electrical devices	Electric accelerometers for providing basic acceleration data	Speed-graphic (still) camera for blow-up shots at instant of maximum body collapse	
Human subject grease-paint for providing torso flexure references for kinematic behavior study	Recording oscillographs for recording loadings, accelerations and similar information	Moderately high-speed g-resisting camera for recording subject responses to collision from within car	Human subjects for evaluating human tolerance limits to collision for specific restraining devices and for observing kinematic behavior of the human body during collision
	Delay timing circuit for actuating speed graphic camera	Standard-speed movie cameras to provide documentary film material	
	Frequency standard for providing accurate millisecond timing in the field for micromotion analysis	Calibrated marker boards for micromotion analysis	
		Reference targets for micromotion analysis	
	Probe circuit to give oscillograph record of instant of contact for collision	Camera-oscillograph synchronization device to provide a common time reference	

rear-end collisions were conducted for the purpose of determining the cause of allegedly serious injuries resulting from relatively light rear-end impacts. Also conducted were high-speed oblique impacts with a 12-in. curb and concrete-type bridge rails.

Automobile Collision Instrumentation

As far as the authors can determine from the literature, the first attempt to instrument a car for collision was made by the University of California. Instrumentation for these early tests consisted only of equipment essential for micromotion analyses of high-speed motion picture film of the collision. During subsequent tests, conventional and special instrumentations were added in order to obtain data pertaining to the many complex aspects of the crash injury problem. The instrumentation given by Table 1 has been successfully applied by U.C.L.A. for the collection of data in automobile collision research.

Experience has shown that aircraft, automobile, and specially constructed vehicles, such as the linear decelerator sled used by Stapp [2] develop vibrations of their structural components during collision. In such cases the vibration of the structural member at its natural frequency may completely mask the acceleration measurements sought. For the particular structures of the cars instrumented for the barrier collisions, the natural frequency varied between 60 and 90 cps. (See Table 2.) Accelerometers with natural frequencies in the range of 60 to 120 cps provided reasonably good records.

Multichannel Recording Oscillograph—Only specially designed precision equipment will satisfy the data collection requirements of a collision event that is completed in less than a quarter of a second. It is not only important to know the force-time histories of the various components of the colliding vehicle and its occupants, but it is also valuable to know the time relationship they bear to one another. For this reason, all of the electrical signals detected during a collision event are recorded on a single multichannel oscillograph provided with a crystal-controlled common time reference. (See Fig. 1.)

Camera-Oscillograph Synchronization—In order to provide a basis for relating the physical observations recorded by the camera to the physical factors recorded by the oscillograph, a means of synchronizing these two variable speed-recording functions was required. This consisted of feeding the signal from a magnetic pickup on the drive wheel of the camera to one channel of the oscillograph via a line between the camera and the mobile instrument truck. The camera film speed is identified by a 60-cps timing-light marking on the edge of the film. The photographic paper speed of the Hathaway recording oscillograph is identified by the transverse line applied each one-hundredth sec.

The common origin for the two recording systems was selected as the instant the front bumper of the

Table 2—Natural Frequencies of Car Structure

Position of Accelerometer	Type of Accelerometer	Average Natural Frequency, cps
Car 2		
Door post (center)	Hathaway electrical	92
Door post (center)	Statham electrical	62
Door post (base)	Gross mechanical	60
Door post (top)	Gross mechanical	60
Car top	Hathaway electrical	67
Car 3		
Door post	Hathaway electrical	90
Door post	Statham electrical	65
Door post (top)	Gross mechanical	70
Car top	Hathaway electrical	65

[2] AF Technical Report No. 5915, Part 2 (December, 1951), "Human Exposure to Linear Deceleration," by J. P. Stapp.

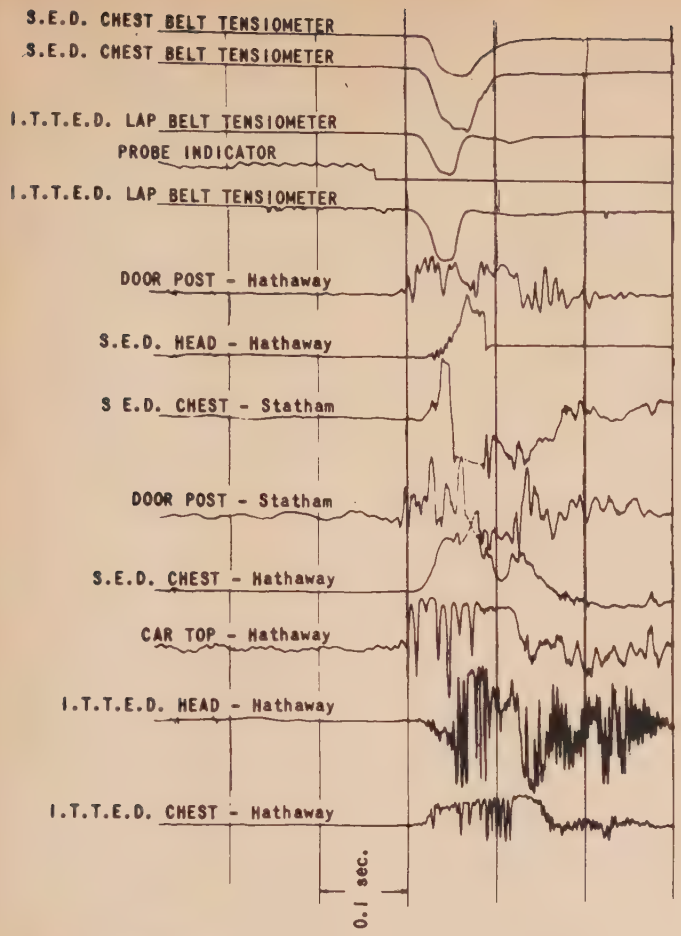


Fig. 1—Oscillograph record of collision with fixed barrier at 29 mph

car contacted the barrier. A probe was mounted so that it extended through the radiator flush with the front bumper. A slight axial movement of this probe actuated a switch connected to one channel of the oscillograph. (See "probe indicator," Fig. 1.) The broad end of the probe was clearly visible to the high-speed camera so that the instant of contact was easily identified.

Human Engineering Instrumentation—The instrumentation for the physiological aspects of the collision event is the most difficult to accomplish. Obviously, physiological aspects can best be evaluated through the use of human subjects. Considerable limitations are associated with this procedure, however, as should be apparent. The anthropometric dummy is a mechanical substitute for a human subject. This device has permitted the evaluation of many human engineering problems without exposing human subjects to especially hazardous conditions. For most collision conditions involving accelerations or decelerations in excess of approximately 5*g*, the kinematic behavior of the anthropometric dummy corresponds reasonably well with that of the human subject. The magnitude of the force system does not have to be overwhelming before the human subject's performance corresponds closely with the dummy subject because the rate of application of the forces will, in most instances, exceed human response time. To bridge the gap between the human subject and his mechanical substitute, and as an additional precaution, some human engineering problems require evaluation using animal subjects representative of the human for the particular exposure to injury under investigation. Experimental restraining devices, steering wheel configurations, and padded instrument panels are examples of automotive safety equipment which could be so evaluated. At least one of the larger automotive companies has already followed this procedure.

Vehicle Collision Deceleration Patterns

Car Body Stucture and its Relationship to Injury Minimization—Virtually all types of vehicular collisions could occur without motorist injuries if it were practical to build onto the car a sufficient amount of structure of suitable strength to keep within safe limits the decelerations (and the accelerations) encountered during collision. An opposite extreme of the protection afforded the motorist by such a hypothetical car structure is typified by the motorcycle. For most collisions it provides neglible protection because the motorist in this case is outside the structure instead of the converse. Obviously, a reasonably satisfactory compromise lies somewhere in between these two extremes in motorist protection. Actually, the American automobile has a structure which is rather effective in moderating the force systems applied during most collisions. This property is illustrated in Fig. 2. Here the car frame has been instrumented for the purpose of determining the profile of peak de-

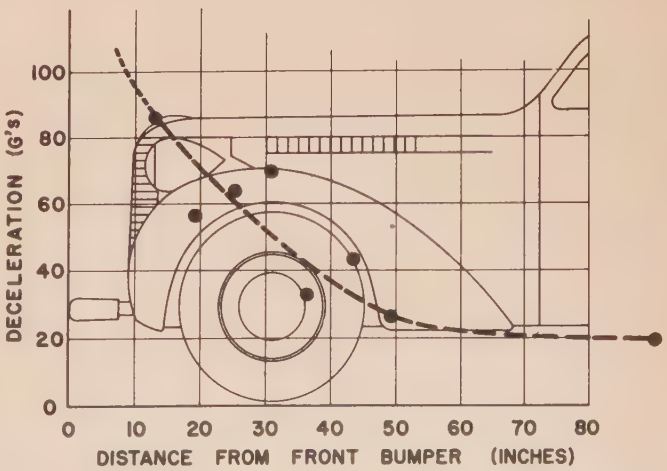


Fig. 2—Moderation of collision forces by the front-end structure at 25-mph impact with fixed object

celerations from the front of the car to the motorist compartment during a direct impact with a large-area fixed barrier.[3] For this type of impact, because of the progressive collapse characteristics of the front-end of the car, the deceleration rate of the intact portion of the car body surrounding the motorist has been reduced to less than one-tenth the force applied to the bumper of the car. In many cases, as will be pointed out later, the motorist derives little or no advantage from this force reduction unless, of course, he is restrained by a suitable device, such as a safety belt.

Similarity of Deceleration Patterns for Different Vehicles—An interesting comparison showing the similarity of deceleration patterns for an automobile-barrier collision, an aircraft-barrier collision and a simulated collision by a linear decelerator is provided by Fig. 3.[1] Perhaps the most striking similarity is that the peak deceleration of the subject's chest, regardless of the kind of vehicle he rides in, always exceeds the peak deceleration of the intact portions of the body of the vehicle in which he rides. Within the limits of practical design for passenger compartmentation, this condition will probably always exist (but not necessarily in the degree shown), even when the passenger is effectively restrained, as in the examples given by Fig. 3. In addition to the yield provided by the limited degree of freedom necessary for comfort, within a restraining device, other factors contributing to the excessive deceleration of the subject are:

1. Plastic and elastic deformations of the anchorage, hardware, and webbing of the restraining device.
2. A degree of yielding of the unrestrained portions of the subject, such as flailing of his limbs and movements of his upper torso and head.
3. Elastic deformation of the subject for the areas of the body subjected to the restraining pressures.

These factors also account for the second similarity conspicuous in Fig. 3, namely, the delayed onset of decleration of the subject relative to the vehicle in which he is riding. A third factor is the characteristically abrupt rate of onset of deceleration for the subject's chest as the deceleration increases to a peak. This initial peaking is followed by a rather abrupt, though brief, partial recovery which leads to a second abrupt increase in deceleration followed again by a partial recovery which, after its initial phase, decreases at a rather moderate rate to zero. A fourth similarity of the curves in Fig. 3 concerns the pronounced double peak in the chest deceleration patterns which is ascribed to the mass-spring relationships of the occupant, his restraining device and anchorages.

Direct Impact with Fixed Object—For the purpose of evaluating the effectiveness of several configurations of restraining devices, three collisions with a fixed barrier were conducted, each involving approximately the same impact conditions. The three deceler-

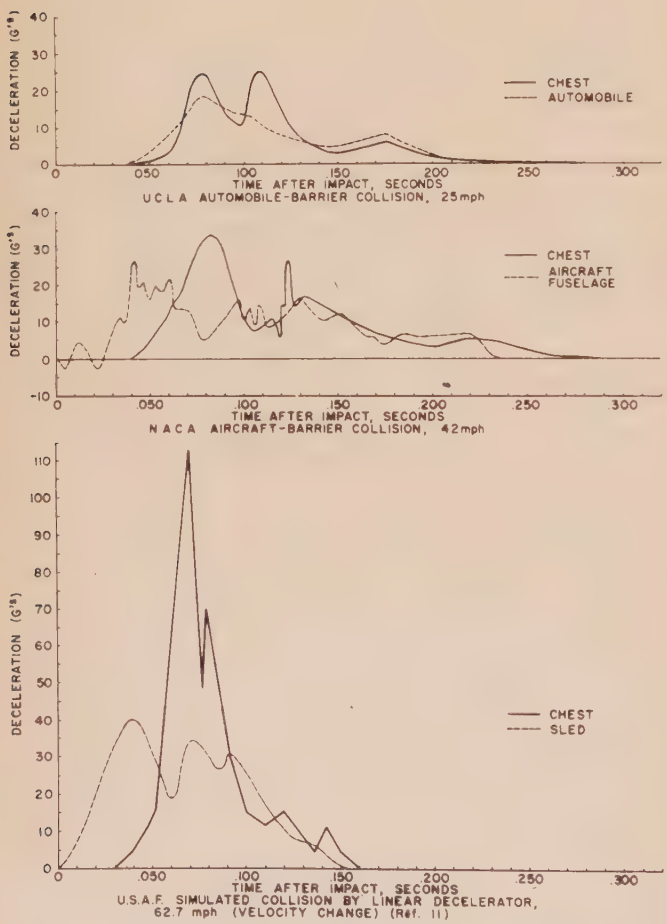


Fig. 3—Comparison of longitudinal deceleration patterns of U.C.L.A. automobile-barrier collision, NACA aircraft-barrier collision, and USAF simulated collision by the linear decelerator

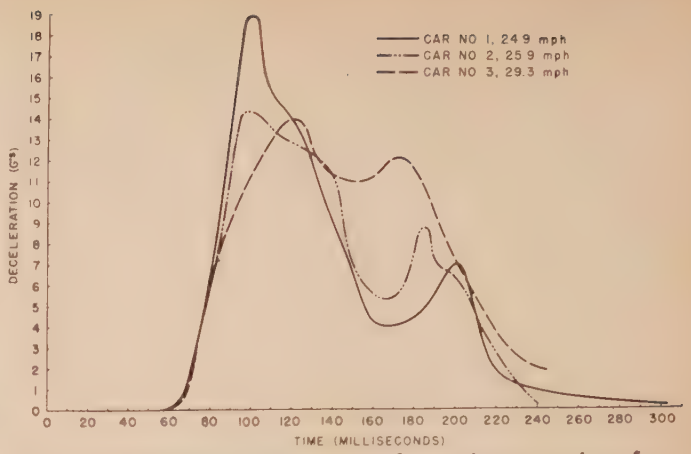


Fig. 4—Deceleration characteristics of same intact portion of car body during collision with fixed barrier

[3] Details concerning the instrumentation used to obtain these peak decelerations are given in the discussion concerning Figs. 18, 19, and 20. The somewhat irregular distribution of peak deceleration rates associated with the front portion of the frame is attributed, a least in part, to the presence of attachments such as the engine mounts. Thus, for example, as the front of the car frame is collapsed, the engine comes into direct contact with the barrier and provides the effect of added structural strength to the frame at the points where the engine is mounted. A homogeneous structure would be expected to fail with an orderly reduction of peak deceleration rates from the collapsed front-end to the undeformed portions of the car body.

Table 3—Comparison of Deceleration Curves of Fig. 4 for Three Collisions with a Fixed Barrier

1	2	3	4	5	6
Car No.	Velocity, mph	Area Under g-t Curves, square units[a]	Velocity-Area Correlation, Column 3/2	Rate of Onset, g/sec	Peak, g
1	24.9	100	4.02	810	18.7
2	25.9	102	3.94	685	14.3
3	29.3	118	4.03	560	14.0

[a] Each square unit represents 1g for 10 millisec.

ation patterns for the same part of each car (the car top) are presented in Fig. 4. In these three experimental runs, the speed did not differ by more than 4.4 mph, and the cars were the same make, model and year, and each separately struck the same fixed barrier by direct impact. The differences in graphs of Fig. 4 show the difficulty encountered in attempting to hold all but one of the variables of a collision constant for the purpose of studying the effect of that variable for a given collision situation. A closer examination of the specific values of these curves shows that they do correlate rather well, considering they represent collision decelerations. The significant characteristics of these curves are given by Table 3.

Resultant Car Decelerations for the Vertical Plane—Collision deceleration is seldom unidirectional in that impacts do not usually involve contacts in which the direction of motion of their mass centers coincide. This means that rotational as well as translational accelerations generally develop during collision.

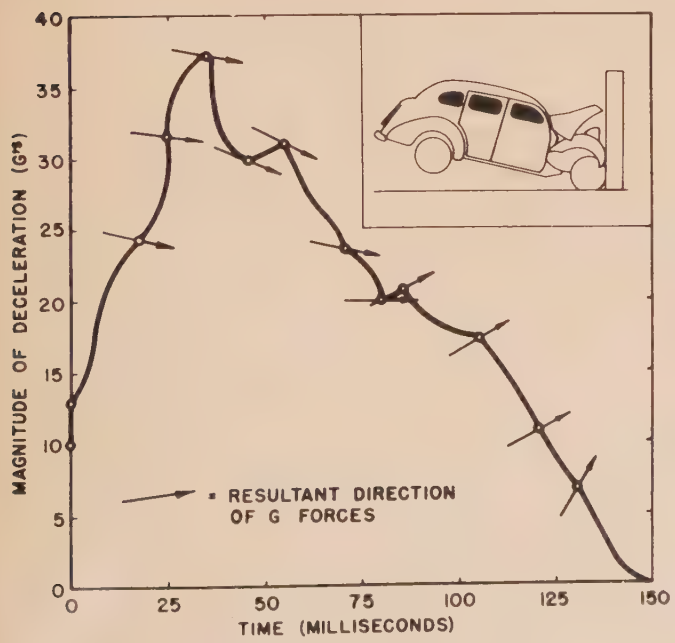


Fig. 5—Resultant direction and magnitude of car frame on left side, under firewall barrier collision

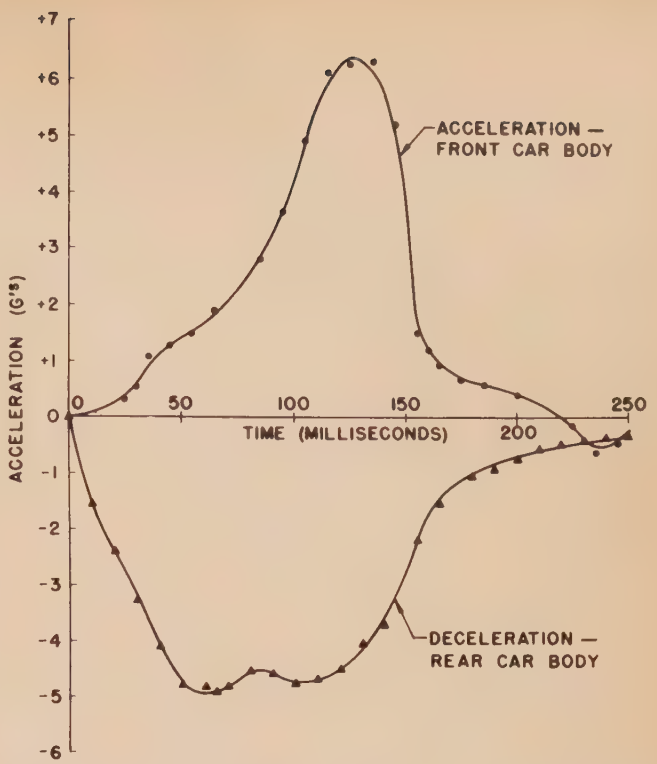


Fig. 6—Deceleration-acceleration patterns for rear-end collision at 20 mph

Even direct impacts with fixed objects having large flat surfaces involve, in addition to longitudinal deceleration, vertical accelerations and decelerations. The extent to which accelerations other than longitudinal are present is important since they also have an influence on the motorist injury pattern. Fig. 5 shows the resultant direction and magnitude of the longitudinal and vertical decelerations for the car frame during direct impact with a fixed barrier. Initially and during most of the collision the decelerations are predominantly longitudinal but toward the end of the impact period a relatively strong vertical influence is evident. The sequence of strong longitudinal decelerations followed by smaller but increasing vertical accelerations may be fortuitous since the motorist is not likely to be dislocated vertically before the high longitudinal decelerative forces are applied. This factor requires consideration when contemplating the use of certain restraining devices.

Vertical accelerations have been observed for head-on and side impacts but no attempt as yet has been made to determine their magnitudes and injury potential. The vertical acceleration values for the barrier collision apply only to this specific accident type. Vertical acceleration for a rear-end collision has been evaluated elsewhere.[4]

Deceleration-Acceleration Patterns for Rear-End Collision—A good example of the complex nature of automobile collisions is provided by the rear-end type. Referring to the deceleration-acceleration pat-

Table 4—Oblique Impact Data

	1949 Ford 2-Door Sedan	1949 Ford 4-Door Sedan	1946 Buick 4-Door Sedan
Barrier Configuration Struck	Bridge rail and curb No. 3	Bridge rail and curb No. 3	12-in. curb type B
Impact Velocity (Micromotion Analysis), mph	50	40	41
Angle of Impact (Measured at Site), deg	20	30	18
Peak Longitudinal g (Micromotion Analysis)	17	10	1
Average Lateral g (Micromotion Analysis and Calculation)	100g for 8 millisec[a]	90g for 20 millisec	80g for 20 millisec
Remarks	Car struck bridge railing obliquely at 50 mph and careened to parallel railing at 38 mph	Right car door sprung open and closed again after impact. Deflection on impact similar to 2-door sedan	Car hurled over curb with 11-deg change in direction

[a] Effect of force is not too significant because of extremely short duration.

terns of Fig. 6, it will be seen that both the rates of onset and peak accelerations differ for the colliding vehicles. The rear car is initially abruptly decelerated. The rate of acceleration for the stationary vehicle is, on the other hand, initially relatively gradual, but because of a combination of factors described elsewhere [4] the injury potential for the occupants of the struck car for speeds up to at least 20 mph is many times the injury potential for the motorists in the striking car. The presence of a headrest extension on the seatback would allow the motorist to benefit from the slower rate of onset of acceleration. The unsupported head develops a velocity differential with respect to the shoulders and the whip-lash injury mechanism is consequently set into motion.

Oblique Impacts with Fixed Objects—The Materials and Research Department, Division of Highways of California, has conducted many automobile impact tests for the purpose of evaluating different curb and bridge rail configurations.[5] The University of California cooperated in three such impacts as in Fig. 7 for the purpose of determining the force systems acting on the motorist during these collisions. Quantitative information is presented in Table 4. These moderately high-speed oblique impacts were characterized by relatively low longitudinal decelerations

Fig. 7—High-speed oblique impact with bridge railing

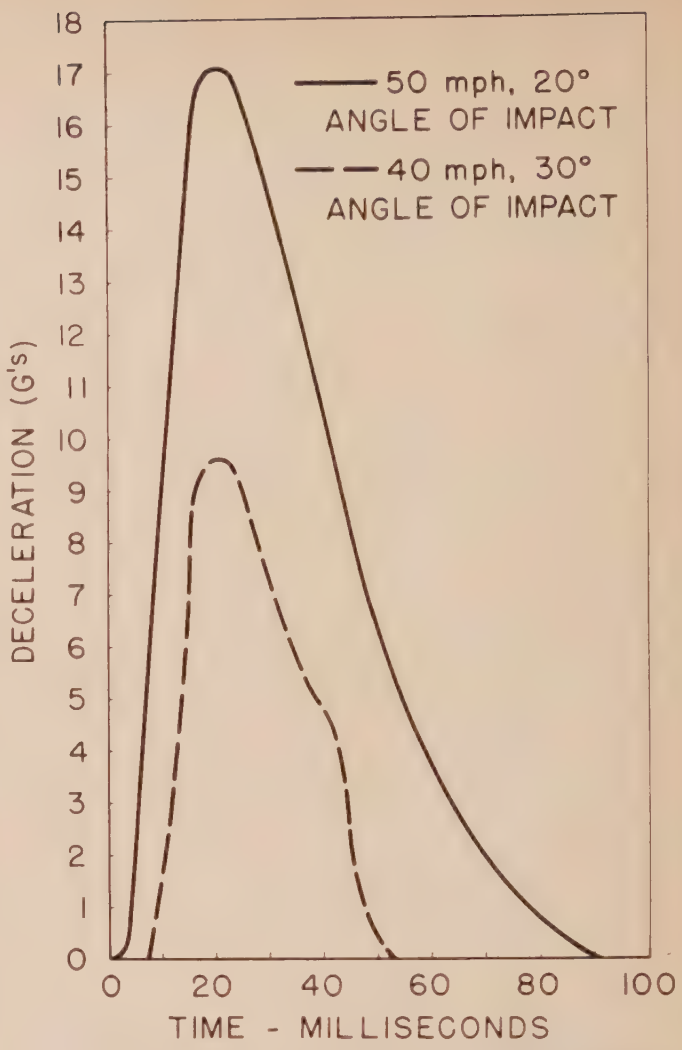


Fig. 8—Deceleration patterns of 1949 Fords colliding obliquely with bridge railing

which last approximately 100 millisec accompanied by very high lateral declerations which last for no more than 20 millisec. When these lateral decelerations are of very short duration, and for those cases when the car doors remain closed, they tend to be more of a hazard in disorienting the motorist (thereby leading to loss of control of the car) than in directly producing injury.

During these collisions it has been observed through high-speed photography, that the doors open and close in a period of time so brief that the action escaped the notice of trained observers. For some oblique impacts, this brief period is nevertheless sufficient, considering the prevailing force system, to eject the motorist through the door.

The recent improvements of door lock mechanisms made by several manufacturers should correct or

[4] Canadian Medical Services Journal, Vol. 11, November, 1955, pp. 727–759. "Controlled Automobile Rear-end Collisions—An Investigation of Related Engineering and Medical Phenomena—Series I," by D. M. Severy, J. H. Mathewson, and C. O. Bechtol.

[5] State of California Department Public Works, Division of Highways, Materials, and Research Department, December 31, 1953: "Roadway Barrier Curb Investigation," by J. Beaton and H. Peterson.

Table 5—Performance of Three Belt Configurations

Restraint Configuration	Peak-Force, lb	Belt Stress Duration, sec	Force-Time Units	Relative Restraining Effectiveness[a]	Remarks
Chest	1735	0.105	2300	100	The chest belt dissipates a greater amount of body crash energy than the other (see below) two units without increasing the peak stress to the body. Physiologic considerations are discussed elsewhere
Lap	1725	0.057	1500[b]	77	The lap belt is a somewhat less efficient restraining device for front seat occupancy and for the front-end impact situation. It does not prevent the head and upper torso from striking the forward surfaces of car interior. Energy is then absorbed from the most vital parts of the anatomy by injury-producing mechanisms
Shoulder Loop	1735	0.143	1550	68	The shoulder belt dissipates the least amount of body crash energy. However, it maximizes (for a single restraint) the protection of the most vital parts of the anatomy, the head, and trunk

[a] The area under the force-time curves (Figs. 10, 11, and 12) given in "units" by this table provides an index of the relative restraining effectiveness of each configuration. For ease of comparison, these values are divided by a constant to make the most efficient equal to 100. Consideration was given to the fact that lap and chestbelt tensiometers were arranged in parallel so that their force-time values were additive while the shoulder belt tensiometers were arranged in series so that their values were averaged.

[b] This value must be corrected by the factor 200/170 to account for weight differences of dummies before determining the relative restraining effectiveness of the lap belt.

greatly improve this situation. The longitudinal deceleration patterns for the 2-bridge rail impacts are presented in Fig. 8. These patterns refer to decelerations of an intact portion of the car body (car top).

Evaluation of Motorist Restraining Devices

Human Body Dynamics as Affected by Changes in Belt Configuration—The dynamics of the human body during a collision were evaluated for five conditions of restraint (Fig. 9). Four of the collisions were with a fixed barrier at 25 mph (using anthropometric dummies), while the fifth was a 20-mph rear-end collision between a stationary car occupied by a dummy subject and one driven by a human subject. The results of this latter test have been used as the basis for estimating body dynamics for a collision with a fixed object at 25 mph. Quantitative data for three conditions of restrain are given in Table 5.

Load Applied to Various Belt Configurations—It should be apparent that, providing the restraining devices are properly designed, the configuration

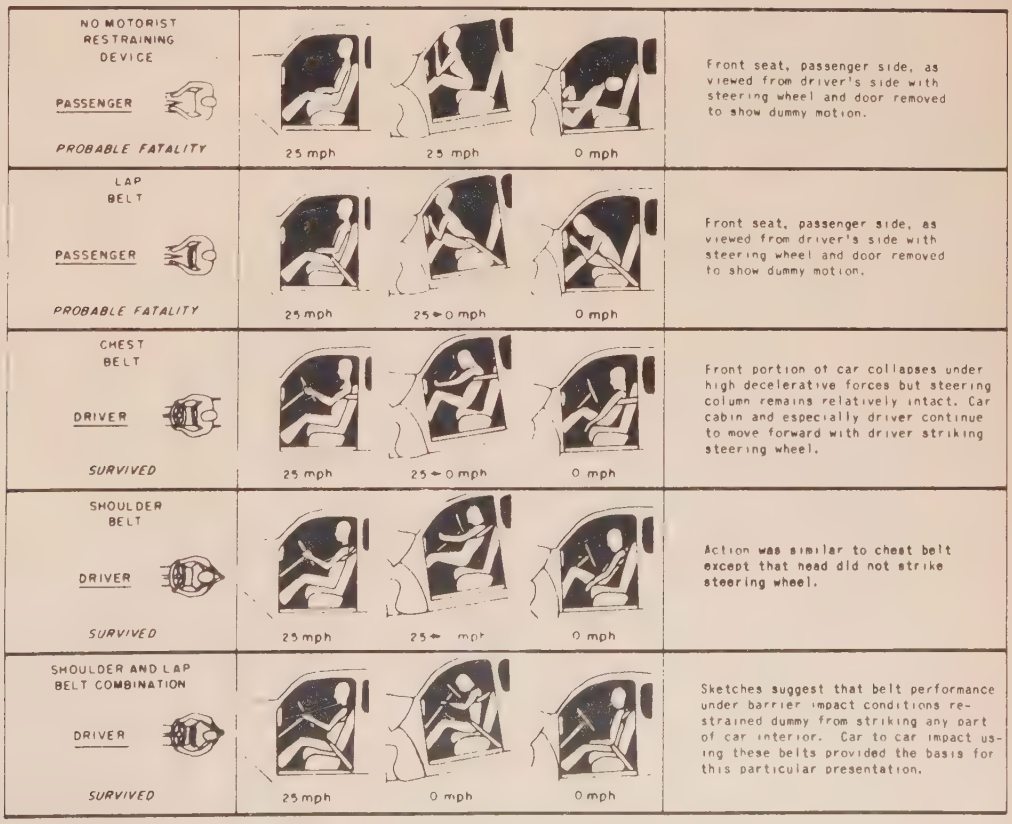


Fig. 9—Effect of varying safety belt configuration for 25-mph collision with fixed barrier, reproduced from high-speed motion pictures of actual collisions.

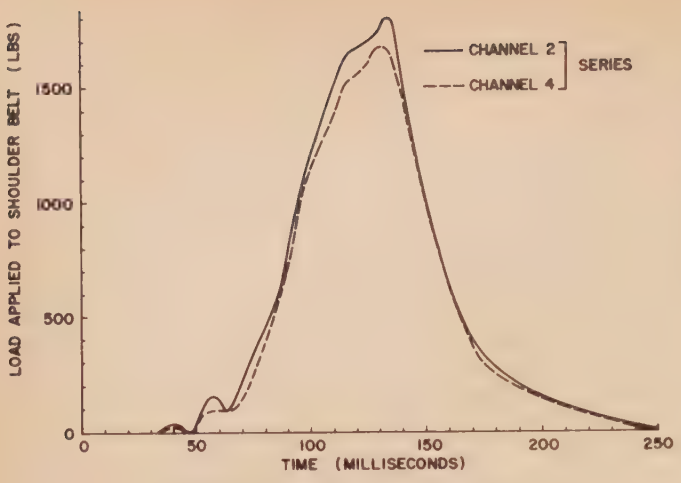


Fig. 10—Shoulder force-time history, S.E.D., car 2

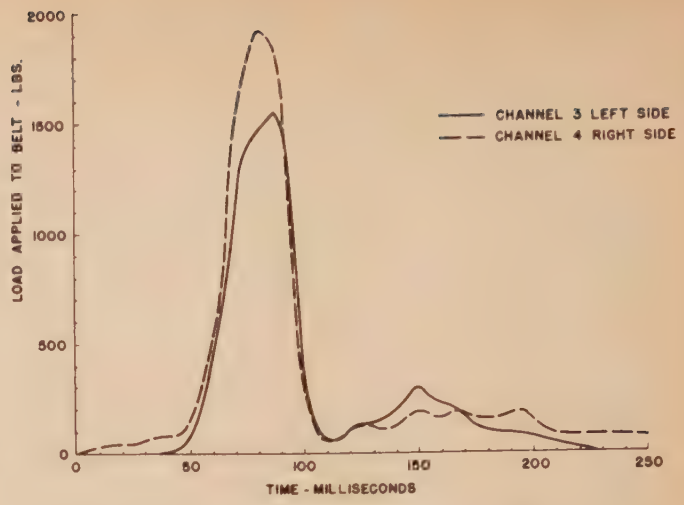


Fig. 12—Lap belt force-time history, I.T.T.E.D., car 3

which most effectively prevents the body from striking the car's interior during identical collision conditions will be that having the best force-time performance. Belt tensiometers provide a force-time history for the loading conditions to which the belts were subjected. Tensiometers were used in pairs for the chest and lap belts, one near each anchorage of the belt, in order to detect differential loadings due to lateral acceleration components. The fact that average peak forces for all three belt configurations were approximately the same must be regarded as coincidental even though the cars were crashed at approximately the same speeds. Variations in dummy weight (170- and 200-lb dummies), deceleration rate, and belt configurations explain why this similarity in average peak force must be considered coincidental. The tensiometer records are shown in Figs. 10, 11, and 12. In Fig. 10 the curves show the close correspondence of data for the shoulder-loop belt since the two tensiometers, placed in series, were measuring the same loading at the same point. Also significant is the fact that the rate of onset and subsequent reduction of force for the shoulder belt is relatively gradual. Figs. 11 and 12 are tensiometer readings for chest and lap belts during the same collision. The tensiometers on the left side for both dummies (both in front seat) gave lower readings than those on the right side, indicating that frame failure on the left side was more extensive than on the right. Examination of the frame deformation plot confirmed this conclusion. Each belt configuration, per se, is capable of effectively restraining the body against the decelerative forces of this collision. Additionally, however, it is important to point out that the effectiveness of each configuration is a function of the geometry of the front seat compartment; that is, the extent to which the spatial arrangement precludes the striking of occupants' body components against forward interior surfaces. To the extent that human body components dissipate their energy by striking the cars' interior surfaces, the belt tensiometers record correspondingly reduced force values for a given collision event. The superiority of the shoulder and chest belt restraints over the seat (lap) belt restraint can be seen by comparing the areas under the curves of Figs. 10, 11, and 12. Conjecture arises, of course, as to the force-intensity level at which each configuration becomes an injury-producing mechanism by virtue of the force concentrations or distributions it produces on the body. Physiological evaluation of each configuration should provide the answer. No attempt was made to include this factor in Table 5 because no tests have been conducted to determine these force-tolerance levels.

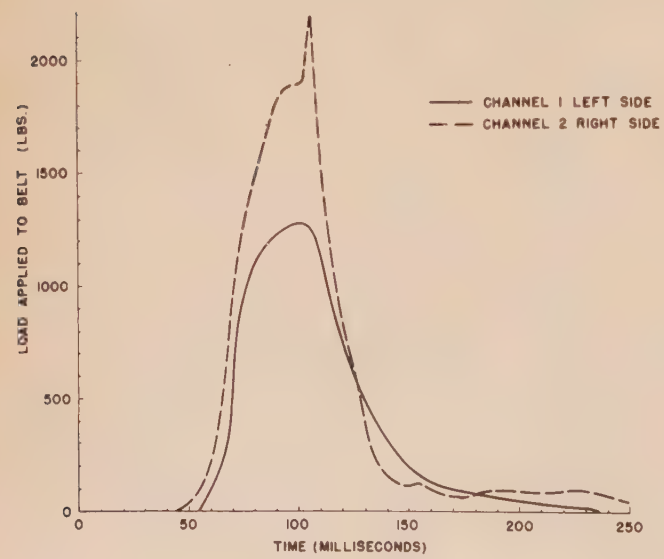


Fig. 11—Chest belt force-time history, S.E.D., car 3

Restraining Devices other than Safety Belts—The term, motorist-restraining devices, has been used in

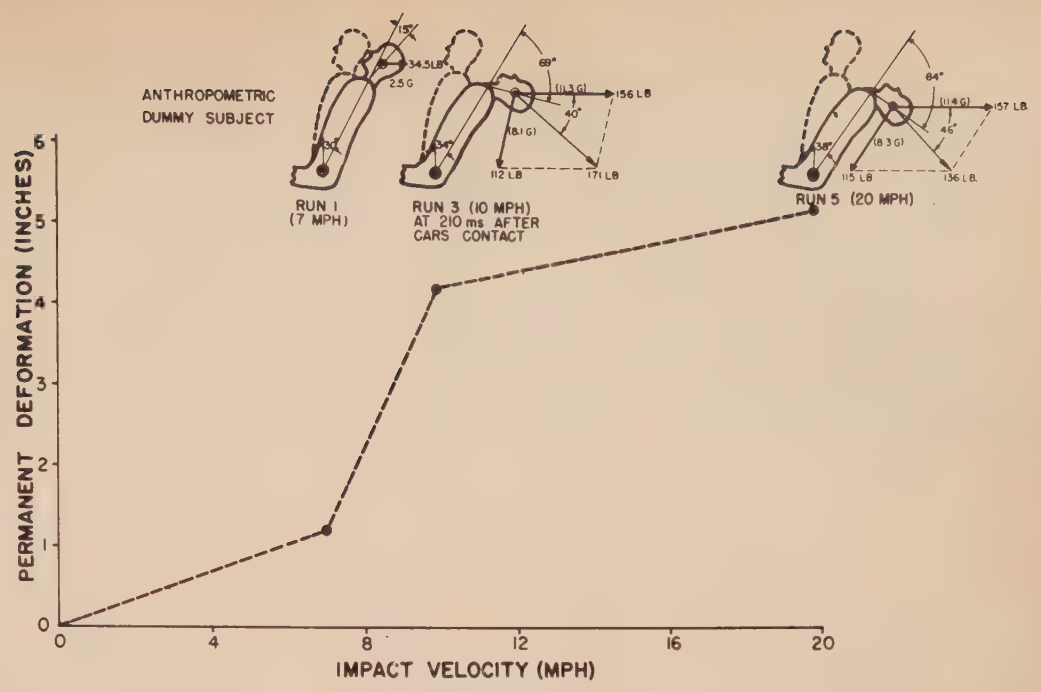


Fig. 13—Permanent deformation of car struck in rear versus impact velocity and their relationship to head whip forces

order to allow for recognition of features other than safety belts which may provide restraint against collision forces applied to the motorist. For example, anything that adequately provides support for the head and neck against the forces of a rear-end collision should be recognized as a restraint. The diagnostically difficult injury pattern identified with the low-speed rear-end collision is discussed in a separate study.[4] The medical profession refers to this type of injury as "whip-lash injury of the neck." In a series of rear-end collisions at speeds up to 20 mph, the force system acting on the driver's head was determined. (See Fig. 13.) It was found that the magnitude of the force applied to the subject's neck did not necessarily correlate with the impacting velocity. For example, in a comparison between a 10-mph impact and a 20-mph impact, using a dummy subject in each, it was found that the force delivered to the head was larger for the 10-mph impact. It is recognized, of course, that the height and strength of the seat back as well as the collapse characteristics for the contacting sections of the cars are variables which cause the forces applied to the motorist during a rear-end collision to be somewhat unpredictable. In fact, it is this aspect of the problem that leads to confusion.

It was found that in a rear-end collision, the cars crush together about 18 in. before the struck car begins to accelerate sufficiently to affect the driver. Due to the car acceleration, the driver's back is then pressed, with increasing force, against the seat causing it to flex backwards until it more highly resists further bending (Fig. 14a). At this stage of seat flexure, the shoulders are supported against further rearward movement, but the head and neck continue to be forced rearward to develop the whiplash injury. (See Fig. 14b). An analogous, but corrected, situation is provided by the carrier-based military aircraft. Here, the pilot has been provided with a headrest that protects him from receiving whiplash injuries of the neck during catapult launching of his aircraft, and this head support has been designed so that it does not

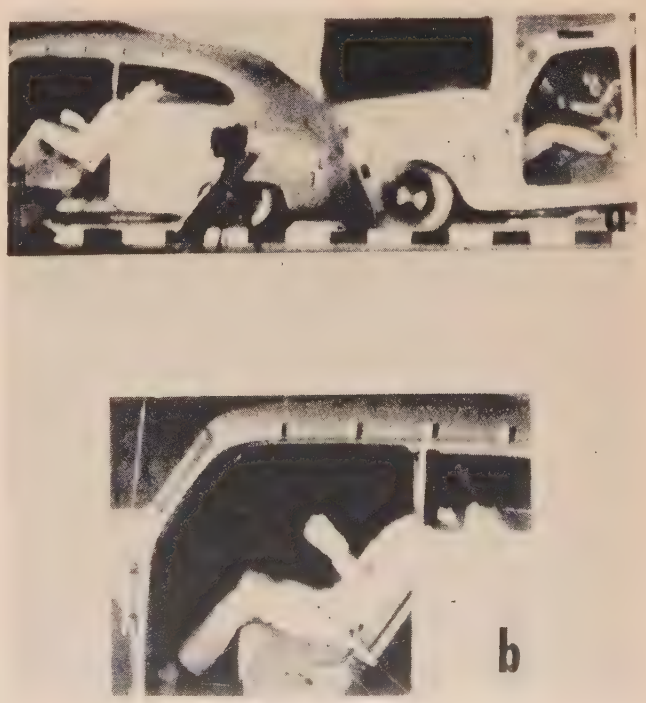


Fig. 14—a: rear-end collision at 125 millisec after contact; b: rear-end collision at 200 millisec after contact

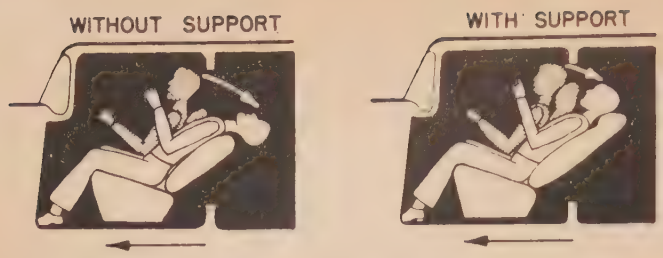


Fig. 15—Effect of head support

impair his vision to the rear and rear-sides of his aircraft. The protective function of a comparable head support for the motorist is illustrated by Fig. 15.

Basic Engineering Information from Collision Tests

Rear-end Collision—1. Velocity patterns: The changes in velocity with respect to time after the two cars come into contact for a 20-mph collision are given by the upper portion of Fig. 16. This graph is self-explanatory but it should be noted that following the instant of maximum car deformation (at 145 millisec) the front car still gains additional velocity, and the rear car continues to lose velocity due to the restitutional forces of the cars and this process requires 525 millisec before it is completed by separation of the cars.

2. Mutual deformation curves: Much of the collapsing of car structures which occurs during a collision is not evident following the collision. This fact is presented graphically in the lower portion of Fig. 16. As the cars crush together, a time is reached when the cars have the same velocity.[6] At this instant the cars have reached the point of maximum mutual deformation. This amounted to 24.4 in. for a 20-mph collision of which the permanent deformations for the front and rear cars were 5.2 and 6.1 in., respectively. This means that the mutual plastic deformation was 11.3 in. and the mutual elastic deformation, 13.1 in.

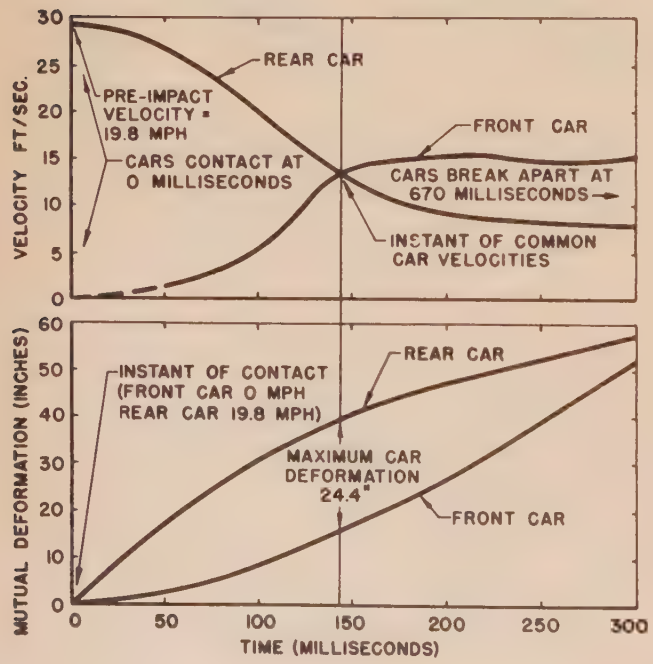


Fig. 16—Rear-end collision-velocity and mutual deformation patterns for front and rear cars

3. Permanent deformation patterns: Permanent deformations of both front and rear car frames are given for 7-mph, 9.9-mph and 19.8-mph impacts by Fig. 17. The procedure for obtaining these data is described in detail.[7] The 7-mph collision developed a permanent deformation of 1.2 in. for the front car, but no permanent deformation for the rear car. In contrast, the front car received 0.9 in. less permanent deformation for the 20-mph impact than did the rear car. For the 10- and 20-mph collisions, the rear of the front car did not deform uniformly. Adequate information for structural modification of the rear ends of cars to improve their *collision performance* during a rear-end collision cannot be properly derived from a single series of tests. This series of tests serves to identify the problem.

Data from the Automobile-Barrier Impact—1. The fixed-barrier collision: The research engineer, for economy purposes, attempts to reduce the system under analysis to a laboratory model where observations may be made under conditions which are more easily controlled. After considering a variety of laboratory models it was decided that none of them could be developed to duplicate accurately all of the possible reactions of the automobile under impact without exceeding the time and cost of full-scale investigation. Full-scale automobile-impact studies eliminate the unknown factors which appear when attempting to interpret the laboratory findings in terms of their application to the full-scale situation.

These considerations led to the decision that crashing test cars into a fixed barrier appeared to be the system which provided the best experimental control without deviation from the concept that the tests should duplicate the physical conditions common to a severe impact. An 8-ft high, 14-ft wide barrier was constructed of large diameter utility poles which were backed by suitable cross members and braces as well as by tons of dirt.

The barrier impact imposes a more rigorous test of the protective qualities of a motorist restraint than would generally be encountered in a car-to-car head-on collision. In the latter there would be mutual penetration and, consequently, a lower deceleration rate than would be the case with the car-barrier impact in which one of the two colliding objects is essentially nonyielding and nonpenetrable as shown in Fig. 18.

[6] For example, both cars have the same velocity 145 millisec after the cars contacted each other, for a 20-mph rear-end impact.

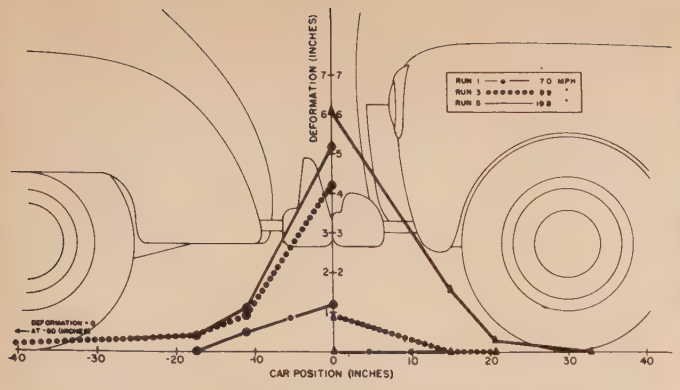


Fig. 17—Car-frame deformation pattern for the rear-end collision

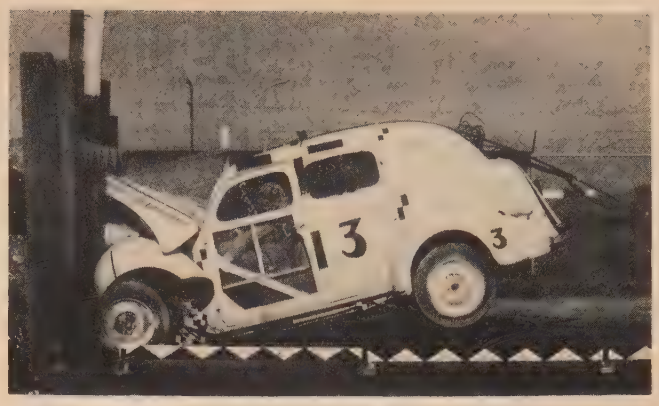


Fig. 18—The barrier collision

2. Frame deceleration: The mechanism of collapse of a structure during collision is of interest to the engineer. Knowing the order and magnitude of deformation of each section of the structure provides a basis for evaluating the total structure in terms of its efficiency for attenuating or moderating the collision forces and, therefore, the decelerations of the intact portions of the vehicle.

A system was developed by the Institute of Transportation and Traffic Engineering for instrumenting each 6-in. section of the front of the car frame back to a point about opposite the rear edge of the front wheel. It was necessary for this system to provide accurate instrumentation for a point on the structure obscured from view by the front wheel. This was accomplished by welding quarter-in. steel drill rods to the frame at these points in such a manner that they are nearly flat against the frame and point toward the rear of the car. A checkered target is painted onto the 3¼ x 3¼ x $\frac{1}{16}$-in. plate which is welded to the unsecured end of the rod.

The timed-displacement of these checkered flags as the car frame decelerates and deforms is photographed by high-speed cameras. (See Fig. 18, area immediately to the rear of the front wheel.) Since the metal flag is relatively weightless, and the rod was loaded essentially axial, the errors of measurement for this system may be expected to be low. However, a control for this instrumentation was introduced for the purpose of evaluating the error associated with deceleration flag measurements. A deceleration flag, hereafter referred to as the control flag (Target No. 10), was welded to the frame of the car about 1 ft in front of the door post and oriented so that the rod pointed back nearly flat against the frame. The rod was the same length as the longer units used in the front so that the error of measurement associated with this device would be representative of the error for the front devices. The checkered flag for this control unit also may be seen in Fig. 18 against the frame under the rear door. The deceleration pattern observed for this control flag as contrasted with the reference (Target No. 9), painted onto the frame immediately in front of the welded end of the flag is given by Fig. 19.

The peak g of the control flag differs by 11% from the peak g of the reference reading on the same point on the car structure. The area under these curves representing the total force-time history for that section of the car structure differs by less than 1%. The degree of accuracy of the family of deceleration curves obtained from seven deceleration flags mounted at 6-in. intervals along the front third left side of the frame has thus been established. (See Fig.

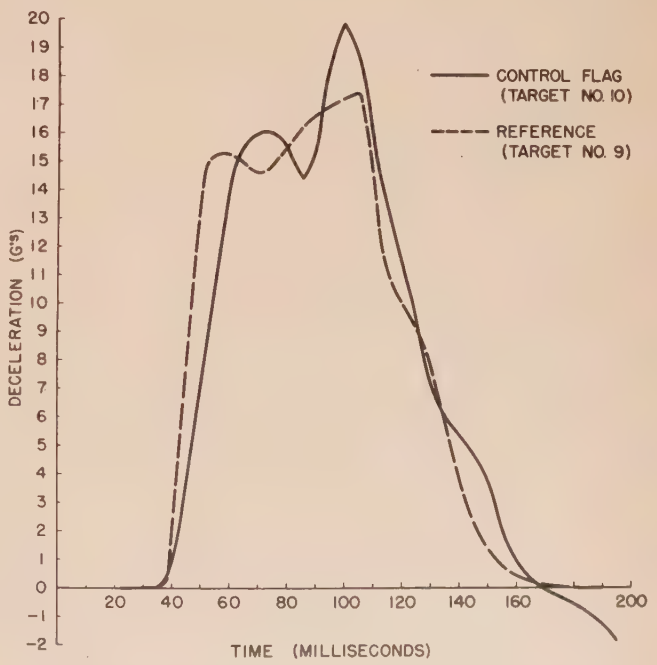


Fig. 19—Discrepancy between deceleration patterns for the control flag and the reference

[7] See Highway Research Board, National Academy of Sciences, Bulletin 91, January, 1954: "Automobile-Barrier Impacts," by D. M. Severy and J. H. Mathewson.

Table 6—Automobile Impact Analysis

Item No.	Description	Derived From	Car No. 1	Car No. 2	Car No. 3
1	Year and make	Data	1937 Plymouth	1937 Plymouth	1937 Plymouth
2	Body type	Data	4-door sedan	4-door sedan	4-door sedan
3	Measured weight with dummies	Data	3077	3380[a]	3240[a]
4	Velocity before impact	Data	24.9 mph (36.5 fps)	25.9 mph (38.0 fps)	29.3 mph (43.0 fps)
5	Velocity after impact	Data	−3.9 mph (−5.8 fps)	−3.1 mph (−4.6 fps)	−3.75 mph −5.5 fps
6	Deceleration rate (peak for car top)	Data	605. fps² (18.7g)	460. fps² (14.3g)	449. fps² (14.0g)
7	Duration of impact (time for > 1g)	Data	0.170 sec	0.166 sec	0.19 sec (approx.)
8	Amount of crushing of car, elastic plus plastic	Data	2.3 ft	2.4 ft	3.6 ft
9	Mass (w/g)	No. 3/g	95.6 slugs	105.0 slugs	100.6 slugs
10	Momentum before impact	No. 4 x No. 9	3490 lb-sec	3990 lb-sec	4330 lb-sec
11	Momentum after impact	No. 5 x No. 9	−554 lb-sec	−483 lb-sec	−553. lb-sec
12	Total change in velocity	No. 4—No. 5	42.3 fps	42.6 fps	48.5 fps
13	Kinetic energy before impact	½(No. 9)(No. 4)²	63,700 ft-lb	75,800 ft-lb	93,000 ft-lb
14	Kinetic energy after impact	½(No. 9)(No. 5)²	1610 ft-lb	1110 ft-lb	1520 ft-lb
15	Coefficient of restitution	No. 5/No. 4	0.16	0.12	0.13
16	Change in momentum	No. 10—No. 11	4040 lb-sec	4470 lb-sec	4880 lb-sec
17	Loss of K.E. during impact	No. 13—No. 14	62,100 ft-lb	74,700 ft-lb	91,500 ft-lb
18	Average force acting	No. 17/No. 7	23,800 lb	26,900 lb	25,700 lb
19	Average rate of energy dissipation	No. 18/No. 7	365,000 ft-lb per sec	450,000 ft-lb per sec	482,000 ft-lb per sec
20	Crash hp	No. 20(60) 33,000	664 hp	818 hp	876 hp

[a] These two cars each carried an additional dummy weighing 200 lb which accounts for their marked weight difference over car No. 1.

20.) The specific values of peak deceleration for these curves will, of course, vary with the parameters of the collision such as, the speed, type vehicle, the object struck, and similar factors.

In addition to the value of providing the design engineer with information relating to the problem of collision performance for the front third of the car, Fig. 20 serves to indicate the kind of information on collision performance for front-end structures which may be derived from a relatively simple instrumentation technique. It is recognized that following the development of this mechanical-photographic instrumentation technique at U.C.L.A., accelerometers capable of operating under severe impact conditions have been made commercially available. When the proper accessory equipment can be obtained, use of these instruments, although considerably more expensive, is recommended.

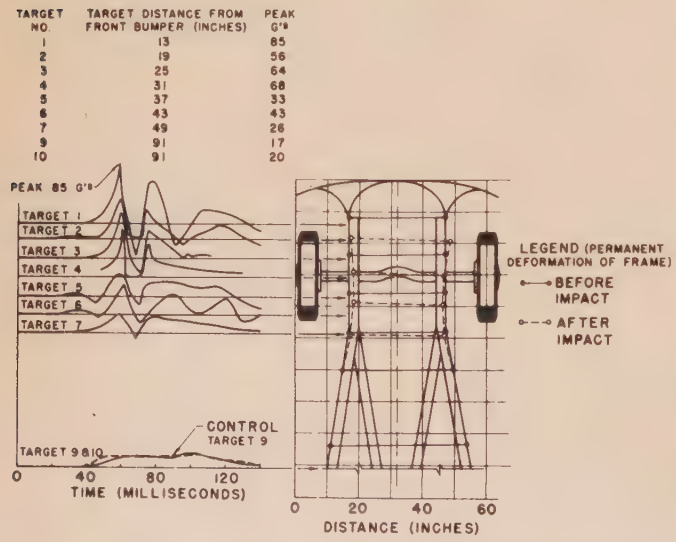


Fig. 20—Profile of deceleration during collision with a fixed object, 25 mph

3. Frame deformation: One method of obtaining useful engineering information from a collision is to measure specific car-frame positions before and after impact in order to evaluate the location and magnitude of collision deformations. Positions on each side of the car frame were marked with application of metal screws at points approximately 1, 2, 3, 4, 5, and 7 ft back from the front edge of the bumper. The distances between these points were measured by projecting them onto a horizontal plane both before and following the collision. Details of this procedure have been previously studied.[7] The permanent deformations of the frame for one of the 25-mph barrier collisions are also provided by Fig. 20 (right portion)

Three cars of the same make, model, and year were crashed, independently, into the same rigid barrier at approximately the same speeds. Cars 2 and 3 may differ slightly from car 1 because in the case of the latter, the two front seats were mounted on a pair of specially instrumented structural steel 4 x 8 I-beams. The gross weight was 3077 lb for one car[7] and 3240[8] and 3380[8] lb for the two other cars.[9] A comparison of the deformation patterns for these three nearly identical vehicles crashed under essentially the same conditions is given by Fig. 21. Reasonable correlation exists between cars 2 and 3. The removal of the two left side doors and the alteration of the weight and balance of car 1 may account for

[8] These two cars carried an additional dummy weighing 200 lb, which accounts for their marked weight difference.

[9] See "Automobile-Barrier Impacts, Series II," by D. M. Severy and J. H. Mathewson. Presented at SAE San Diego Section, January, 1955.

Table 7—Coefficients of Restitution for Automobile Collisions

	Impact Situation	Coefficient of Restitution
1935 Oldsmobile 8 4-Door Sedan and 1936 Buick "60" 4-Door Sedan	Head-on collision, closing velocity 60 mph	0.102
1936 Buick "40" 4-Door Sedan and 1937 Dodge 4-Door Sedan	Head-on collision, closing velocity 74 mph	0.006
1937 Plymouth 4-Door Sedan	Collision at 25 mph with flat fixed barrier	0.16
1937 Plymouth 4-Door Sedan	Collision at 26 mph with flat fixed barrier	0.12
1937 Plymouth 4-Door Sedan	Collision at 29 mph with flat fixed barrier	0.13
1947 Plymouth 2-Door Sedan and 1941 Plymouth 4-Door Sedan	Rear-end collision with the 1941 Plymouth at 20 mph, striking the 1947 Plymouth when stationary	0.29

its significantly different deformation pattern. Although only cars were selected which appeared not to have been involved in an accident, their history of use and abuse may also have a bearing on this divergence in correlation.

4. Automobile impact analysis: A comparison has been made of the physical factors of three collisions with a fixed barrier by practically indentical cars crashed under essentially the same conditions. This data is presented in Table 6 in a form which is self-explanatory.

5. Coefficients of restitution for automobile collisions: In the past, the coefficient of restitution for the crashing automobile was estimated as being nearly zero. Table 7 presents the coefficients from some of the U.C.L.A. collision tests. These values show that the coefficients may vary for the same accident type and can be expected to vary for different accident types with a range from near complete plasticity to partially elastic reactions.

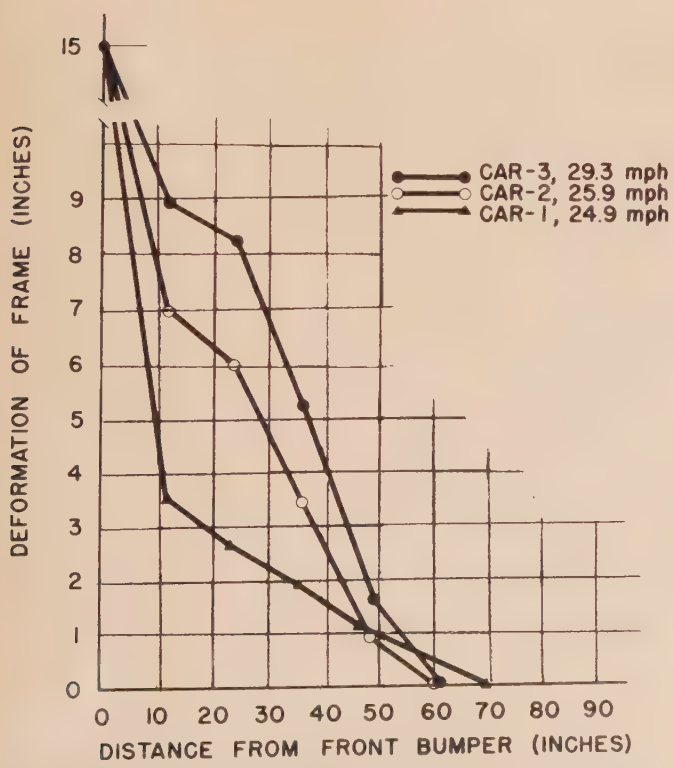


Fig. 21—Frame deformation patterns for barrier collisions

ORAL DISCUSSION

Reported by J. D. Leslie

General Motors Corp.

E. C. DeSmet, Willys Motor Corp.: Are the present design trends which point toward the elimination of roof-supporting structures (pillarless bodies) in passenger cars and forward control commercial vehicles in conflict with improved crash safety?

Mr. Severy: No experiments have been conducted by U.C.L.A. on this subject. Evaluation of the roll-over type of accident is worthy of attention but it is not as important as front and side impacts, the cause of the greater majority of fatalities.

H. C. MacDonald, Ford Motor Co.: Could the bumpers be designed to absorb more crash energy?

Mr. Severy: The effect of some bumpers becomes negligible in many crashes occurring at speeds greater than 10 mph. There is strong possibility of improvement along this line.

A. L. Haynes, Ford Mortor Co.: What method was used in correlating dummy injuries with those of human beings?

Mr. Severy: Abrasions and punctures in the dummies can be translated to the same type of injury in human beings. There is, however, considerable variation in the strength of human bodies. The variation in bone strength for human bodies of comparable stature has been determined to be as great as 1/12. The use of animals as crash subjects is being studied as a means for providing better correlation between dummies and human beings.

P. Skeels, General Motors Proving Grounds: To what extent might the use of a shoulder harness or chest belt cause head rebound or whiplash?

Mr. Severy: In a barrier collision where the head is thrown forward, it often oscillates through 1½ cycles before dampening down. It is conceivable that with an upper torso restraint there could be considerable head whiplash effect, but the resulting injury would be relatively minor compared with an unrestrained upper torso.

C. S. Wilkinson, Goodyear Tire and Rubber Co.: Have experimental results proven the effectiveness of the combination of lap belt and crash pad on the instrument panel?

Mr. Severy: The instrument panel crash pad has a definite limitation in that it does not conform to the contours of the face and under impact tends to alter those contours. A better approach would be to eliminate the horizontal surface of the instrument panel and recess the instruments forward of the lower edge of the windshield so as to eliminate any forward surface against which the face might be driven in a crash.

F. C. Matthaei, American Metal Products: What changes could be recommended in seat structures to improve crash impact strength?

Mr. Severy: The anchorage to the floor of the seat adjustment mechanism must be made very rugged if it is to withstand crash impact forces. A seat assembly weighing 100 lb would be subjected to a 2000-lb force in a 20-*g* deceleration considering only the effect of its own weight.

Looking to the future, it would be desirable to design the seat and seat back unit to withstand the forces exerted by upper torso restraining devices such as a snap-on shoulder loop belt.

At the present time, there is still some public apathy toward the use of the lap belt—although actually it is little different than the seat belt which the public readily accepts on airliners. The difference in attitude is a matter of custom and viewpoint. Drivers should recognize the fact that when they are sitting in their car, they are in a position more vulnerable to injury than when sitting with comparable comfort in a chair at home.

POWERMATIC

A New Automatic for Chevrolet

POWERMATIC, an all new automatic transmission, shown in Fig. 1, was developed in cooperation with the Allison Division for the 1956 Chevrolet medium- and heavy-duty trucks. The time and energy expended in such an undertaking may prompt a familiar question: why undertake development of an entirely new unit when others exist, within the corporation itself, which could be reasonably adapted to our use?

THIS paper describes the development of a truck automatic transmission, from a statement of broad objectives through the growing pains, to road testing of the final product. Emphasis is placed upon original thinking that led to the decision to undertake such a project, compromises that suggested themselves throughout the various stages, and features tried and found wanting as well as those retained.

The finished product is described full though not in detail, stress being placed upon interesting and novel design features.

Part of the answer is basic GM policy. Each division is in open competition with the other divisions. The spirit of rivalry is deliberately promoted to be as fierce and unrelenting as it is with outside companies.

With this policy goes the freedom to compete. Each Division manager is commissioned to solve his own market problems. With the aid of his chief engineer, he decides whether an existing design should be adapted to his needs or a completely new component developed. In this case, existing units were surveyed and various adaptations that suggest themselves were seriously considered. None satisfied all our particular needs and objectives.

Some three years ago we expanded our truck engineering group considerably. Most of the new men were truck engineers, but a select few were experienced primarily in passenger cars. One of their first requirements was to become qualified truck drivers.

They had not witnessed first hand the measures that had been taken over the years to coax more performance out of trucks.

What they did notice was that they had only two hands, two feet, two eyes, and one brain to operate all the gismos and gimmicks required to keep the

Transmission Heavy-Duty Trucks

H. O. Flynn, General Motors Corp.

This paper was presented at the SAE Annual Meeting, Detroit, Jan. 11, 1956.

truck rolling. One man suggested an optional co-pilot! Some of their criticism was more pointed. This group steadfastly refused to believe a truck must necessarily be a clanking, smoking monster.

To sum up, we felt we had the incentive, the time, and the talent to make a worthwhile contribution to the trucking industry.

Truck driving today, undeniably, does require a rather high degree of know-how as well as much tedious work. Fleet owners report that it has reached

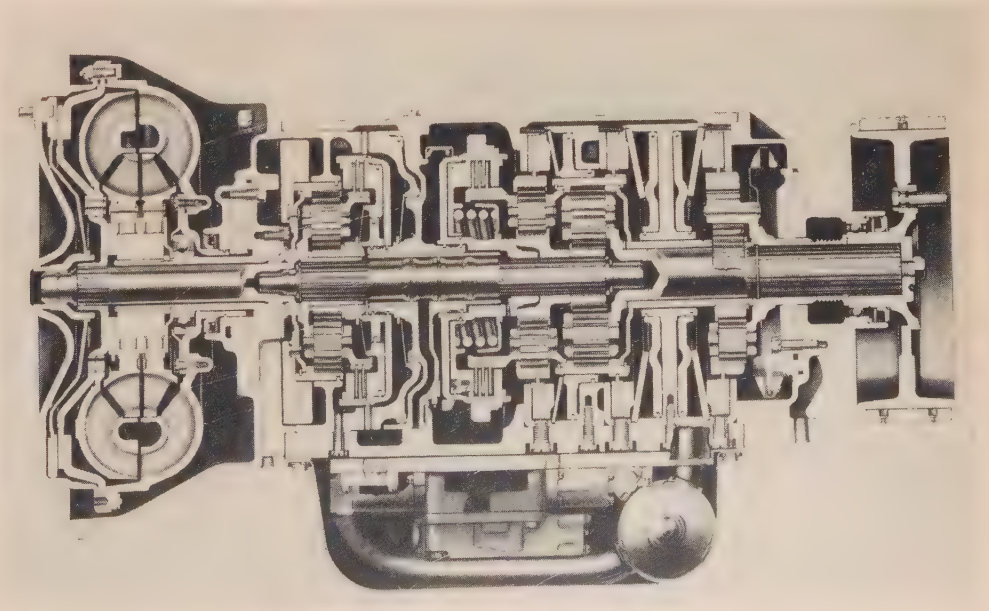


Fig. 1—Sectional view of Powermatic transmission

the point where the actual worth of their vehicles is largely determined by the driver's ability and willingness to use all the available performance aids all the time.

When evaluating the merit of any device purporting to simplify driving, the trucker is now ready to accept the possibility that it might also increase the earning capacity of his vehicle. Other benefits that may accrue, not directly related to the cost per ton-mile, stimulate less owner interest.

In outlining the requirements for our heavy-duty trucks, the factors that would contribute to their earning capacity were given first priority. We sought principally to reduce the skill-requirement, fuel bill, portal-to-portal time, and down time.

The following objectives were drawn up as the goal during design and development stages of the Powermatic.

1. Increased vehicle mobility and flexibility under all conditions by—

Fig. 2—First design phase

Fig. 3—First design phase finished product

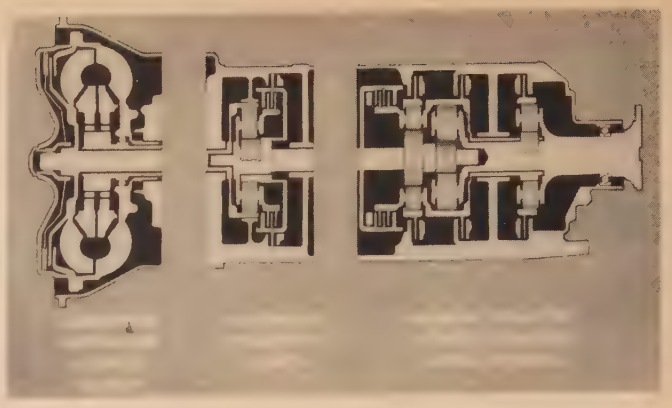


Fig. 4—Second design phase

(a) Applying engine horsepower effectively through well-chosen geared ratios.

(b) Full power shifts up and down to any gear ratio.

(c) Full automatic shifting with driver control of specific ratio ranges for specific operating conditions.

(d) Controlled downhill braking.

2. Improved vehicle fuel economy by—

(a) Low parasitic losses in transmission.

(b) Allowing the engine to operate in its best speed range for good economy.

(c) Allowing use of reduced N/V axle ratio for part throttle economy.

3. Improved vehicle durability and reliability by—

(a) Reduced clutch, drive line and axle maintenance.

(b) Eliminating band-type control elements.

(c) Designing the transmission in one unit with simple controls.

(d) Having no adjustments in the transmission.

(e) Including an oil filter in the transmission.

4. Improved vehicle adaptability and usability by—

(a) Providing SAE power take-off openings on both sides of the transmission.

(b) Using the torque converter for all starts forward and reverse.

For maximum installation simplicity and adaptability it is quite apparent that a single package transmission is highly desirable. This also makes for less bulk and weight and greatly simplifies the control problem. SAE standard power take-off openings on each side of the unit would be a welcome addition. All control valves and the valve body should be located where they are readily available for inspection and service. Since dirt is the greatest enemy of hydraulically controlled transmissions, a large replaceable filter unit is needed.

Further in keeping with ease of installation and general simplicity, the transmission shift control quadrant and the hand brake lever for the transmission mounted parking brake should be mounted directly on the transmission.

As previous experience has shown, it is most desi-

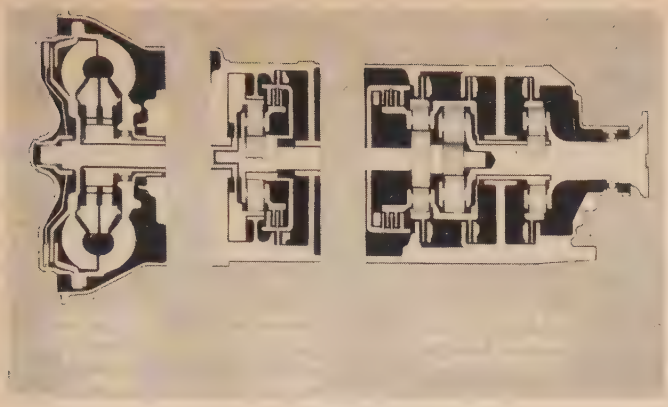


Fig. 5—Third design phase

rable to eliminate any and all adjustments of the friction elements in a hydraulically controlled transmission. In place of adjustable friction elements to time shifts, the control system should be so arranged that this entire timing function is performed hydraulically, eliminating periodic adjustments to compensate for wear.

While all these characteristics were certainly to be desired in the finished product, we felt that the economic feasibility of including them all, or choosing judiciously between them, could be most advantageously explored by trying the simplest solution first.

For purposes of illustration we have divided the early development program into four broad phases. Actually, of course, the development was typically continuous and had no clear dividing lines.

The first design attempted (Fig. 2) was, for economic reasons, kept as simple and small as possible. It employed simple planetary sets to give low, intermediate, direct, and reverse drives. In series with the gearbox was a single-stage four-element converter with high stall ratio. The unit was power shifted through a manual control.

Several units were built and tested extensively. (See Fig. 3.) The test program clearly established that this type design would not provide performance features we believed necessary for our trucks. But it also demonstrated something else: remarkable durability of planetary gears, clutches, converter, and pumps.

The first phase of our design study can be summarized as follows:

Converter
Three speeds forward, one reverse
Manual shifts
No lockup
Air cooling

Two major changes were incorporated to overcome the deficiencies of the first design. First, an automatically actuated lockup clutch was added to minimize converter drive losses over extended operation at or near wide open throttle conditions. Lockup also provided better engine braking. The second feature was an automatic shifting planetary "splitter" unit in front of the main gear set (Fig. 4).

In this design we had converter starting, automatic throttle-controlled upshifts and downshifts at the splitter unit, a lockup clutch, and three manually selected forward speed ranges. Gear ratios gave a uniform geometric pattern from direct to 5.29/1. Several of these units were subjected to continuous testing to demonstrate durability of the splitter gear set, clutches, and lockup clutch. Performance and economy the unit made possible left no question that we were on the right track.

Phase 2 can be summarized as:

Converter and lockup clutch
Three gear ratios plus splitter
Semiautomatic control
Liquid cooling

The next significant step in the development was an all-out effort to improve downhill braking control. This resulted in the addition of what we call a hydraulic retarder, Fig. 5, an independent, dynamic braking device.

Phase 3 can be summarized as:

Converter and lockup clutch
Three gear ratios plus splitter
Semiautomatic control
Liquid cooling
Hydraulic retarder

The finished product (Fig. 6) incorporated completely automatic controls, three specialized drive ranges, and driver overcontrol for unusual driving situations.

The finished product incorporated:

Converter and lockup clutch
Three gear ratios plus splitter
Automatic controls
Hydraulic retarder
Election of speed ranges by the driver

Wide variations in loading and routes peculiar to truck operation complicate the control problem. However, our aim was not simply to supply ratios

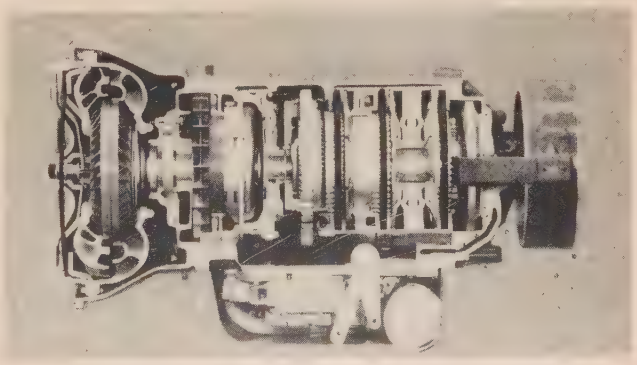


Fig. 6—Final design of Powermatic

that read well on a performance table, but to make certain the right ratio is automatically sent into action at the right time. The control system is no more complex than is required to accomplish that purpose.

With total ratio of up to 7.5/1 available for starting, drive range can be used for practically all driving conditions. The range includes the top four gear ratios plus the converter for starting (Fig. 7). All starts in this range are made in third gear; 2.691/1 ratio times converter ratio which at stall is 2.8/1.

The normal shift schedule is: lockup clutch engages at 2000–2600 engine rpm, depending upon throttle opening. Shift from third to fourth gear is made at 2600–3800 engine rpm, depending upon throttle opening. With light throttle settings the next shift is from fourth to sixth gear. At wide open throttle, the shift is from fourth to fifth, and then to sixth gear, both shifts coming at 3800 engine rpm.

At 12–30 mph in sixth gear a slight throttle pressure will force a downshift to fifth gear. Forcing the throttle to detent will give a downshift to fourth gear. Forcing the throttle through the detent will give a downshift to third or fourth gear for maximum performance, depending upon vehicle speed.

At road speeds of 30–50 mph in sixth gear, forcing the throttle through detent will give a downshift to fifth gear.

All wide open throttle upshifts in drive range are made at 3800 engine rpm. Downshifts occur at 2400 engine rpm.

Low range, which includes converter and first, first, and second gear is a very specialized range used for maximum tractive effort in moving heavy loads on extreme grades, maneuvering at low vehicle speed where rolling resistance is great, and other like situations where utmost tractive effort is required (Fig. 8).

Normal operation includes all starts in first gear and converter with the converter lockup clutch engaging at 1800–2400 engine rpm, depending upon throttle opening. Shift from first to second gear occurs at 2600–3800 engine rpm, again depending upon throttle opening.

Intermediate range restricts operation to converter and third, third gear, and fourth gear. All starts are made in converter as in drive range.

This range is most important for use where additional vehicle control is desired at low speeds by permitting higher engine speeds for acceleration or deceleration. We have found this range very useful in driving in heavy traffic with a heavily loaded rig. It is also very effective in combination with the hydraulic retarder (Fig. 9).

There is no provision to allow an automatic upshift out of intermediate under any engine overspeed situa-

Fig. 7—Use of Powermatic in drive range

Fig. 8—Use of Powermatic in low range

Fig. 9—Use of Powermatic in intermediate range

tion. As such it gives the driver absolute overcontrol on upshifting.

To acquaint you more thoroughly with design details of individual components we will start at the front of the transmission and move rearward.

Converter

The converter, an all-aluminum casting, is a four-element, single-stage, three phase type with two overrunning stators (Fig. 10). Torque multiplication is 2.8/1 at stall. The seal between converter and front pump is a Teflon material, chosen for its ability to withstand high temperatures, and for its low coefficient of friction.

The converter is not required to function efficiently as a direct drive coupling, so a large share of the usual compromise in blade design is avoided.

Design freedom resulting from the lockup arrangement permits blade configuration to give a relatively high, efficient multiplication factor during the multiplication stage, comparable to that ordinarily attained in considerably milder converters (Fig. 11).

Lockup ordinarily occurs just past the fluid coupling point. When conditions favor delaying lockup, such as when maximum performance is desired, the driver may do so by forcing the accelerator through the detent position.

Fig. 10—Converter elements

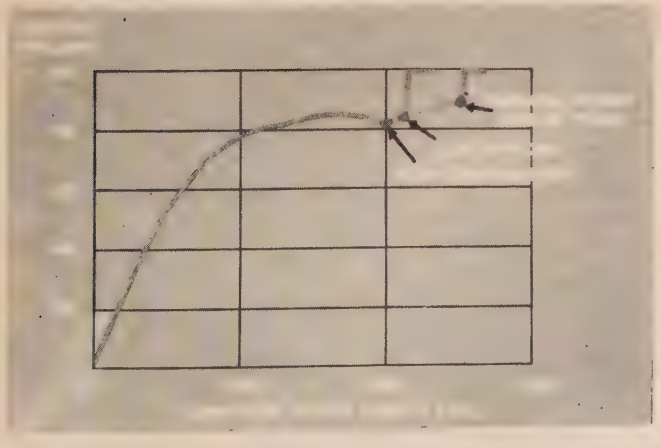


Fig. 11—Converter efficiency

So that best possible fuel economy would be realized over the most used driving range, a hydraulically actuated lockup clutch is arranged in parallel with the converter (Figs. 12 and 13).

The driven plate hub is splined to the transmission input shaft and carries the converter turbine. Nonmetallic friction faces are molded into a single biscuit with internal driving teeth.

The reaction plate is carried by bolts connecting the converter cover to the converter pump housing assembly, and the pressure plate is driven with the converter cover through dowels.

The pressure plate is shrunk on its own diaphragm return spring. These two pieces constitute the clutch apply piston. A ring on the pressure plate seals the outside diameter, and the spring forms its own seal against a ring on the cover assembly hub. Positive clutch control is provided by the following pressure schedule:

When operating in converter, charging pressure is

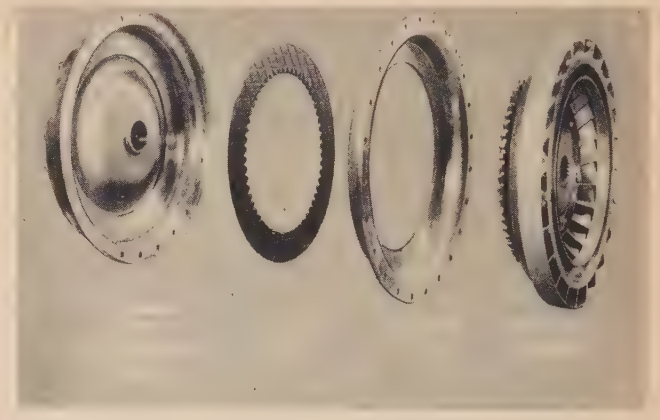


Fig. 12—Lockup components

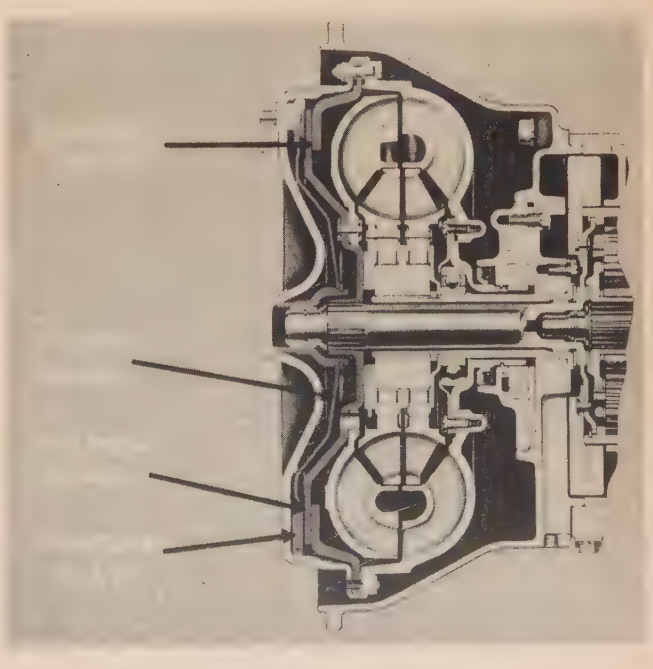


Fig. 13—Lockup clutch

maintained at a sufficiently high level to eliminate cavitation and to provide a circulation rate through the converter to satisfy oil temperature requirements. Charging pressure plus spring force holds the clutch out of engagement during this phase.

Concurrent with lockup engagement, charging pressure is lowered, and apply pressure is directed to the forward side of the pressure plate. Raising and lowering converter charging pressure alternately with lockup clutch apply pressure assures positive and accurate control of this clutch under all conditions.

Oil Pumps

The front pump (Fig. 14) is a conventional internal-external spur gear type selected because of its ease of manufacture and reliability. Large displacement is necessary to supply oil volume needed for hydraulic retarder operation.

The rear pump is also a positive displacement gear type, but has only half the displacement of the front pump. In addition to supplying pressurized oil for push starting, the rear pump maintains main line pressure for the control system whenever pump speed is high enough. The front pump continues to handle volume requirements when they arise.

Hydraulic Retarder

Improved vehicle mobility was one of our primary objectives. In highway truck operation, the ability to maintain high average road speeds depends not only upon the rate at which a truck can be brought up to speed, but also the rate at which it can safely be slowed down. Ease and accuracy with which deceleration can be controlled affects overall performance even on comparatively flat roads.

This portion of the performance factor is, of course, magnified in proportion to the number and size of grades to be negotiated. With the gear ratios established, we predicted the effect of engine braking and later confirmed the prediction by actual test. The results fell short of our objectives.

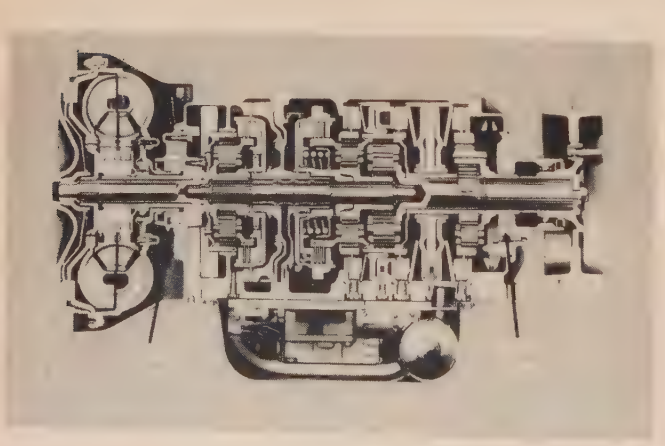


Fig. 14—Front and rear oil pumps

Fig. 15—Hydraulic retarder

Fig. 16—Hydraulic retarder components

Since hydraulic fluid under pressure was available, it appeared that an integral hydraulic retarder would be a natural. Further study indicated that by locating the retarder properly in the transmission we could use the transmission gearing to give added effectiveness and the maximum possible range of vehicle braking (Fig. 15).

The retarder (Fig. 16) functions, in effect, as a double-circuit fluid coupling with the output element grounded. Thus we have effectively a fluid coupling operating at stall. The impeller is attached to the splitter set ring gear. The grounded element is cast as part of the case in the rear and part of the converter ball housing at the front. This vaned cavity in the case is open to the retarder control valve at the lower right hand side.

A foot-operated pedal left of the steering column controls retarder control valve movement through direct linkage. When the pedal is operated, an exhaust port is immediately blocked and oil from the pumps allowed to pass through the control valve to the cavity. Oil circulates through the cavity, absorbs energy in the form of heat, passes back through the retarder control valve body at a controlled rate, and then

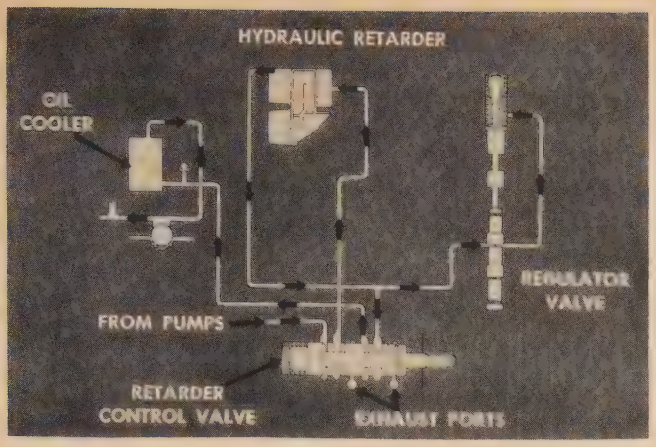


Fig. 17—Hydraulic retarder control circuit

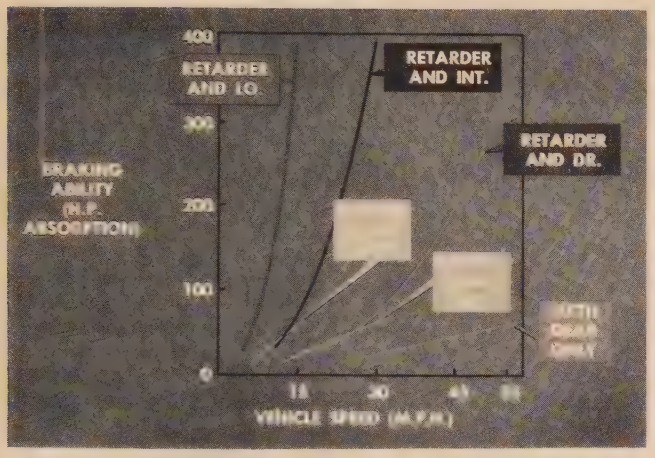


Fig. 18—Hydraulic retarder effectiveness

directed to an external oil cooler. The valve is shown in a partially applied position in Fig. 17.

Oil from the retarder is sent to the main regulator valve to increase holding force of the clutches to meet increased need.

When the pedal is released, the control valve returns to its closed position, main oil supply is cut off, and centrifugal pump effect of the impeller forces remaining oil out of the cavity into the sump.

"Feel" of the retarder differs from conventional brake pedal operation because cavity filling is not instantaneous.

After becoming accustomed to the approximate one second fill time, one soon knows what to expect and uses the control accordingly. Gradual fill eliminates the possibility of putting severe shock loads on the drive line and axle during application.

As seen from the horsepower absorption-vehicle speed curves (Fig. 18), the retarder gives three ranges of retarding which, at higher engine rpm is in the region of 300–400 hp. The exponential relationship of the horsepower-speed curve gives very effective retarding down grades and in situations where it is necessary to use the retarder constantly. However, our experience has shown that there are very few grade situations where retarding is required for prolonged periods.

We feel certain that the operational value of this new transmission is enhanced considerably because of incorporation of the hydraulic retarder. Certainly if our experience of doubled brake life is encountered by the user, he will feel immensely gratified.

A plate-type heat exchanger is located in the bottom tank of the engine radiator (Fig. 19) and utilizes the engine cooling system to maintain satisfactory oil temperatures. Our tests have shown that requirements for the engine cooling system were very nearly the same, whether ascending grade at full throttle, or descending using the hydraulic retarder.

Governor

The brain of any robot must receive clear, accurate signals in order to energize the control circuts that set up the behavior pattern. In any multiple-shift automatic transmission, the governor or speed-sensing device assumes major importance. (See Fig. 20.)

Governors used in Powermatic employ the familiar pitot tube in a new role.

The front governor, by translating turbine shaft speed in terms of oil pressure, determines the lockup point and influences the 1–2 shift point. All subse-

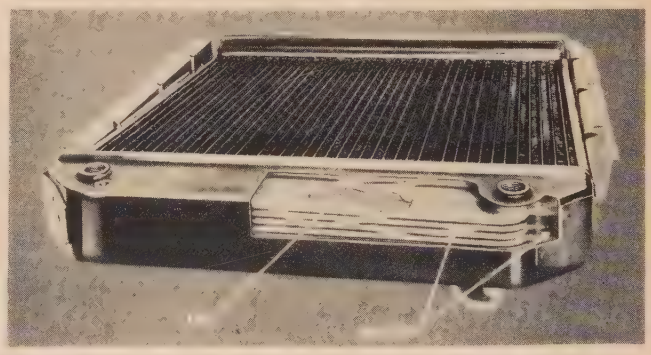


Fig. 19—Oil cooler

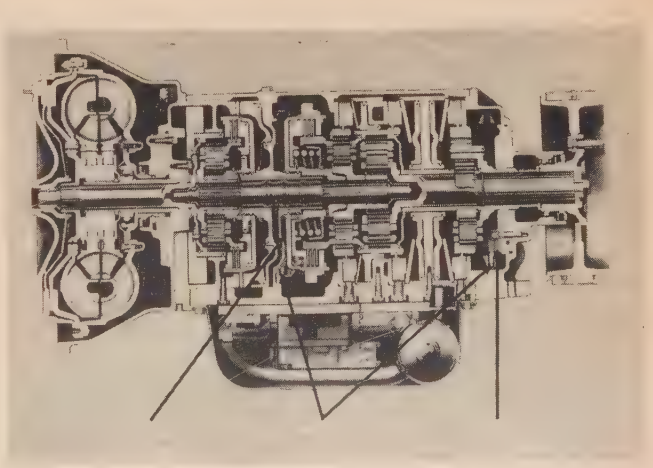


Fig. 20—Governors

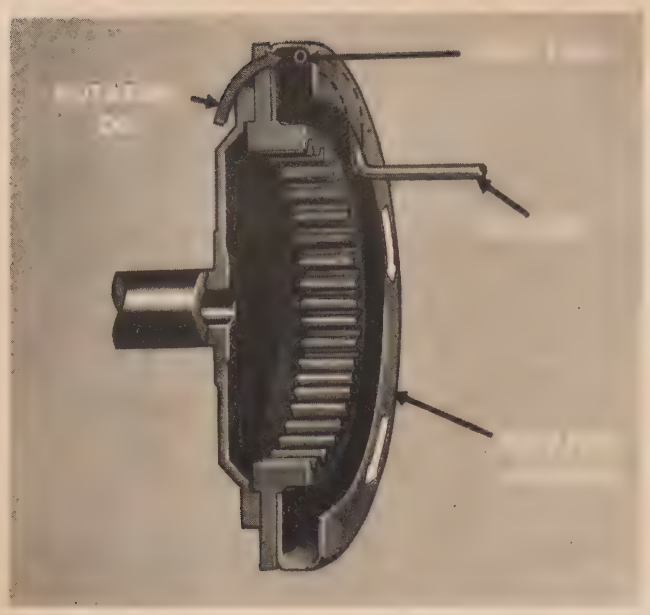


Fig. 21—Pitot velocity head governor

Table 1—Gear Ratios

Gear	In Reduction	Ratio
First	Splitter and low	5.294/1
Second	Low only	3.810/1
Third	Splitter and intermediate	2.691/1
Fourth	Intermediate only	1.936/1
Fifth	Splitter only	1.390/1
Sixth	None	1.000/1
Reverse	Splitter and Reverse	6.042/1

quent shifts are influenced by the rear governor sensing output shaft speed.

Pitot tubes measure fluid flow.

In each governor, a vaned oil-collector ring is attached to a rotating member whose speed may be measured hydraulically. No amplification is needed. Oil entering the tube acts directly on the control valves involved.

As the oil-collector ring rotates, the vanes cause oil to rotate with it (Fig. 21). Since the pitot tube is stationary, oil velocity at the point of contact with the tube is reduced to zero. This causes a pressure which can be found from the $v = \sqrt{2\,gh}$ relationship. In addition to velocity pressure, the tube also measures centrifugal pressure caused by weight of the rotating ring of oil. This pressure is also calculable.

The tube measures the sum of these two pressures, both of which vary with the square of the speed of rotation.

Accuracy depends first upon keeping oil level constant. A bleed hole from the oil pumps feeds an adequate oil supply to keep the collector ring full, and the open ID of the ring limits the top level the oil can reach.

The photo of the rear governor (Fig. 22) shows the simplicity of manufacture.

The pitot is a small steel tube welded to a flat plate which, in turn, is fastened by two screws to the rear pump body.

The collector ring stamping has a series of vanes welded to its inner surface. These few parts constitute the entire governor. The only machining required is the drilled passage in the pitot tube bracket. Adding to the ease of manufacture is the fact that established tolerances affect control variation by only 0.5%.

Gear Train

The gear train (Fig. 23) consists of four simple, rugged, spur gear planetary sets. Series arrangement of the planetaries allows the first set to become a "splitter" for the second and third forward sets and the fourth or reverse set. Employing the "splitter" principle makes it possible to provide six usable forward gear speeds following a geometric pattern with approximately 39% steps.

The splitter gear ratio is 1.39/1 in reduction and is used in combination with the 3.81/1 "low" gear set to produce a 5.29/1 reduction in first gear and with the 1.94 1 "intermediate" set to make a 2.69/1 third gear, while the splitter alone in reduction is the fifth gear. To complete the six speed pattern, the splitter is locked in "direct" and the basic low and intermediate gear sets provide second and fourth gears as shown in Table 1.

Clutches

Clutch development for the heavy-duty, full-torque shifting automatic transmission commands top priority from the very conception of the program. In this particular case, shifts must have acceptable smoothness in an empty truck at about 6000 lb curb weight and yet be durable enough to handle working loads through the 50,000 lb gcw range.

So that side thrusts on shafts, gears, and housing would not be introduced into an otherwise coaxial design, use of bands with attendant linkages, anchors,

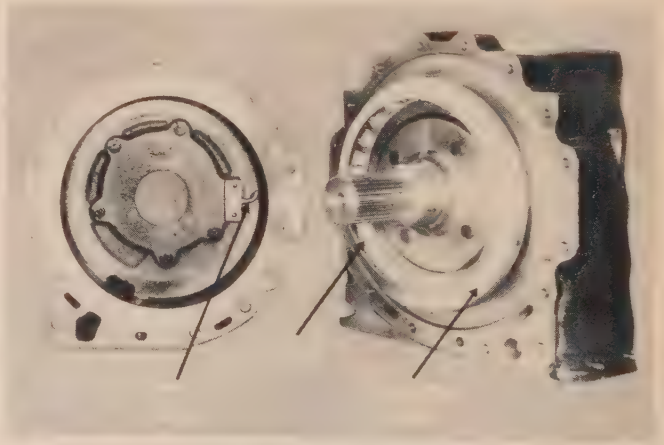


Fig. 22—Rear governor

Fig. 23—Gear train

and servos was dismissed in early considerations. The decision to go to an all-clutch design proved twice-blessed when the hydraulic retarder entered the picture. Because clutches have equal holding capacity in either direction, the retarder brought no new clutching problems. Consequently all clutches are plate type, either single or multiplate, and are either "direct acting" or "polar lever" applied. (See Fig. 24.)

Direct-acting clutches include lockup, splitter high, splitter low, and high clutches, all of which carry much lower maximum torque load than is required of the polar lever applied clutches.

It is interesting to note that of the four directly applied clutches, three distinctly different types of clutch plate material are used:

Type one: the lockup clutch which must absorb full engine torque, is a rigidly molded, single-plate biscuit incorporating a synthetic rubber augmenting agent.

Type two: the splitter-low clutch is a fully metallic single plate with sintered facings and steel core.

Type three: the splitter-high and high-gear clutches, is constructed of newly developed steel-backed, nonmetallic plates.

The four direct-acting clutches differ in construction details as well as material. Each is designed to meet specific requirements placed upon it. Splitter clutches, for instance, must absorb only about 39% of turbine shaft torque, while the high-gear clutch must absorb up to 1.025 times turbine torque. However, splitter clutch plates must have adequate holding capacity and low peak torque characteristics for smooth shifts. Splitter clutches must also have a high coefficient of heat transfer, or "*k*" factor, to withstand the repetitive application cycle with little time interval for cooling.

To meet these requirements, the fully metallic single plate was chosen for the splitter-low clutch where slip speeds are highest and heat conduction area greatest. This single plate, having a mean radius of 4.6 in. and area of about 20 sq in., had better working conditions than its companion clutch, the splitter-high.

The high-gear clutch has three nonmetallic plates because it must handle high torque loads with less slip speed on engagement than the splitter clutches.

In all clutch construction considerations, judicious use was made of the knowledge that well-directed lubrication can help dissipate heat generated during clutch engagements since oil has approximately five times more heat dissipating capacity than metal parts.

Through use of the belleville polar lever (Fig. 25), one clutch plate can do the work of many. In the case of the intermediate clutch, which carries the major share of the work, (being the starting clutch in drive range), the belleville spring multiplies piston apply force 3.25 times. When releasing its stored energy on disengagement, a 360 lb force aids in giving fast, clean breakaway. Low and reverse clutches have a 5:1 lever advantage to provide needed capacity for maximum torque requirements.

The outstanding advantage of this type clutch application lies in the fact that a moderate hydraulic line pressure can be used to produce high-torque holding capacity in a clutch with a minimum of active friction members. For example, in the intermediate clutch, an 80 psi line pressure produces a clutch force of 7600 lb. In a direct-apply clutch, the same

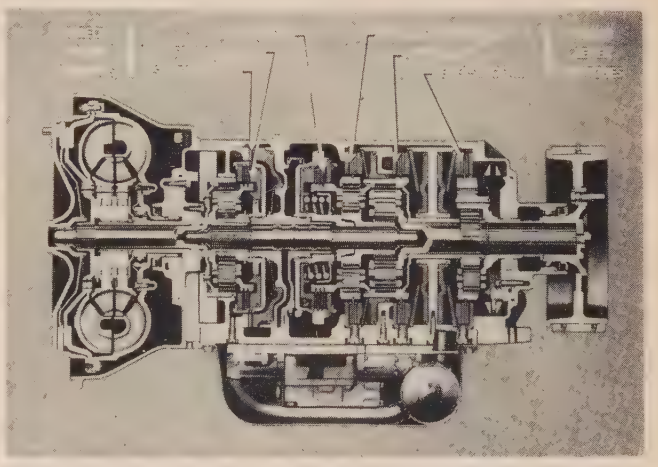


Fig. 24—Range clutches

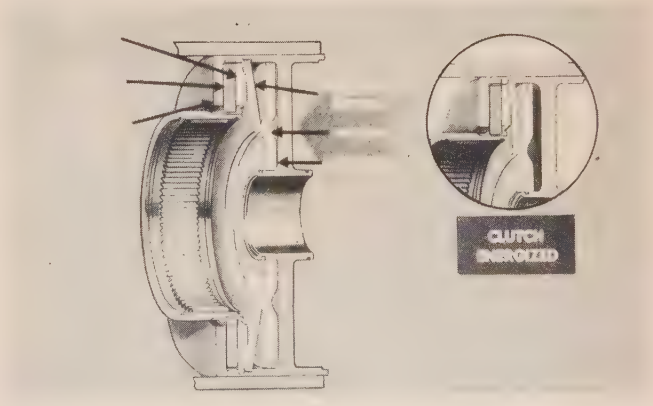


Fig. 25—Range clutch

Fig. 26—Reverse clutch

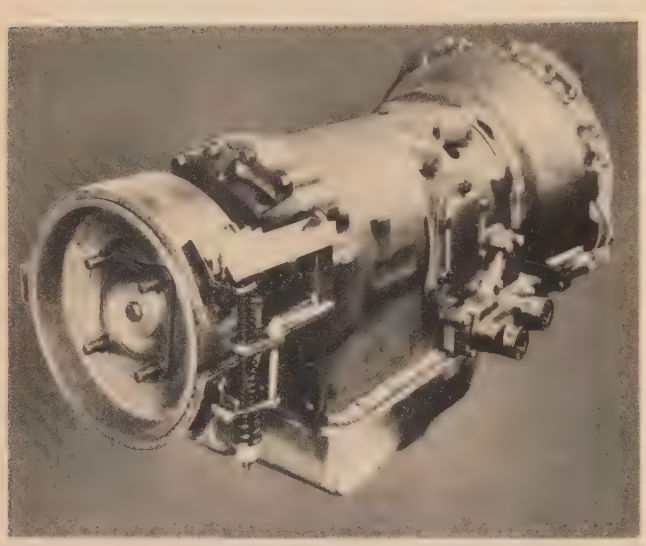


Fig. 27—Parking brake

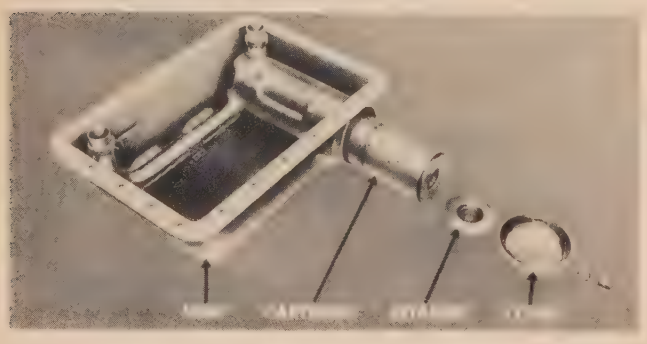


Fig. 28—Oil filter

piston area would require a line pressure of 200 psi or a piston of about three times the area for the same line pressure.

In addition to complete compatibility with coaxial transmission design, the belleville-applied clutch provides a fully controlled smooth engagement with a built-in accumulator. Since hydraulically applied clutches automatically compensate for wear, no mechanical adjustments are necessary.

The reverse clutch (Fig. 26) is typical of the polar lever type.

Polar lever clutch applications are not new in the industry, but application to the Powermatic was not an "engineering breeze." Several factors had to be reckoned with, each of which would be a story in itself if research and development were reported individually. To highlight some of these problems, we had to resolve how many and what type clutch plates to use in the intermediate, low, and reverse ranges. Because the mean radius of each of these clutches was 4.16 in., we could not use many plates and still keep inertia and spin losses acceptably low. Use of a single plate seemed the best answer. Construction calculations were based upon maximum available engine hp and torque, converter torque and gear ratios, general "rule of thumb" design factors, and manufacturers' unit loading recommendations for various clutch materials.

To find the area in sq in. of active friction material per hp absorbed during engagement, hp absorbed was calculated.

$$\text{hp} = \text{hp/min} \times \frac{\text{engage time, sec}}{60}$$

$$\text{hp/min} = \frac{2\ ST}{33{,}000}$$

where:

S = Average speed or $\frac{\text{slip speed}}{2}$ at engagement

T = Transmitted torque

To compute the clutch capacity:

$$T = PRNF$$

where:

T = Maximum torque

P = Apply force (normal force)

R = Mean radius of clutch face

F = Coefficient of friction

N = Number of active faces

Realizing what clutch demands were, the proper clutch material had to be found and developed to fit our requirements of low torque peaks on engagement, maximum durability with negligible wear, ability to withstand high unit pressure, and a reasonably high "k" factor.

Materials were tried one by one and discarded for one reason or another. One example is a molded biscuit type plate which seemed to be the answer. It had excellent durability, high tensile and compressive strength and relatively low peak torque. However, the material was too brittle and would eventually crack under shock loading such as that experienced in rocking the vehicle from low to reverse. Nothing would correct this fault without destroying some other favorable characteristic, so it was discarded.

Concurrent with these investigations there were development tests on the effect of grooving the facing to reduce spin losses and shift shock, to improve lubrication, and to reduce devastating effects of excessive engagement temperatures. Results of these tests indicated that for our particular application, grooving configuration was all important.

For example, some configurations caused a centrifugal pumping action with high oil flow which resulted in higher spin loss than caused by oil film shearing. Other patterns provided too many grooves for the amount of oil flowing and resulted in insufficient contact area, causing high rates of wear. Conversely, too few grooves caused plates to skate on the reaction plates from increased dry-out time.

Another outstanding factor was the importance of effectively scavenging throw-off oil from the plates. Lack of scavenging can be the major contributor to high spin losses and clutch dry-out time.

Shift shock and torque peaks are also directly affected by grooving. Test results indicated that for our application a compromise had to be made that would provide maximum contact area for smooth engagements, yet be compatible with durability and low spin losses. The answer was the diamond shaped grid pattern used on the intermediate, low and reverse, and lockup clutches.

Concurrent with development of friction material and groove pattern, various clutch plate configurations were being developed and evaluated. Of these various designs it is noteworthy that the most generally satisfactory design proved also to be the most economical to make. Use of nonmetallic friction material was in itself a sizeable cost saving, besides the fact that mating steel surfaces do not require expensive heat-treatment and surface finishing. Thus a fourth type clutch plate construction was added to the Powermatic: two nonmetallic molded facings incorporating a synthetic rubber augmenting agent bonded to a steel core and splined to a ring gear.

Parking Brake

The view of the transmission in Fig. 27 shows the 9½ by 3-in. external contracting parking brake. It is operated by direct linkage from a hand lever on the cab floor.

Oil Filter and Hydraulic Controls

The full-flow oil filtering system in Fig. 28 has a replaceable cartridge located in the sump pan. It is accessible simply by removing one nut on the oil pan cover.

A great deal of investigation lead to selection of an Orlon material as the filtering element.

We will not dwell upon intricacies of the control system, but will cover a few interesting design features.

The main regulator valve, for instance, maintains mainline pressure at various levels depending upon what gear the transmission is in, trottle position, and energy absorption of the hydraulic retarder.

To eliminate the need for specially machined valve seats, nylon rather than steel ball-check valves are used throughout. Cast aluminum seats withstood ex-

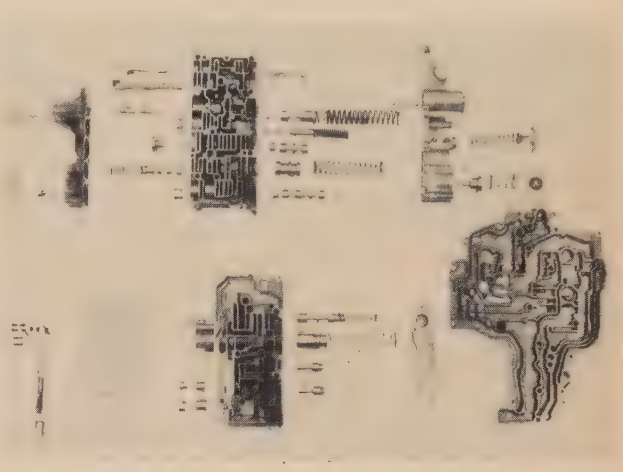


Fig. 29—Hydraulic control components

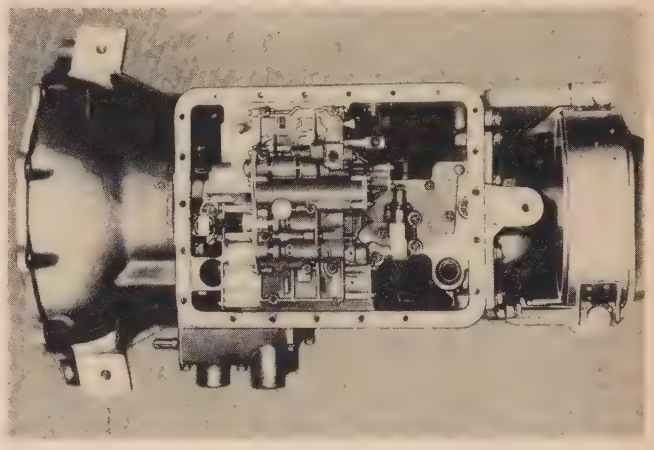


Fig. 30—Valve body

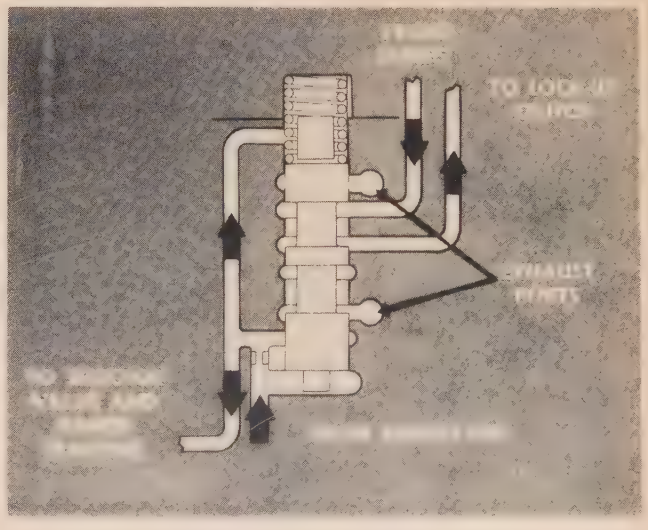


Fig. 31—Lockup cutoff valve action

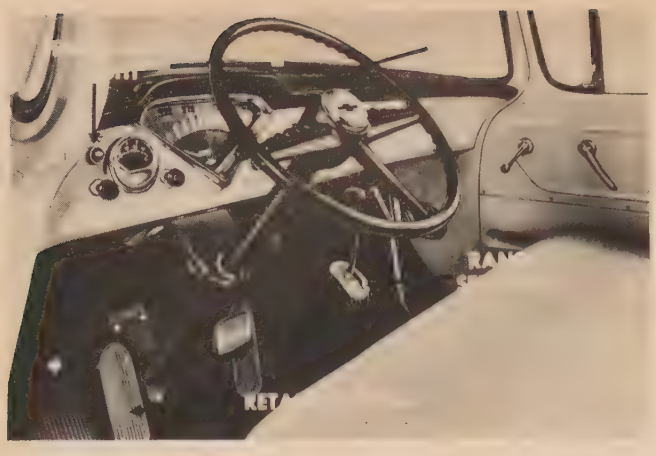


Fig. 32—Cab controls

tremely severe testing with no measurable wear or deformation.

Valve body and all control valves (Fig. 29) are situated in one convenient location (Fig. 30). They are accessible for servicing simply by removing the sump pan.

One interesting control system design detail is the manner in which the lockup clutch is momentarily disengaged during shifts.

Oil going to all planetary clutches is routed through a metering orifice at the lockup cutoff valve (Fig. 31). While the clutch is filling, oil flow causes a pressure drop across the orifice.

The cutoff valve in parallel with the orifice receives high pressure on one end and low pressure on the other. The valve moves against a spring and exhausts the lockup clutch. When the planetary clutches are filled, flow stops, pressure differential disappears, and the spring returns the lockup cutoff valve to its lockup position. The clutch immediately re-engages.

By making the cutoff valve completely dependent upon clutch fill rate, timing of lockup clutch engagement and disengagement is made automatically accurate.

Cab Controls

Powermatic driver controls (Fig 32) consist of accelerator pedal, range selector, and hydraulic retarder pedal.

A decal on the instrument panel warns the driver against shifting from drive to intermediate above 30 mph and from intermediate to low above 15 mph. The tell-tale light indicates when the retarder pedal should be released momentarily to allow oil to be cooled.

Tractive Effort

The tractive effort curve (Fig. 33) demonstrates effectiveness of the torque converter as a starting device. Point "A" represents maximum tractive effort obtainable in the same truck with a Chevrolet conventional four-speed transmission and two-speed axle.

Though Powermatic does not stake its entire performance capabilities on tractive effort curves alone, two points are noteworthy. First, maximum tractive effort is considerably higher than the conventionally shifted job, and second, at the lockup point, the curve is continuous. The conventional performance loss while gear shifting is absent in Powermatic even when shifting from low to drive which is accomplished at the flick of a finger. Continuity of delivered power will enable the most inexperienced driver to realize good performance from the Powermatic-equipped truck.

To place automatically good performance in the hands of any driver, the transmission must deliver very nearly the same horsepower to the rear wheels after an upshift as it did before the shift. Because there is no loss in vehicle speed during shifts and top coverage ratio steps are small, delivered horsepower consistently remains in the high output range. (See Fig. 34.)

Converter starting and efficient upper-range gear changing is the basic combination that delivers performance, economy, and driving ease we sought for our trucks.

Among the most interesting phases in development of a new product are the road trips. Most often certain difficulties are anticipated and the trip serves to enlighten the engineers on some aspects of their problems and to warn them of the seriousness of others. Occasionally new ones they didn't even know existed manifest themselves. We have chosen to describe briefly two typical trips; one to the far Southwest, the other to Pennsylvania.

Two Trips

Despite countless months of searching, probing, laboratory testing, and thousands of miles of proving ground operation, the same burning question kept bothering us that haunts every group of engineers developing a new product: how will it perform under actual working conditions and uncontrolled conditions? What will it do in fair competition with the open highway, tortuous mountain grades, and desert operation? The answer: go find out!!!

We went West in search of trouble. Ours concerned possibly overtaxed cooling systems, overheated brakes; perhaps failed clutches or gears. The vehicle was a nominal 2-ton rated truck loaded to maximum legal axle loading, and included a maze of thermocouples, gauges, and flowmeters—a rolling laboratory. Destination—any tough mountain grade.

Our first real encounter was Colorado's Monarch Pass. Three days of round trips from base to summit on those 5.7–6% grades indicated that we had not yet met our master, but must pay due respect to the obvious burden that the 11,312 ft altitude places on an overloaded engine. Results of tests on these long

grades gave us our first real proof that the hydraulic retarder feature would pay dividends.

We studied the manner in which professional truckers negotiated the grades, timed their speeds, observed their braking habits, and talked to them on all phases of their operations in that type terrain. The result indicated we were definitely on the right track by providing a positive means of augmenting engine braking to place service brakes on emergency standby duty.

To the man behind the wheel, the important part of the story of how fast hilly terrain can be negotiated is not only how fast he can climb uphill, but also how fast he can safely run downhill.

We found that we could safely descend any of these mountain grades, maintain a good road speed with little, and in most cases, no application of the service brakes, by selecting the proper gear range and augmenting normal engine braking with the hydraulic retarder.

To illustrate this point, here is an excerpt from actual test data taken on a descent from the summit of Monarch Pass down the eastern slope on 5 miles of 5-6% downgrade using the retarder only.

Transmission range: Intermediate to fourth gear
Vehicle speed: 27-30 mph
Hydraulic retarder apply: 3500 engine rpm
Hydraulic retarder release: 3000 engine rpm
Total retarder applications: 37
Maximum time per application: 10.8 sec
Average time per application: 5.3 sec
Maximum oil cooler inlet temperature: 375 F

Coffee breaks at truck stops provided an excellent opportunity to do a little research on the location and character of real tough routes and grades. Consequently we charted our course as we went and gave a real going over to places like Wolf Creek Pass,

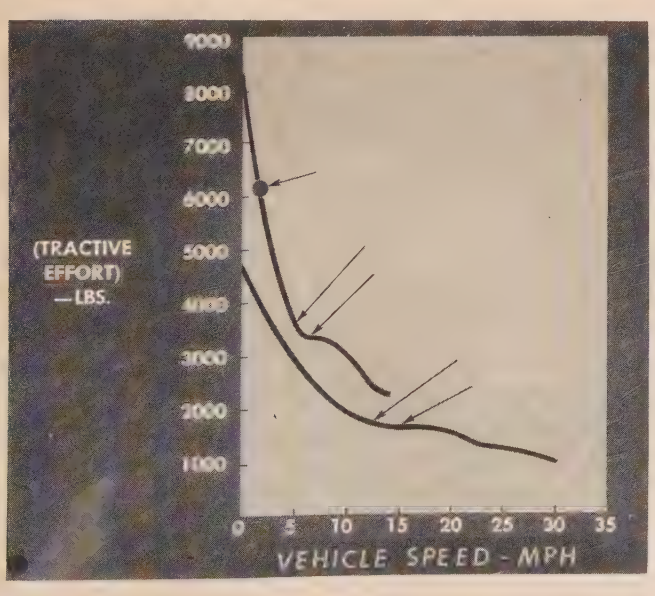


Fig. 33—Drawbar pull versus vehicle speed

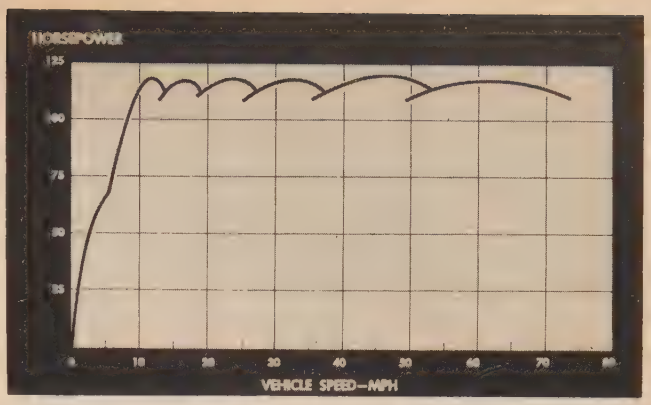


Fig. 34—Transmission output hp versus vehicle speed

Oak Creek Canyon, Yarnell Hill, Black Canyon, Devils Canyon, Salt River Canyon, and all points between.

The stretch of highway which most nearly satisfied our needs for a good test site was one we had previously encountered on alternate U.S. Route 89 between Clarkdale and the old ghost town of Jerome, Ariz. This particular grade is about 5 miles long and averages approximately 6% with short sections that may approach 8%. Elevation rises from 3550 ft at Clarkdale to 5280 ft at Jerome. With only one bad switchback near the top, this grade offered a chance to make some test runs where generally the visibility was good for at least ½–1 mile ahead.

One of the first things we learned the hard way was that low-temperature solder and high-temperature oil are not compatible. Lead soldered fittings in the transmission-oil-cooler feed line literally came "unglued," and it took all the extra pipe fittings in the the Village of Cottonwood, Ariz. to repair the damage and make us mobile again. Practicing the plumbers' trade after dark on the side of a mountain helped reassure us we were pioneering a new field.

On this trip we found the requirements for the engine cooling system were very nearly the same whether ascending grade at full throttle or descending using the hydraulic retarder. Other problems became evident, such as need for more hysteresis on the 1–2 shift valve to prevent hunting on long grades. More development was needed on the lockup clutch valve because of its extreme sensitivity to change in flow rates as affected by high oil temperature leak rates in the valve body.

The first production Powermatics demonstrated on our proving grounds that they measured up, and on most counts, surpassed the handhoned experimental prototypes. The prime objective this time was the Blue Ridge Mountains on U.S. Highway 30 in Pennsylvania, between Ligonier and Fort Loudon. The task force consisted of four fully instrumented trucks, two tractor-trailer rigs, and two straight jobs, one of each type having synchromesh and the other Powermatic.

From the moment the trucks left the proving

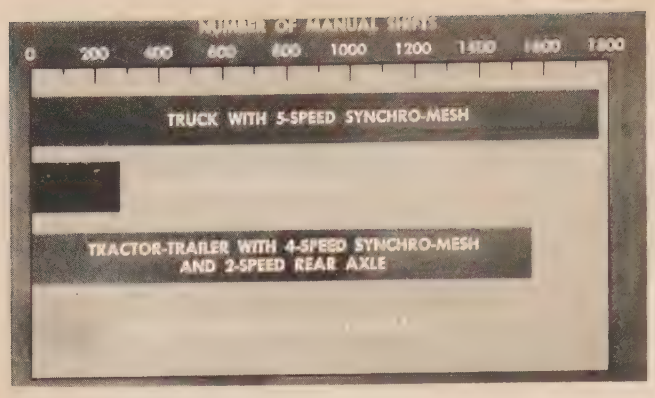


Fig. 35—Comparison of number of manual shifts with synchromesh and Powermatic transmissions on 900-mile trip

ground gates until they returned, operation of every component was very carefully scrutinized, recorded, and double-checked against operation in the other trucks. Total number of transmission manual shifts and brake applications, both service and hydraulic retarder, were automatically counted and recorded. Every member of the trip logged equal time driving and observing in each vehicle.

Several days of concentrated testing on the eastern slopes of Bald Knob and Sideling mountains provoked the realization that it was too early to be taking bows. It seems the transmission-oil filter cartridge then under development tended to become too efficient with age and sharply reduced volume flow through the retarder, producing some fantastic temperature rises. Making further capital of this unfortunate situation we found we had an oil seal on the retarder apply valve that shrunk with temperature rise and literally blew out of its mounting!

Climaxing our problems was the discovery that in partial retarder application, the control valve allowed so much oil to bypass the cooler that it raised sump oil temperature above normal limits. Subsequently these malfunctions were corrected. One notable development was the entirely new full-flow filter of Orlon material.

Chargeable also to this experience was the realization that an instrument-panel-mounted warning light was desirable to warn the driver if and when retarder out oil was approaching the top safe limit of operating temperature. When the warning light flashes red, the driver should momentarily release the retarder and apply it again within a few seconds. In this way he can use his service brakes only to occasionally snub the vehicle if extreme length and steepness of grade warrants it. An example of such a grade is Sideling Hill, which has a constant grade of approximately 7% for nearly 3 miles with no relief.

It has been our experience that the majority of highway grades 6% or over seldom have constant grades much over one mile without some short relief sufficient to allow retarder temperature to recover before the next application. We are speaking of course of heavily loaded vehicles in the 30,000-50,000 lb gvw and gcw class. The lighter vehicles do not usually experience these heat problems because of lower horsepower the retarder absorbs.

Another interesting fact is that for the 900-mile round trip to Bedford, Pa., not including actual hill test operations, regular highway driving provided comparison between the synchromesh transmissions and Powermatics shown in Fig. 35.

Conclusion

The design and development program was long, exhaustive, and, we think, rewarding.

At the outset, variety of opinion on which tack to pursue made it apparent that if there was to be a meeting of minds, certain vital considerations must be removed from the realm of conjecture to demonstrated fact.

Working out test-worthy prototypes for each stage of development is, admittedly, a time-consuming approach. It is an approach, however, that sees a greater majority of well-thought-out ideas actually worked out. Only after a fair trial are any ideas thrown out.

Powermatic, the end product, satisfies the objectives we set out to achieve. Its ultimate success will, as always, be decided in the market place.

ORAL DISCUSSION

Reported by E. B. Cooper

Chrysler Corp.

Question: When will Chevrolet trucks with Powermatic be available to the public?

Mr. Flynn: This information will have to be imparted by the Sales Department. It was believed that an official announcement will be forthcoming soon about their 1956 trucks.

T. Backus, Fuller Transmission Co.: In the West Virginia test what was the total number of transmission shifts for the manual versus automatic units tested?

Mr. Flynn: I do not have total figures available, but typical examples of other comparable road testing are as follows:

1. 1800 manual shifts versus 225 automatic shifts.
2. 475 manual shifts versus no automatic shifts.

O. Brower, Swift & Co.: How does fuel consumption compare between manual and automatic transmission equipped trucks?

Mr. Flynn: No detailed specific data is available, but general information indicates the following:

1. In fleet testing of semitrailer loads (70,000 gcw loaded—25,000 empty), the automatic transmission gave approximately ½ mpg better mileage. (This was on the run from Flint to Saginaw and return.)

2. In Pennsylvania mountain testing the automatic transmissions gave as good as or better gasoline mileage.

3. It was generally noted that more freight or payload could be handled with Powermatic, or if load was the same as for manual units, trucks maintained better road speeds with Powermatic.

S. J. Tompkins, Chrysler Corp.: Had any time comparison tests been made to indicate the relative ability of the Powermatic to maintain better road speeds than the manual units?

Mr. Flynn: No specific data is available; however, during testing over U.S. Highway 30, manual units were always placed

ahead and Powermatic units invariably caught up to and were crowding the manual units.

G. L. Corsi, National Motor Bearing Co.: What is the maximum safe oil temperature when using the retarder?

Mr. Flynn: The warning light on the instrument panel to indicate high retarder oil temperature was set to come on at 365–385 F. The retarder holds approximately 1 qt, and by disengaging the retarder, heat is dissipated quickly, enabling the driver quickly to reapply the retarder.

V. J. Jandasek, Ford Motor Co.: (a) What torque capacity does Powermatic have? (b) What stall speed does the Powermatic have?

Mr. Flynn: (a) Powermatic has a torque rating of approximately 300-325 lb-ft. (b) Converter stall speed is 1800 rpm and retarder stall speed is 2400 rpm.

Mr. Brower: What effect does Powermatic have on start-and-stop performance and on the entire vehicle drive train?

Mr. Flynn: It is generally agreed and accepted that hydraulic converters bring about lowered maintenance costs on drive train components by smooth and cushioned delivery of power to the wheels.

Question: Is the converter needed for normal operation or is it primarily a starting device?

Mr. Flynn: It is needed and used both ways. In the low range it increases torque multiplication and smoothness of delivered power. In normal use, admittedly it is used very little, but does have quite an effect in smoothing shift points.

R. L. Page, Truckstell Mfg. Co.: What effect does Powermatic have on overall brake life?

Mr. Flynn: No direct analysis as been made; however, testing on the West Coast revealed that early automatic transmission testing *without* retarder had required brake adjustments once a week. After adding the retarder, brake adjustments were required only about once every 3 months.

G. W. Michel, Clark Equipment Co.: (a) What oil capacity is required for Powermatic? (b) What is the overall weight of Powermatic?

Mr. Flynn: (a) Sixteen quarts dry and ten quarts wet. (b) Powermatic is expected to average about 500 lb complete and will be approximately 280–305 lb heavier than conventional installations.

W. H. Esser, Shell Oil Co.: Does Powermatic have dual power take-off openings?

Mr. Flynn: Yes.

P. J. Smith, Dana Corp.: How much variation is there in governor control between hot and cold oil?

Mr. Flynn: The pitot tube usage in the governor is not at all sensitive to oil temperature changes, therefore for all normal operation, no variation could be noted.

Question: How much faster road speeds, especially downhill, can be maintained with the retarder?

Mr. Flynn: No specific testing to determine this has been made; however, with the retarder, downhill speeds of 30-35 mph had been made. Manual transmission vehicles over the same route without a retarder device would average 10–15 mph.

G. W. Conover, Jr., Detroit Diesel-Engine Division, GMC: What use does the Powermatic unit have, and especially the retarder unit only, on icy roads?

Mr. Flynn: Using the retarder only on slick roads gave good control by the cushion-even resistance that can be metered easily by the driver.

J. Miller, Clark Equipment Co.: (a) What of the power take-off speeds? (b) Can the power take-off be operated in neutral and in gear?

Mr. Flynn: (a) I can't supply immediate figures, but I'd like to discuss this item following the meeting. (b) The unit can be operated in either neutral or in gear, but must be stopped to be engaged.

D. F. Manz, Reo Motors, Inc.: (a) Is the torque convertor using due multiplication during use of gears? (b) Why was the torque convertor used in place of the fluid coupling?

Mr. Flynn: (a) I can't answer that directly because of the variation according to operating conditions. Curbs would have to be considered to determine torque multiplication obtained for any given engine speed and power and road-load conditions. (b) To give greater flexibility for starting, the torque converter is needed, whereas a fluid coupling has no torque multiplication.

W. F. Lomas, Ford Tractor & Implement Division: (a) Why were spur gears used in place of helical gears? (b) Was there any difficulty encountered with noise from using spur gears?

Mr. Flynn: (a) In the coaxial design of the unit one of the primary reasons for using spur gears was the absence of axial thrust. (b) As for noise, there is some nosie which can be basically attributed to spur gears but it is considered to be an unreasonable amount.

N. A. Stenzel, Thompson Products, Inc.: Is the full capacity of the radiator utilized with retarder operation?

Mr. Flynn: About one-half the theoretical heat dissipation requirement was apparently needed in actual usage. The cooling needs for the retarder were approximately 2100 to 2200 Btu. Concerning cooling requirements, approximately the same was needed downhill with retarder as was needed uphill under power.

J. D. Collins, Warner Gear Division, Borg-Warner Corp.: Is it feasible to use the engine governor with this unit, and what speed should be required for governor seating?

Mr. Flynn: A governor can be used with the unit. However, since the shift points under full power were about 3800 rpm, the governor should not be set below approximately 4000 rpm for full performance.

M. M. Schall, Dana Corp.: With lockup disengaged, will engine speed be excessive with normal down shifting?

Mr. Flynn: There are no overspeed devices (inhibitors) to prevent wrong downshifting. If abused, overspeeding and transmission difficulties could be experienced. However, when normally and practically used, the unit has behaved quite satisfactorily and does give the driver feature power over certain automatic shifting sequences.

R. B. Singer, Dana Corp.: What happens to engine speed during shift points?

Mr. Flynn: Since shifts are automatic and under full power, no noticeable effect on engine rpm is encountered.

W. F. Meyers, Fuller Mfg. Co.: (a) On tests reported to give approximately ½ mpg better with Powermatic, were the same carburetors used with both types of transmissions? (b) Are transmission controls connected to throttle controls?

Mr. Flynn: (a) The same basic carburetors were used for both types of transmission test vehicles. (b) Yes, it is a throttle-sensitive design.

Regarding fuel consumption, during the test run which required 475 manual shifts, with normal shift time of 2–2.5 sec being required, there were approximtaely 1000 sec in which no power could be transmitted to the wheels. Furthermore, with double clutching, 950 depressions of the accelerator would use that many accelerating pump shots of fuel, or approximately 200 cc of fuel wasted.

Question: At what speed does the torque converter stop multiplying torque?

Mr. Flynn: No direct answer can be given since it would depend on engine speed and power output in conjunction with road-load requirements.

Question: Will Powermatic be available in units up to 21,000–22,000 GVW?

Mr. Flynn: Our sales department will have to impart such information.

A. A. Dach, Chestnut Farms-Chevy Chase Dairy Co.: What start-stop testing has been done, similating city delivery-type service?

Mr. Flynn: The Powder Valley testing was primarily of this type, and involved an 11% grade approximately 1½ miles long.

W. H. Brown, U-Drive-It Co.: What engines will be available with Powermatic?

Mr. Flynn: All engine sizes will have Powermatic available.

Question: Why were two stators and single-stage converters used?

Mr. Flynn: The two stator design was superior to the single stator type, and with the lockup device, it does not operate as a fluid coupling. Therefore, all design could be directed toward maximum converter efficiency.

Question: Referring to the above answer, why were two stators used at the start *before* the lockup mechanism was developed and added to the system?

Mr. Flynn: This question obviously entails many ramifications and facets of explanation which are too detailed and lengthy. Actually, this was while we were still learning.

Compressor and Turbine Matching

THE present trend toward use and development of turboprop engines has resulted in the need for a better understanding of this type engine. The purpose of this paper is to point out some design considerations and characteristics noted from matching studies made for three engine configurations of current interest. Effect of flight conditions and mode of engine operation on engine performance and design requirements will also be discussed.

The type turboprop engines discussed are shown in Figs. 1, 2, and 3. A schematic of a single-spool turboprop engine is shown in Fig. 1. In this version, as well as other versions of the turboprop engine, jet thrust is usually of secondary importance compared to thrust obtained from the propeller. It is this characteristic that gives the turboprop engine, in comparison

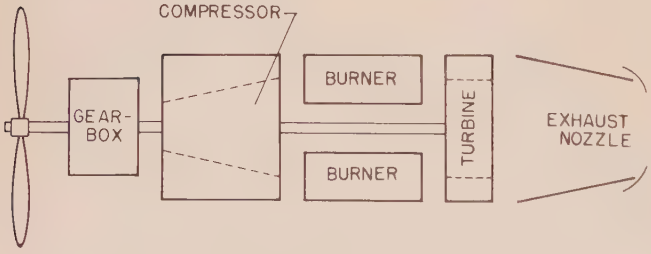


Fig. 1—Schematic of single-spool turboprop engine

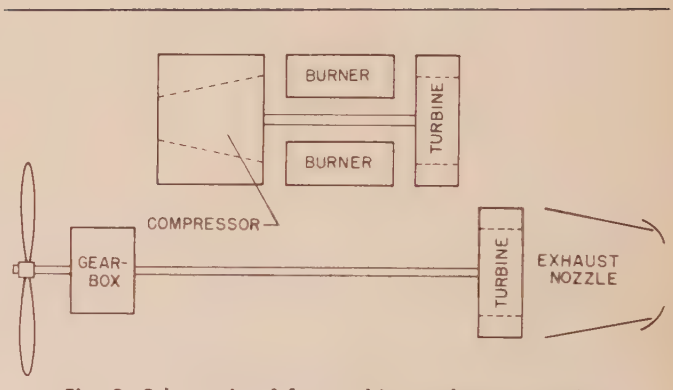


Fig. 2—Schematic of free-turbine turboprop engine

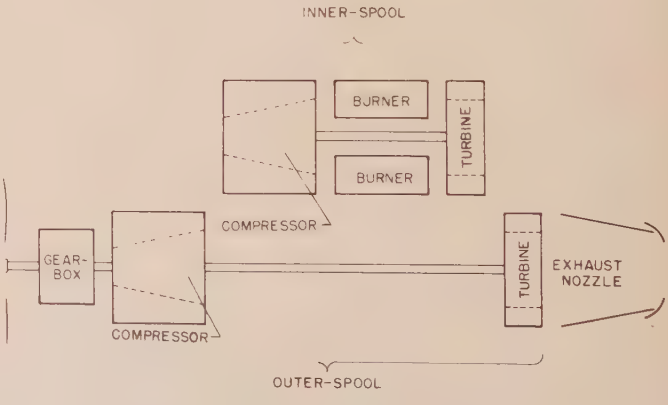


Fig. 3—Schematic of two-spool turboprop engine

Considerations in Turboprop Engines

Elmer H. Davison, NACA

This paper was presented at the SAE Annual Meeting, Detroit, Jan. 13, 1956.

MATCHING studies of three turboprop engine configurations were made for flight conditions from sea-level static to 600 mph at 40,000 ft.

It is concluded that turbine frontal area, stress, and pressure ratio requirements made exhaust-area adjustment desirable. Sfc depended primarily upon flight conditions and turbine temperature, with lowest sfc occurring at highest turbine temperature, flight velocity, and altitude.

Free turbines restricted turbine temperature range and produced critical turbine requirements. Increasing a two-spool engine's outer-compressor pressure ratio increased turbine temperature range and made turbine requirements less critical.

to the turbojet engine, its high specific thrust at take-off and good propulsive efficiency at low flight speeds. In the single-spool engine, turbine power in excess of that required to drive the compressor is transmitted to the propeller through a gear box. Another version of the turboprop engine, the free-turbine engine, is shown in Fig. 2. In this type engine a gas generating unit composed of a directly coupled compressor and turbine delivers hot gas to the free turbine. Power developed by the free turbine is transmitted to the propeller through a gear box. Still another version of the turboprop engine, the two-spool engine, is shown in Fig. 3. This type engine is composed of two directly coupled compressor and turbine combinations. The inner compressor and turbine shaft is usually concentric with the shaft of the outer compressor and turbine combination. The inner turbine develops only enough power to drive the inner compressor. The outer turbine develops power in excess of that needed by the outer compressor; this excess power is

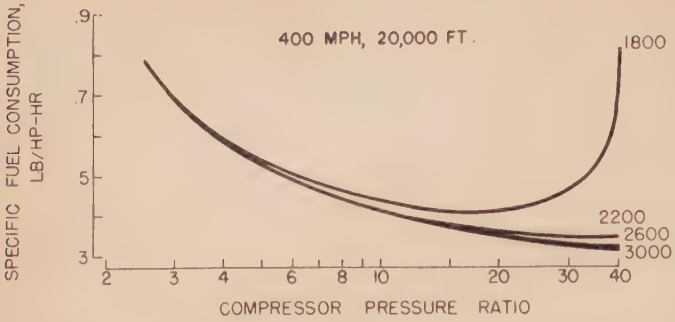


Fig. 4—Effect of turbine temperature and compressor pressure ratio on sfc

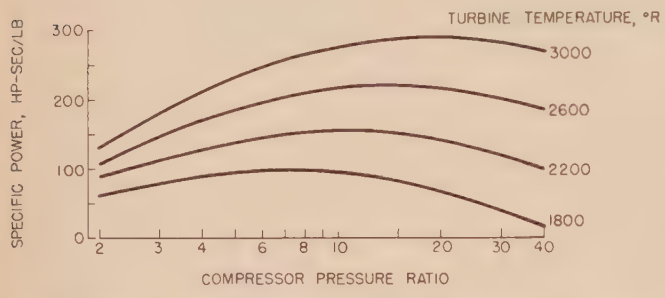


Fig. 5—Effect of turbine temperature and compressor pressure ratio on specific power

transmitted to the propeller through a gear box. Other differences between these three types of engines will be discussed later.

The trend in turboprop engines is to higher compressor pressure ratios and higher turbine inlet temperatures because they lead to better economy and lighter, smaller engines. This is readily apparent from Fig. 4, which shows typical variations of sfc and specific power with compressor pressure ratio and turbine inlet temperature. Fig. 4 is a plot of sfc against compressor pressure ratio for various values of turbine inlet temperature. Fig. 5 shows horsepower developed per pound of air passed through the engine against compressor pressure ratio for the same turbine inlet temperatures shown in Fig. 4. It is seen from Fig. 4 that a major reduction in sfc is obtained by going to higher compressor pressure ratio, while little, if any, is obtained by increasing turbine inlet temperature. The chief advantage in going to higher turbine inlet temperatures is to increase the specific power, as shown in Fig. 5. In order to realize the full potential of an increase in turbine inlet temperature, it should be accompanied by an increase in compressor pressure ratio, as shown in Fig. 5.

Single-spool and free-turbine engine analyses are based on hypothetical performance of a single-spool compressor with a design pressure ratio of 14.4. Single-spool pressure ratios of 14.4 are not current and some designers may choose to use two-spool compressors when compressor pressure ratios of 10 or higher are contemplated. An overall design compressor pressure ratio of 12.0 was used in the two-spool engine analysis because it could be divided easily between the two spools. Other effects of compressor pressure ratio will be discussed later.

The single-spool engine for constant rotative speed mode of operation is discussed first. We will be concerned with the effect of flight condition and exhaust-nozzle area on, and the interrelation of, such things as turbine pressure ratio requirements, turbine stress, turbine to compressor frontal area ratio, sfc, and engine power. Integrated into this discussion will be a comparison with several other modes of engine operation. The free-turbine and two-spool engine discussions will be concerned mainly with the difference between these two types and the single-spool engine. A summary discussion will be presented.

Single-Spool Engine

Cycle analyses for the single-spool engine were based on performance of a hypothetical compressor. Performance of this hypothetical compressor is shown in Fig. 6. A conventional plot of compressor pressure ratio against equivalent weight flow per unit compressor frontal area is shown. Lines of constant compressor equivalent speed are shown as a percent of design speed. In addition, surge limit and contours of constant adiabatic efficiency are shown. At design-point operation, equivalent tip speed of this hypothetical compressor is 1168 fps, pressure ratio is 14.4, equiva-

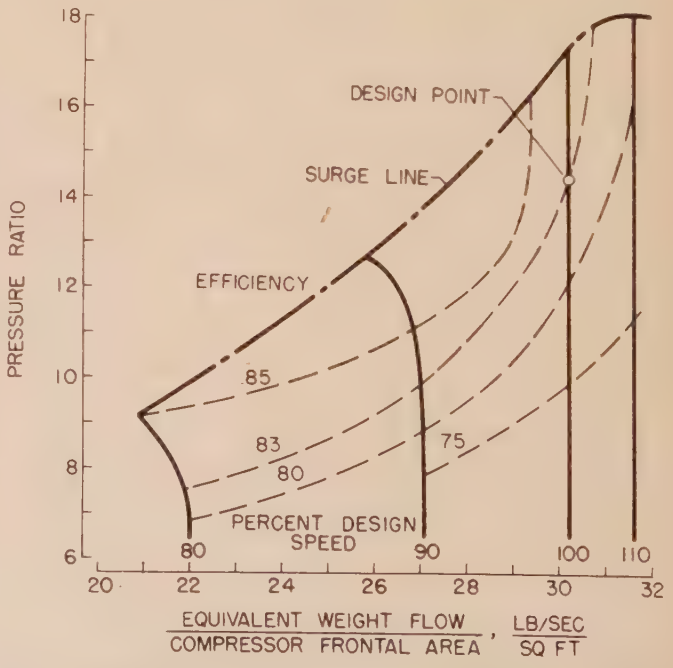


Fig. 6—Compressor performance for single-spool and free-turbine engines

lent weight flow per unit compressor frontal area (based on compressor blade tip diameter), is 30.2 lb per sec per sq ft, and compressor peak efficiency is near 83%.

Flight, Exhaust-nozzle Area, and Turbine Inlet Temperature Effects—There are many modes of engine operation possible, but we will confine ourselves at this point to engine operation at constant design rotative speed (hereinafter referred to as mode 1). For this mode of operation the compressor will operate along a different constant equivalent speed line for each flight condition, with compressor pressure ratio increasing as turbine-inlet temperature increases. Maximum turbine-inlet temperature is determined, therefore, by the compressor surge limit.

For mode 1 and at sea-level static conditions, Fig. 7 shows variation of power per unit compressor frontal area with ratio of exhaust-nozzle area to compressor frontal area for constant values of turbine-inlet temperature. Lines of constant turbine total-pressure ratio are also shown. Power plotted in this and subsequent figures is the sum of shaft power to the propeller plus equivalent shaft power of the jet thrust. It is apparent from Fig. 7 that the greatest power at a given turbine-inlet temperature is obtained when exhaust-nozzle area is as large as possible. For a given turbine-inlet temperature and flight condition, compressor operating conditions and, therefore, fuel flow, remain fixed. With constant fuel flow lowest sfc also occurs at that point if exhaust-nozzle area is as large as possible. Fuel flow changes, however, as turbine-inlet temperature changes. A reasonable ratio of exhaust-nozzle to

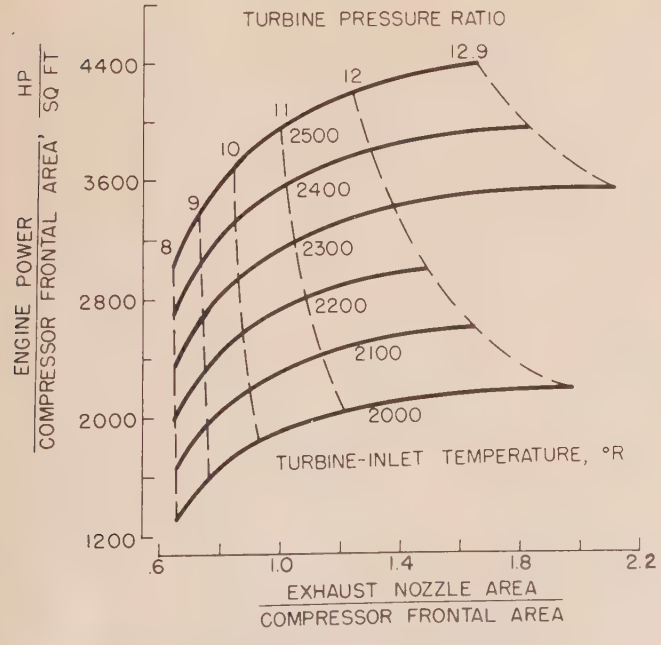


Fig. 7—Single-spool engine requirements operating at mode 1 at sea-level static conditions

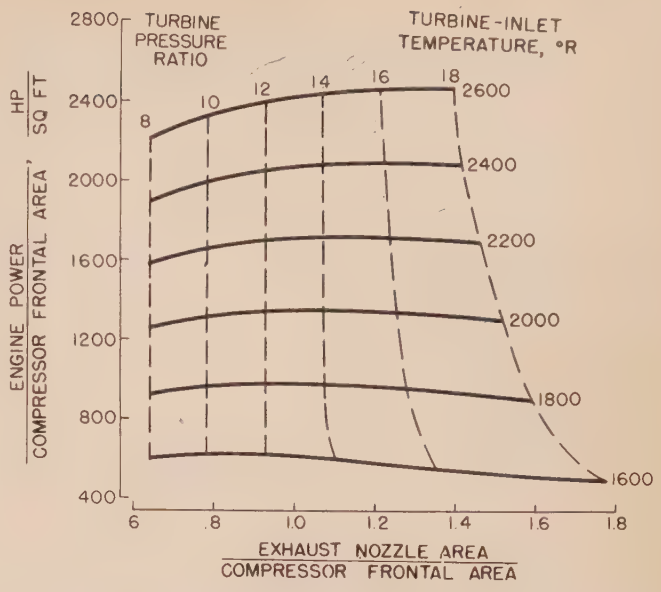


Fig. 8—Single-spool engine requirements operating at mode 1 at 600 mph and 40,000 ft

compressor frontal area would be about 1.4, which as we will see shortly, is a good compromise between sea-level static and other flight conditions if the engine is operated with constant exhaust-nozzle area. Note also that only minor reductions in sfc and minor increases in power can be achieved by increasing the area ratio above 1.4. With area ratio constant, the required turbine pressure ratio increases with turbine-inlet temperature.

In Fig. 8 we have the same plot for a flight velocity of 600 mph and a 40,000-ft altitude. In Fig. 8 temperature lines have leveled out, indicating that exhaust-nozzle area size doesn't have the influence on power output it did at sea-level static conditions. This also means that sfc along a constant temperature line doesn't vary greatly with exhaust-nozzle area size. If the ratio of exhaust-nozzle to compressor frontal area of 1.4 prevailed at this flight condition (that is, operation with constant exhaust-nozzle area), then regardless of the turbine-inlet temperature selected, the required turbine pressure ratio has increased. However, the value of required turbine total-pressure ratio could be reduced to the values at sea-level static (approximately 12.0), by reducing the area ratio to between 0.9 and 1.0. If the engine were operated with constant exhaust-nozzle area, the turbine would have to be designed for a range of pressure ratio from 11.5 to 18.0. This would make turbine design much more difficult than for operation with variable exhaust-nozzle area and would also require more turbine stages. We will see later that operation with a variable exhaust-nozzle area is also desirable from turbine frontal area and stress considerations. Flying at lower flight velocities and altitudes would reduce the range

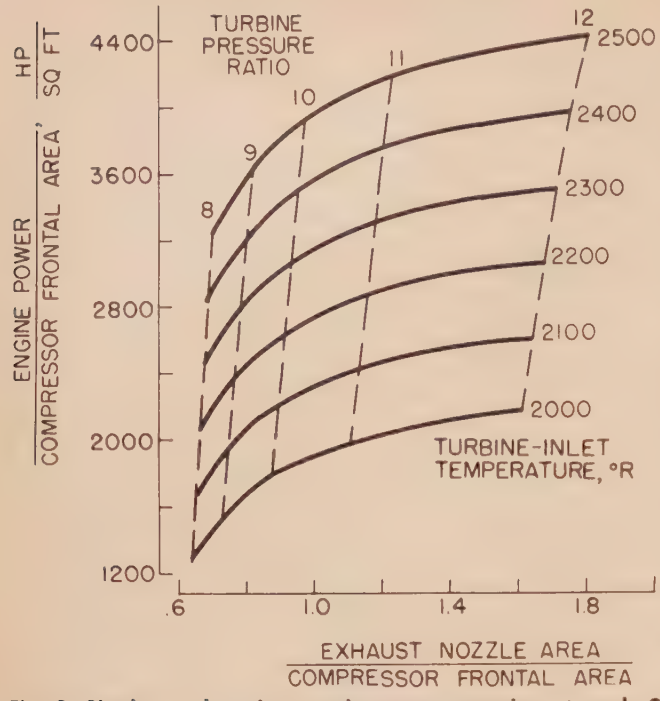


Fig. 9—Single-spool engine requirements operating at mode 2 at sea-level static conditions

of turbine pressure ratio required for constant exhaust-nozzle area operation.

Mode of Operation Effects—The other two modes of engine operation considered were:

Mode 2: Compressor operating at design equivalent conditions at all times. This is accomplished by assuming the turbine could accomodate resulting variations in turbine-inlet equivalent weight flow by turbine stator adjustment.

Mode 3: Compressor operating at constant equivalent design rotative speed.

Variations of power with exhaust-nozzle area for the same flight conditions previously noted are shown in Figs. 9 and 10 for mode 2. Similarly, power variations for mode 3 are shown in Figs. 11 and 12. It can be seen from these two sets of figures that essentially the same conclusions drawn from Figs. 7 and 8 for mode 1 apply to the other two modes of operation. The differences between modes 1 and 3 are negligible. For mode 2, higher turbine-inlet temperatures were obtained because the compressor didn't enter the surge region. However, mode 2 requires the complication of adjustable turbine stators, and it was assumed that the required variation in turbine-inlet equivalent weight flow (perhaps as much as a 25% increase over design values), could be handled by adjusting the turbine stators. In actual practice, the variation in choking inlet equivalent weight flow that can be handled in a multistage turbine by stator adjustment is restricted. This, in turn, restricts the range of turbine-inlet temperature over which one can operate under mode 2. In practice, therefore, the range of turbine-inlet temperature for mode 2 might be less than for the other two modes, even though surge is not a limit. Figs. 7–12 show that the range of turbine pressure ratio required of the turbine for constant exhaust-nozzle area operation would be somewhat less for mode 2 than for either modes 1 or 3.

Turbine Stress and Frontal Area—Let us look at the effect on turbine blade stress of the flight and operating conditions considered previously. Other parts of the engine and propeller present stress problems, but the turbine blade stress problem is usually paramount and often dictates certain engine compromises. Turbine blade stress is made critical by the high temperature prevailing in the turbine, and the stress of greatest importance results from centrifugal force acting on blades.

Centrifugal stress at the hub of the last turbine rotor (highest in the turbine), for operation under mode 1 with constant exhaust-nozzle area ($A_e/A_c = 1.40$), is presented in Fig. 13. Of the engine conditions investigated, mode 1 with constant exhaust-nozzle area operation usually results in the largest required exit annular area and highest rotative speed. Fig. 13, then, represents the highest stresses encountered because centrifugal stress in rotor blades varies directly with annular flow area of the rotor and the square of the rotative speed. However, stresses calculated for these operating conditions were minimized by assuming the turbine to be designed at limiting loading. When exit axial Mach number of a turbine reaches approximately 0.7 (actual Mach number is a function of blade trailing-edge blockage), turbine work output is at a maximum and we say the turbine is at limiting loading. With the total state at the tur-

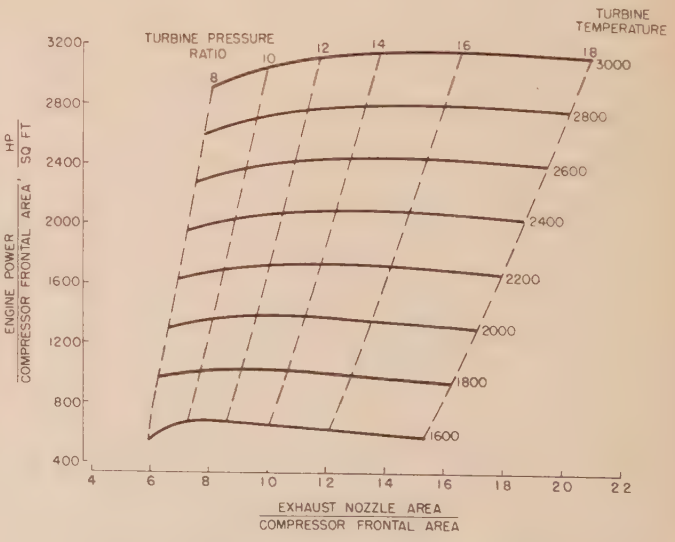


Fig. 10—Single-spool engine requirements operating at mode 2 at 600 mph and 40,000 ft

bine exit (determined by work output and turbine efficiency), and the weight flow known, the minimum permissible exit annular area can, therefore, be calculated by assuming the exit axial Mach number is 0.7.

The left side of Fig. 13 shows the effect of both flight condition and turbine-inlet temperature on stress. Total temperature relative to the rotor blade hub may be determined by following the figure to the right side as indicated. Total temperature relative to the rotor, for practical purposes, is the blade metal temperature at these conditions. A 100-hr stress-rupture curve for a common high-temperature blade material terminated by the curve for the 0.2% offset yield strength of this material is superimposed on the right side of Fig. 13. Stresses to the left or below these curves are within the material stress limits for a 100-hr life.

For flight conditions considered in Fig. 13, turbine-inlet temperatures up to 2600 R can be reached without the exit stress exceeding the 1000-hr stress-rupture limit. However, these stresses are a minimum for the flight conditions shown, and no design safety factor has been employed. For example, increasing the stress-rupture limit to 1000 hr would drop the allowable blade temperature by about 70–80 R, and unless blade cooling were employed, limits turbine inlet temperature to about 2500 R. These are not absolute limits since different materials would have different stress-rupture curves. We might conclude that without blade cooling, stress in the last turbine rotor limits turbine inlet-temperature to about 2600 R.

Increase in stress with altitude and flight speed results from increase in turbine total-pressure ratio required for operation with a constant exhaust-nozzle area. Increased pressure drop across the turbine requires a larger exit annular area and consequently increases stress. This altitude stress, then, could be reduced by decreasing exhaust-nozzle area. For example, stress at a turbine-inlet temperature of 2200 R could be reduced from roughly 70,000 to 49,000 psi by a 35% reduction in exhaust-nozzle area. For the example presented, no reduction in sfc resulted from the exhaust-nozzle area reduction. Decreasing blade stress by reducing exhaust-nozzle area is offset by an increase in the blade metal temperature which reduces the allowable stress. However, a small in-

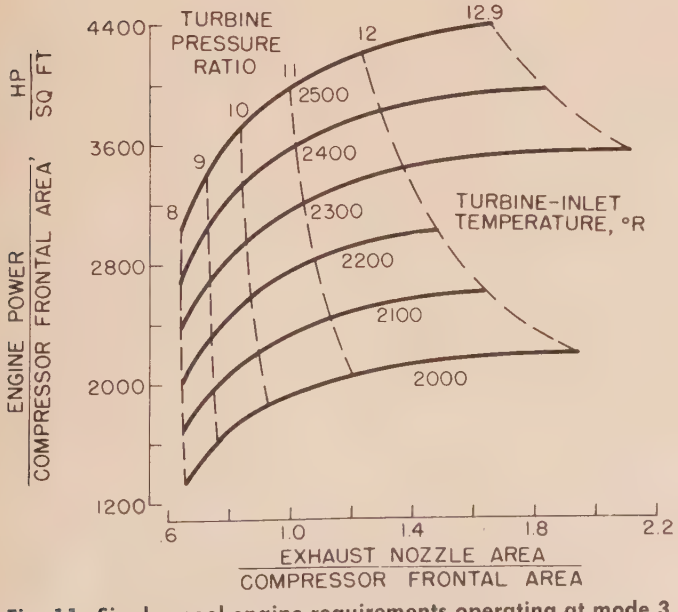


Fig. 11—Single-spool engine requirements operating at mode 3 at sea-level static conditions

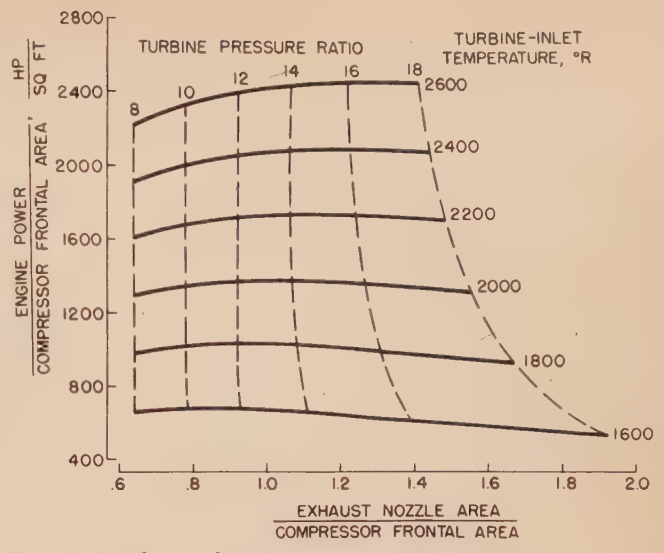


Fig. 12—Single-spool engine requirements operating at mode 3 at 600 mph and 40,000 ft

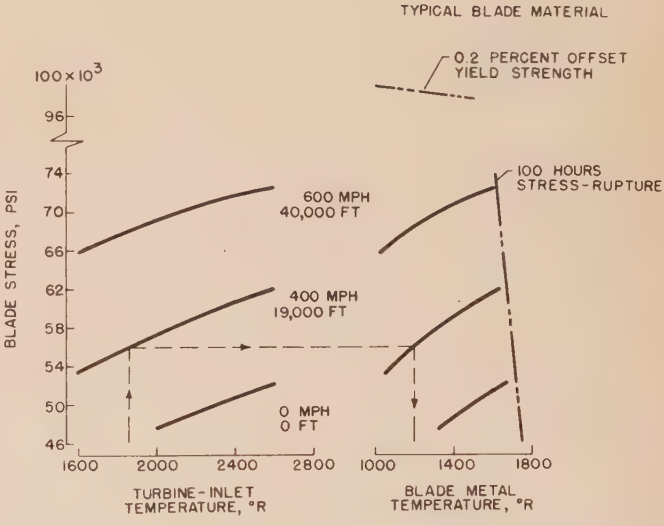


Fig. 13—Turbine stress for single-spool engine with constant exhaust area operating at mode 1

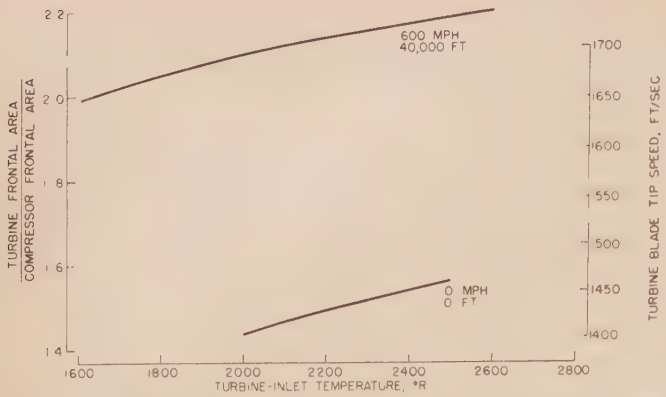


Fig. 14—Turbine frontal area of single-spool engine with constant exhaust area operating at mode 1

crease in the margin between operating stress and stress-rupture limit can usually be obtained by reducing stress. Even a small increase is important since only a 70 or 80 R change in metal temperature can increase the stress-rupture life from 100 to 1000 hr. Actual improvement obtained depends upon the strength properties of the material used. It should be noted that an engine designed for constant exhaust nozzle area operation must be designed with a turbine exit annular area large enough for altitude operation even though a smaller area would suffice at the sea-level static condition. In a design of this type the high turbine stress shown for the 600-mph and 40,000-ft flight condition would prevail at the sea-level static condition, but with a higher blade-metal temperature. The higher blade-metal temperature results from the lower pressure drop across the turbine. A turbine designed for the altitude condition could, therefore, exceed the stress-rupture limit at the sea-level static condition or require a reduction in the turbine-inlet temperature.

Centrifugal stress at the hub of the first rotor was estimated to be between 10,000 and 12,000 psi. The first rotor stress is assumed constant for all flight conditions and turbine-inlet temperatures. Under mode 1 for the operating conditions analyzed, inlet equivalent weight flow to the turbine was assumed constant and the first turbine stator choked. Because these assumptions result in a constant first stator throat area, we predicate that annular area of the first rotor and, therefore, stress are constant. For a 100-hr stress-rupture limit, a blade with a stress of 10,000 psi operating at a turbine-inlet temperature of 2600 R borders on the need for blade cooling. A detailed study of all blade rows in the turbine would be required in order to determine the turbine-inlet temperature at which rotor blade cooling becomes necessary. The analysis for this paper indicates the temperature is around 2500 or 2600 R.

High blade stresses at the turbine exit and low blade stresses at the turbine entrance stem from the high compressor pressure ratios used. Indirectly these stresses indicate other turbine design problems. These stresses are accompanied by very short blades in the initial turbine stages and long blades in the later stages. Obviously such blades can produce severe aerodynamic and structural problems, and both types will present difficult cooling problems if the turbine does have to be cooled.

Variation of turbine frontal area with flight condition and turbine-inlet temperature is shown in Fig. 14. Operation under mode 1 and with constant exhaust-nozzle area (ratio of exhaust-nozzle to compressor frontal area of 1.4), are again considered. Turbine frontal area, like stress, is a minimum for these conditions since the turbine was considered to be at limiting loading at all times. Fig. 14 also shows turbine blade tip speeds based on a compressor blade tip speed of 1168 fps. In calculating turbine frontal area and blade tip speed it was assumed that the hub-tip radius ratio of the last turbine rotor blade was 0.6. A lower value of hub-tip radius ratio would reduce turbine frontal area, but 0.6 was considered a practical lower limit. In addition, lowering hub-tip radius ratio tends to increase the number of turbine stages required. A lower hub-tip radius ratio means a lower hub blade speed or less work output per blade row if aerodynamic limits are imposed on the flow in the turbine.

No limit was placed on turbine frontal area for the present analysis. However, it appears from Fig. 14 that turbine frontal areas at altitude for constant exhaust-nozzle area operation are unreasonably large. Areas as small as those shown for the sea-level static curve could be achieved at the altitude conditions if

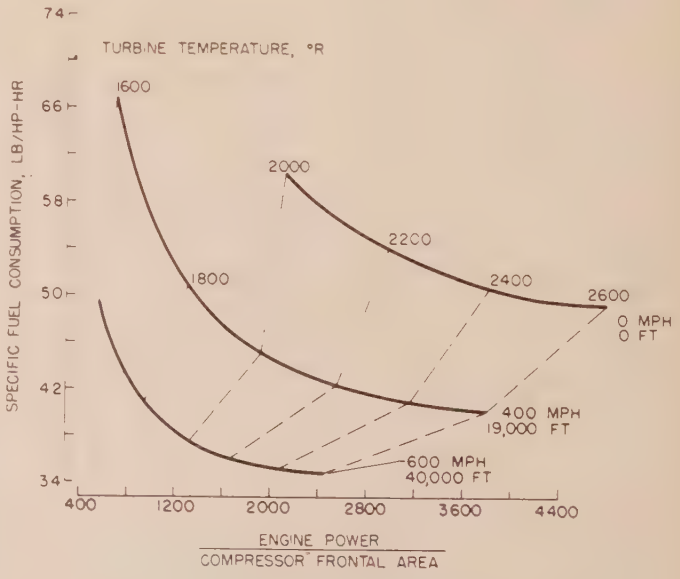


Fig. 15—Sfc for single-spool engine with constant exhaust area operating at mode 1

the required turbine design pressure ratio were reduced by decreasing exhaust-nozzle area. For the example used previously (stress reduction at altitude condition and a turbine-inlet temperature of 2200 R), the ratio of turbine to compressor frontal area could be reduced from 2.14 to 1.49 if exhaust-nozzle area were reduced by 35%. As noted previously for this example, no reduction in sfc resulted from this exhaust-nozzle area reduction. A variable exhaust-nozzle area would, therefore, be beneficial in reducing both turbine frontal area and turbine stress and would make other turbine requirements less critical without seriously affecting engine performance.

Turbine blade tip speeds shown in Fig. 14 are higher than those normally considered in turbine designs. However, high turbine blade tip speeds help in designing an efficient turbine with a minimum number of stages if resulting turbine stress is not a problem.

Turbine frontal area and stress problems are very similar for either mode 2 or 3. Stress at altitude for either mode 2 or 3 would be less than for mode 1 because the actual rotative speed is less. However, at sea-level flight it would be greater.

An obvious means of reducing turbine stress is to reduce engine rotative speed (that is, the design blade tip speed of the compressor). This is undesirable, however, because it increases the number of compressor and turbine stages required to do the job. The rotative speed selected is a compromise which must also take into consideration such things as compressor and turbine aerodynamics. For these high pressure ratio engines the number of compressor and turbine stages required are considerable, and turbine blade stress is pushed to the limit. For a low pressure ratio engine one would probably consider adding stages in order to reduce blade stress. The rotative speed selected for this analysis represents a good compromise of the factors involved. Another means for alleviating a part of the stress problem is to use a free turbine whose rotative speed is independent of compressor characteristics. This type engine will be discussed in detail later.

Specific Fuel Consumption Considerations—The sfc is as important as any consideration in determining the optimum engine. Let us, therefore, look at the sfc variations of the single-spool engine.

Sfc variation with power output is shown in Fig. 15 for three flight conditions. The curves shown are for mode 1 and constant exhaust-nozzle area operation. Lowest sfc occurred at the highest flight velocity and altitude considered. It was found, in general, that either decreasing altitude below the tropopause or decreasing flight velocity increased sfc. It is also seen that for any flight condition sfc decreases as turbine-inlet temperature (or engine power), increases. Note also that very little reduction in sfc is obtained at the altitude condition by increasing turbine-inlet temperature beyond 2600 R. Lower sfc values than those shown could be obtained if the exhaust-nozzle area were varied. With the exception of some low power

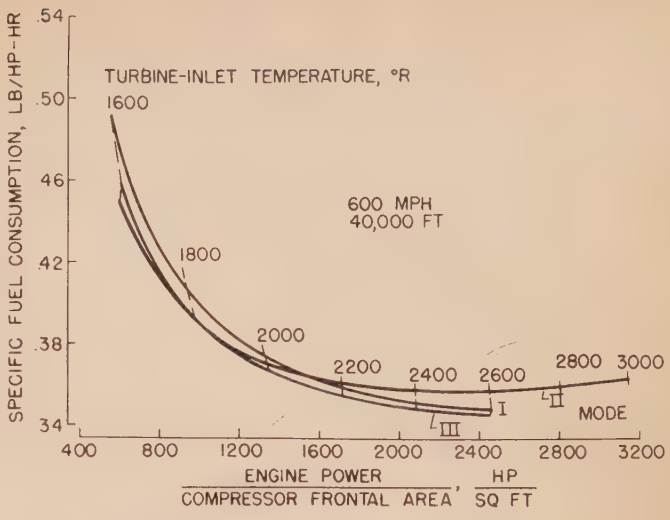


Fig. 16—Effect of mode of operation on sfc

conditions, the improvement would be minor. With respect to sfc, then, constant exhaust-nozzle area operation would be satisfactory. However, we showed earlier that reducing exhaust-nozzle area at altitude was desirable because it reduced required turbine frontal area, turbine blade stresses, and range of pressure ratios over which the turbine must operate. This reduction of exhaust-nozzle area in comparison to constant exhaust-nozzle area operation is accompanied at high power by as much as a 3 or 4% increase in sfc and at low power by as much as a 13% decrease in sfc. In the power range of interest the difference is, in general, quite small. It is possible, therefore, to achieve these desirable objectives without suffering a severe penalty in sfc.

Differences between sfc variations for the three modes are shown in Fig. 16. Only one flight condition is presented, again, with operation at constant exhaust-nozzle area. Within the accuracy of the analysis, the advantage of one mode over another was negligible (see Fig. 16).

In calculating sfc variations it was assumed that turbine adiabatic (or total-pressure ratio), efficiency was constant, at a value of 85%. As ratio of turbine power to compressor power decreases, the assumption of constant turbine efficiency becomes less appropriate because turbine efficiency variations become increasingly critical in determining sfc variations. For conditions of this analysis an assumption of constant turbine efficiency is reasonable, and in particular, for mode 1 when exhaust-nozzle area at altitude is reduced and consequently turbine pressure ratio variation is small.

We noted from Fig. 15 that operating the engine at full power (or limiting turbine-inlet temperature), and flying at the highest flight speed and altitude considered resulted in minimum sfc for the engine. It can be shown analytically that the airplane has its

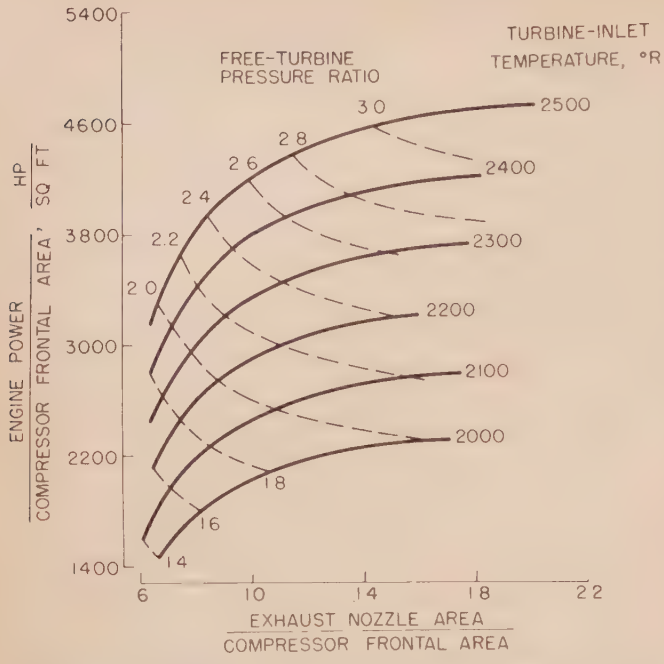


Fig. 17—Free-turbine engine requirements at sea-level static conditions. Turbine temperature and exhaust area effects

greatest range potential under these conditions if at the same time it is designed to achieve its maximum lift-drag ratio under the same conditions. This is in contrast to airplanes using reciprocating engines which are throttled back in order to reduce sfc and increase range. Reduction of sfc obtained by increasing power, flight speed, and altitude may be countered by a drop in propeller efficiency. At some point this loss may more than nullify the improvement in sfc. The greatest range potential from the standpoint of the engine-propeller combination may occur at flight conditions other than those for best engine sfc. In addition, sfc reductions indicated may not be achieved fully if engine component efficiencies drop as altitude increases. If one also considers a whole flight plan, it may not be advantageous to fly at the higher flight velocities and altitudes and thereby lower engine sfc. If much time is spent loitering, for example, minimum use of fuel per given time is obtained by throttling back the engine and flying at a slower speed.

Free-Turbine Engine

Free-turbine engine analyses were based on the same hypothetical compressor performance shown previously (Fig. 6). Much of the discussion pertaining to the single-spool engine applies to the free-turbine engine. We will concern ourselves, therefore, mainly with differences between these engines. Compressor operation at constant design rotative speed only (mode 1), will be discussed, because results for this mode are typical of the other modes analyzed.

In the analyses equivalent weight flow to the free-turbine was allowed to vary without restrictions. For a turbine with fixed first stator area, inlet equivalent weight flow in the design operating region is constant, or at most has a small variation. Allowing inlet equivalent weight flow of the free-turbine to vary, therefore, implies adjustment of at least the first turbine stator. In actual practice variations in inlet equivalent weight flow that can be obtained in a turbine are restricted, among other things, by the change in blade chord angle or throat area required. Analytical investigations indicate that for a multistage turbine the resulting percent change in inlet equivalent weight flow is less than the percent change in stator throat area. In addition, this discrepancy becomes greater as the number of turbine stages increase. Analytical results for a three-stage turbine indicate that roughly a 10% increase and 20% decrease from design equivalent weight flow are possible by stator adjustment for a given turbine work output.

Variation of engine power with exhaust-nozzle area for the free-turbine engine at sea-level static conditions is presented in Fig. 17. The figure shows lines of

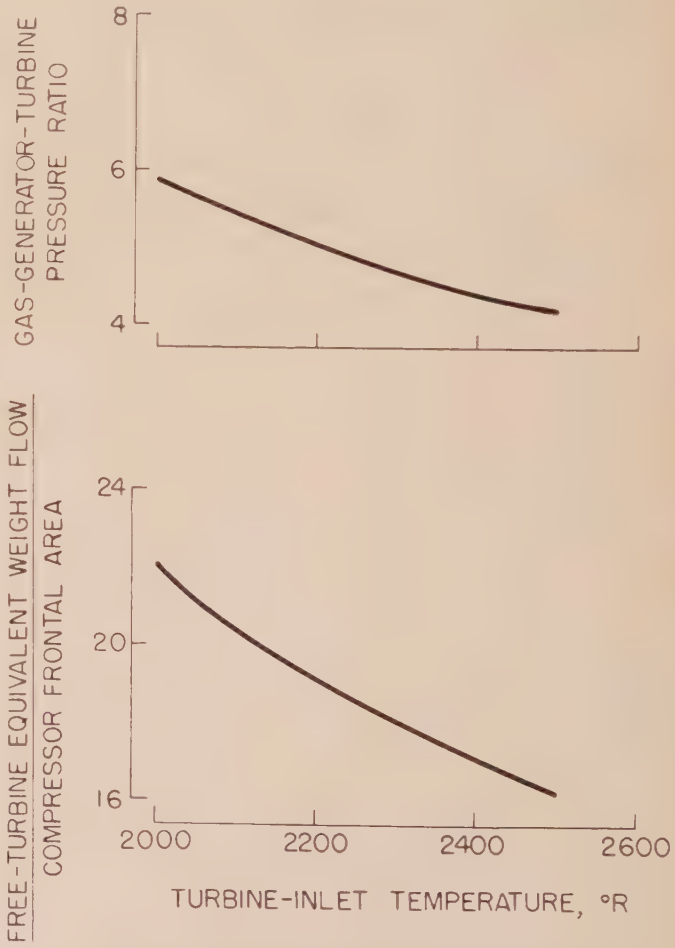


Fig. 18—Free-turbine engine requirements at sea-level static conditions. Turbine temperature effects

constant turbine-inlet temperature and pressure ratio across the free turbine. Auxiliary plots presented in Fig. 18 show variation of gas-generator turbine pressure ratio with turbine-inlet temperature and variation of equivalent weight flow to the free turbine with turbine-inlet temperature.

We see from Fig. 18 that varying inlet temperature results in a large variation of pressure ratio across the gas-generator turbine and of inlet equivalent weight flow to the free turbine. If constant exhaust-nozzle area operation is specified (Fig. 17), variation of pressure ratio across the free turbine is also large.

The variation becomes even greater if the same exhaust-nozzle area prevails at altitude. This is apparent from Fig. 19, which is similar to Fig. 17, but for flight conditions of 600 mph and 40,000 ft. We see from the auxiliary plots in Fig. 20 that the range of pressure ratio across the gas-generator turbine and of inlet equivalent weight flow to the free turbine are also increased by flight at the altitude condition. Variation in pressure ratio across the free turbine could be reduced by closing down the exhaust-nozzle area at altitude. Complete elimination of pressure ratio variation, however, could require reduction in exhaust-nozzle area of over 50%. From this abbreviated discussion it is apparent that turbine design requirements for the free-turbine engine are more severe than those of the single-spool engine. In addition, the problem of matching the two turbines for good overall performance complicates turbine design.

The range of equivalent weight flow the free turbine can accommodate is restricted as discussed above. This in turn restricts the range of turbine-inlet temperature over which the engine can operate. Consider the limiting case when the first turbine stator area of the free turbine is fixed. For a fixed stator area inlet equivalent weight flow to the free turbine and

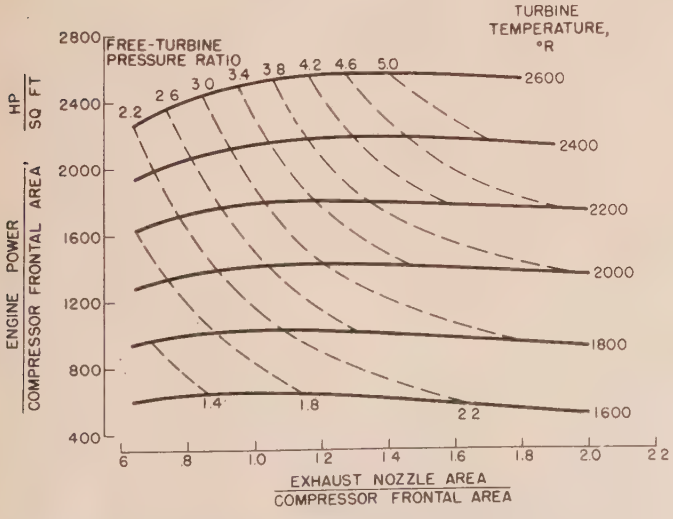


Fig. 19—Free-turbine engine requirements at 600 mph and 40,000 ft. Turbine temperature and exhaust area effects

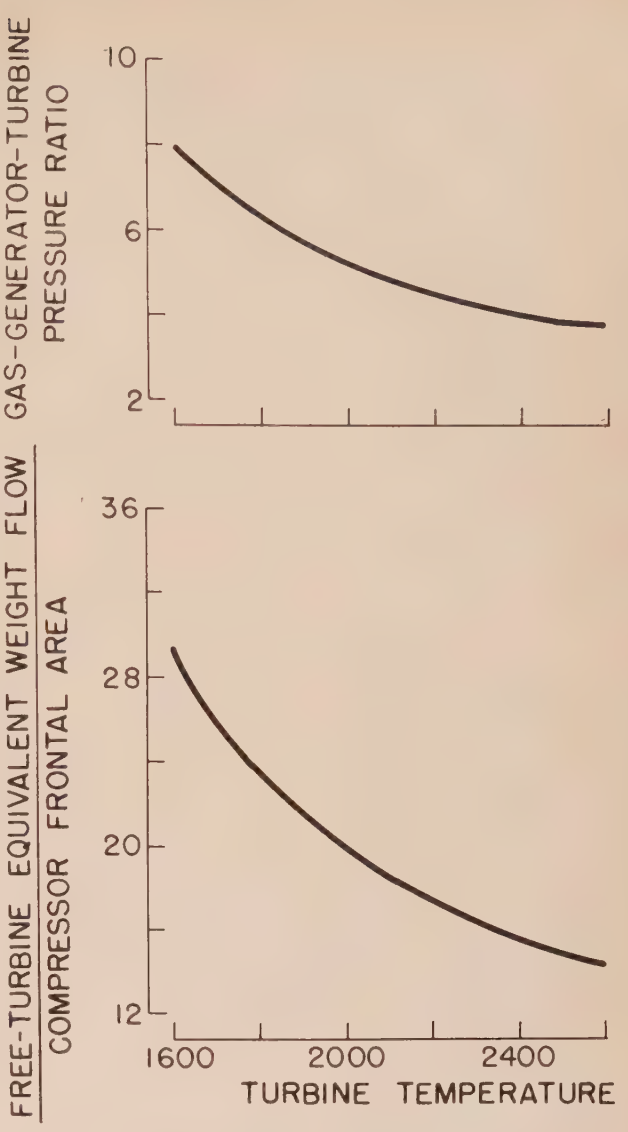


Fig. 20—Free-turbine engine requirements at 600 mph and 40,000 ft. Turbine temperature effects

pressure ratio across the gas-generator turbine are constant for all flight conditions. As an example of how this restricts turbine-inlet temperature Figs. 18 and 20 show that an engine designed for a 2300 R inlet temperature at sea-level static conditions could operate only at an inlet temperature of 2140 R at 600 mph and 40,000 ft. If an increase in turbine-inlet temperature is desired at altitude to obtain more power and lower sfc, it could be obtained by increasing rotative speed of the gas-generator unit of the engine. However, the increase in turbine-inlet temperature that can be obtained in this manner is restricted by increasing blade stress and deterioration of compressor performance. Another alternative would be to design for a high turbine-inlet temperature at 600 mph and 40,000 ft and then derate the engine at sea-level static by dropping engine rotative

speed and not utilizing the maximum temperature possible.

The free-turbine engine, then, is restricted in the range of turbine-inlet temperature over which it can operate and is, therefore, less versatile than the single-spool engine. This restriction is greatest when no adjustment of the free-turbine stator areas is permitted. In general, turbine design requirements of the free-turbine engine are also more severe than for a single-spool engine. Advantages of the free-turbine engine stem from the free-turbine rotative speed being independent of the gas-generator unit rotative speed and from only the free turbine being connected to the propeller. An independent rotative speed, for example, could be used to reduce turbine stresses. In addition, for other applications such as in a helicopter where range of altitudes and flight speeds is small, the free-turbine engine may be more suitable than the single-spool engine.

Two-Spool Engine

As mentioned previously, it is desirable from the standpoint of reducing sfc to go to higher compressor pressure ratios. These compressor pressure ratios may eventually be pushed as high as 20 or 25. Design compressor pressure ratio for the single-spool compressor performance used in the single-spool and free-turbine engine analyses was 14.4. In order to obtain pressure ratios this great in a single-spool compressor, such undesirable characteristics as a deep knee in the compressor surge line may be encountered, with the attendant difficulty of accelerating the engine from idle to design rotative speed. In addition, a single shaft connected to the propeller through a gear box may be undesirable with number and size of compressor and turbine stages increased. An arrangement of this sort could result in structural and propeller operating difficulties. In a two-spool unit the amount of compressor and turbine on any one shaft (in particular, the shaft connected to the propeller through the gear box), is greately reduced. Because the designer also has freedom in a two-spool unit to select rotative speed of the inner compressor as well as the outer compressor, it may be possible to design a two-spool unit lighter than a single-spool unit of the same pressure ratio. It appears likely, in any case, that for pressure ratios of 12 to 15, the two-spool turboprop will also be considered if it does not have some undesirable characteristic. For pressure ratios above these it appears that only the two-spool compressor will be considered.

A preliminary analysis was made for a two-spool turboprop to determine its operating characteristics. An overall design compressor pressure ratio of 12 was assigned and pressure ratio splits between the outer and inner compressors of 6–2, 3–4, and 2–6 were considered. In the analysis, equivalent weight flow to both inner and outer turbines was allowed to vary in order to obtain the most flexible engine arrangement possible and, thus, the lowest sfc. The effect of fixing either or both the equivalent weight flow to the inner and outer turbine could also be determined from this analysis. The mode of operation specified was constant design rotative speed for the outer compressor.

Improvement in sfc obtained for the 6–2 engine by adjusting either or both inner and outer turbine stators was small. (Allowing the turbine-inlet equivalent weight flow to vary implies stator adjustment.) Even with fixed stator areas, the 6–2 engine was capable of operating over a wide range of turbine-inlet temperature (1600–2600 R), at the three flight conditions analyzed. These flight conditions were: sea-level static, 400 mph at 19,000 ft, and 600 mph at 40,000 ft. Fixing stator areas for the 2–6 engine resulted in higher sfc at some turbine-inlet temperatures. In addition, the 2–6 engine without stator adjustment was not able to operate over the range of turbine-inlet temperature the 6–2 engine could. The 3–4 engine had characteristics between those of the 6–2 and 2–6 engines.

An analogy can be drawn between the single-spool and free-turbine engines analyzed and the two-spool engines of different pressure ratio splits. The 6–2 engine had characteristics very similar to the single-spool engine while the 2–6 engine had characteristics similar to those of the free-turbine engine.

Unrestricted variation of turbine-inlet temperature could be achieved for any of the pressure ratio splits by adjustment of stators of both turbines or by adjustment of only the outer turbine stators. Sfc obtained when only outer turbine stators were adjusted was nearly as low as when the stators of both turbines were adjusted. With such an arrangement the 6–2 engine had less difficult turbine design requirements than the 2–6 engine and the analogy given previously could again be drawn.

Stress and frontal area problems were not considered in the preliminary analyses. However, turbine frontal area and stress of the outer turbine as well as the pressure ratio variation across it could be reduced by use of an adjustable exhaust-nozzle area.

Conclusion

Matching studies of three turboprop engine configurations were made for flight conditions ranging from sea-level static to flight at 600 mph and 40,000 ft. A range of turbine-inlet temperatures and various modes of engine operation were considered. From these analyses it was concluded that:

1. An adjustable exhaust-nozzle area is desirable because it makes turbine frontal area, stress, and pressure ratio requirements less critical without penalizing engine performance.

2. Engine sfc depends primarily on flight conditions and turbine-inlet temperature, with mode of operation playing secondary role. Lowest sfc is ob-

tained at full power or maximum turbine-inlet temperature and at highest flight velocity and altitude.

3. A single-spool engine has less critical turbine design requirements than a free-turbine engine.

4. For the flight conditions considered, the free-turbine engine is restricted in the range of turbine-inlet temperature over which it can operate if adjustment of the free-turbine stator areas is not permitted.

5. Nearly minimum sfc and operation over a wide range of turbine-inlet temperature was obtained for a two-spool engine without turbine stator adjustment, if the compressor pressure ratio of 12 was split 6–2 (outer-compressor pressure ratio of 6 and inner-compressor pressure ratio of 2).

6. Adjustment of at least the outer turbine stators of the 2–6 two-spool engine was necessary in order to obtain the sfc and range of turbine-inlet temperature possible in the 6–2 engine without stator adjustment. With such an arrangement the 2–6 engine has much more critical turbine design requirements than the 6–2 engine.

7. Increasing the outer compressor pressure ratio of a two-spool engine results in characteristics similar to the single-spool engine, while reversing the split results in characteristics similar to the free-turbine engine.

ORAL DISCUSSION

Reported by D. Q. Marshall

General Electric Corp.

Question from floor: It appears a single-spool engine is most desirable. Is a 14.4/1 pressure ratio realistic at this time?

Mr. Davison: It is difficult to select one engine as being most desirable without specifying a particular application and range of operating conditions. In addition, there are many other considerations not taken up in this paper which could dictate the type of engine selected for a particular application. A 14.4/1 pressure ratio is not unrealistic in the sense that a compressor could probably be designed to obtain this pressure ratio. However, the compressor might have undesirable characteristics such as a deep knee in the stall line.

Bernard Jeavens, Studebaker-Packard Corp.: How realistic are the turbine-inlet temperatures?

Mr. Davison: In general, higher turbine-inlet temperatures are desirable in order to obtain lower sfc. The maximum temperature that could be employed before turbine cooling is required depends on a number of factors such as compressor pressure ratio, turbine design limits employed, stress-rupture limit, or turbine life specified. For the single-spool engines of this paper, it appeared that, optimistically, a turbine-inlet temperature of about 2500 R could be employed without exceeding a 100-hr stress-rupture limit. It might also be noted that the maximum turbine-inlet temperature that could be employed in some current engines without turbine cooling could be pushed up by designing the turbine to more critical aerodynamic limits than generally employed at present. Whether or not this would be more desirable than a conservative turbine aerodynamic design with turbine cooling would depend on the respective penalties involved. In addition, the reduction in sfc obtained with increasing turbine-inlet temperature becomes negligible at the higher temperature although the power per unit frontal area increase is nearly linear. It might be more desirable, therefore, to limit the turbine-inlet temperature and increase engine size, if more power is required. These are only a few of the considerations and compromises that are involved in any engine design. It is, therefore, difficult to specify truly realistic or universal turbine-inlet temperature limits.

Dan Beard, American Airlines, Inc.: I am much interested in the paper since American Airlines is committed to a simple turbine for the Electra. The single spool has operating limitations which we would like to overcome. The constant speed characteristic creates a lot of noise on the ground due to high propeller-tip speed. I think this could be overcome with a free-turbine design. I also believed that sfc in the holding flight condition would be better with a free turbine. Admittedly, the free-turbine design has a runaway disadvantage which the British have overcome by using an overrunning clutch connected to the high pressure rotor. I believe 500 knots would be the practical speed because of propeller-tip speed limitations due to noise.

Mr. Davison: Regardless of the top speed selected, it appears within the reservations stated in the paper, that the lowest sfc is obtained at the highest flight velocity and altitude. It would, therefore, appear desirable to design and fly an airplane to take advantage of these turboprop-engine characteristics. However, as also pointed out in the paper, the minimum use of fuel for a given time (as when an airplane is loitering) would be obtained at some lower flight velocity and altitude. If a flight velocity and altitude less than 600 mph and 40,000 ft, respectively, had been considered, the other trends noted in the paper would not be changed. They might, however, have less significance.

P. T. Vickers, General Motors Research: Was cooling considered on the curve of turbine-inlet temperature versus turbine-blade temperature?

Mr. Davison: Cooling was not considered in the analysis. Higher compressor pressure ratios are contemplated for turboprop engines than for turbojet engines and larger turbine pressure drops are taken in the turboprop engines. The result for a turboprop engine is low stress in the initial stage and high stress in the exit stage accompanied by low temperatures. In general, therefore, higher exit stresses can be tolerated in a turboprop engine than normally considered possible in a turbojet engine.

F. E. Carroll, United Aircraft Products: Were propeller-tip speeds considered?

Mr. Davison: I did not consider propeller-tip speeds as such. However, a flight velocity of 600 mph at a 40,000-ft altitude corresponds to a flight Mach number of approximately 0.9. It seemed reasonable to me to assume that propellers could be developed eventually to operate with reasonably good efficiency at this Mach number. In addition, the sfc values presented in the paper are based on engine power which is the sum of the shaft power to the propeller plus the equivalent shaft power of the net jet thrust. A propeller efficiency must be assumed in converting net thrust to equivalent shaft power. However, the assumed value of propeller efficiency doesn't have a critical effect on the sfc values presented here because the equivalent shaft horsepower of the net jet thrust is usually small in comparison to the shaft horsepower.

THE NEW DYNAFLOW

THE automotive transmission engineer is confronted with the problem of best utilizing engine power and doing it in a manner which is most pleasing to the car-buying public. Some of the items which have to be considered are car performance, fuel economy, smoothness of operation, and a minimum manipulation of controls by the driver.

The hydrokinetic torque converter has filled this order in a more-than-satisfactory manner. This has been evidenced by the number of new automatic transmissions which have been placed on the market since introducing our 1948 Dynaflow transmission.

To explain briefly the operation of a hydrokinetic torque converter, let us start with a fluid coupling (Fig. 1). Here, the input or pump member having radial vanes or blades is driven by the engine. This member functions like a simple centrifugal pump, moving oil in an outward direction, thereby causing a vortex flow, as shown by the arrows. This oil contains kinetic energy because of its mass and velocity. As this oil enters the output member, which is stationary or turning at a slower speed, it gives up

BUICK's 1956 twin-turbine Dynaflow transmission incorporates a fixed-blade stator between the two turbine elements.

Added to the twin turbine (introduced in 1953) and the variable-pitch stator (introduced in 1955), the author says, this has made an important contribution to reduction in engine rpm, increased low-end performance, and improved efficiency.

Stall ratio has been increased from 2.45/1 to 3.5/1, and "getaway" on low-end car performance has improved.

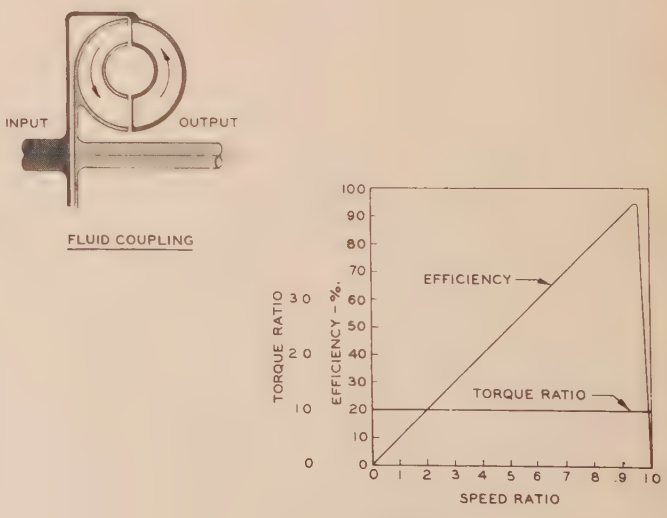


Fig. 1—Fluid coupling

AUTOMATIC TRANSMISSION

R. J. Gorsky, Buick Motor Division, GMC

This paper was presented at the SAE Annual Meeting, Detroit, Jan. 12, 1956.

kinetic energy. By doing so, it imposes a driving force or torque on the driven member. The oil then leaves the driven member, re-enters the driving member, where it is re-energized, and this cycle continues to repeat itself.

The torque ratio and efficiency curves for a fluid coupling show that the torque ratio is always 1/1 (in other words, it does not multiply the input torque), and the efficiency curve is inversely proportional to the slip taking place. As the relative speed of the elements approaches 1/1, the efficiency again falls off to zero, because no torque can be transferred without vortex flow.

The first use of fluid couplings on American built cars was in the 1930's. It is their characteristic to transfer very little torque at low speed, and to slip only a small amount at higher speeds. This enables a vehicle to be brought to a stop without stalling the engine, and thereby permits the elimination of the clutch pedal. Since it cannot multiply torque, it must be used in conjunction with gears to provide satisfactory car performance.

The addition of a stator member between the driven member and the driving member results in a unit which can multiply torque (Fig. 2). This is a three-element torque converter. The blades of the driven or turbine member have a reverse bend to them, so that the oil leaving this member has a reversed direction of spinning velocity. The direction of this spinning oil is changed to a forward direction by the stator member. At cruising speeds, this stator member would form a serious obstruction to the vortex flow of oil if it were not permitted to overrun. To preclude this condition, an overrunning clutch can be used, which permits the stator member to overrun. When this occurs, the three-element torque converter will function in a manner similar to the fluid coupling. As can be seen from the diagram showing typical torque ratio and efficiency curves for a torque converter superimposed on the fluid coupling curves, a torque ratio of more than 1/1 can be obtained (or in other words a torque converter can multiply torque), and the efficiency curve rises faster, peaks, and falls off until it reaches the fluid coupling curve;

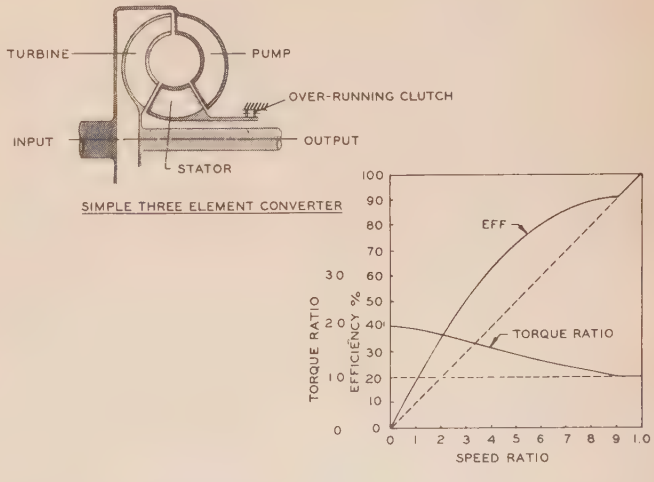


Fig. 2—Three-element torque converter, achieved by adding a stator member between driven member and driving member

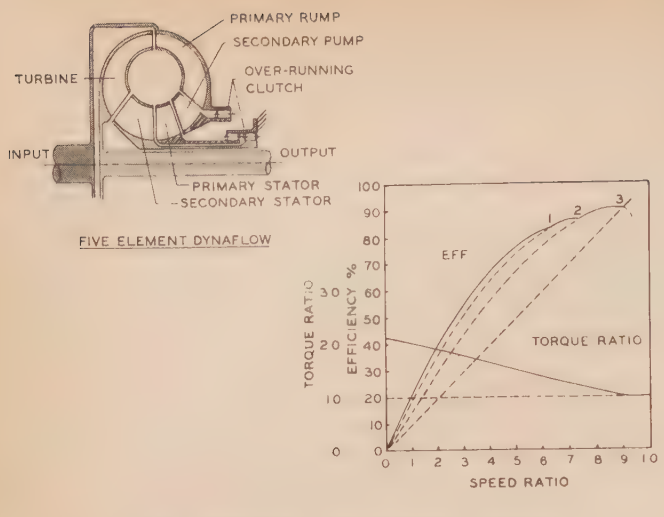


Fig. 3—Original, five-element Dynaflow

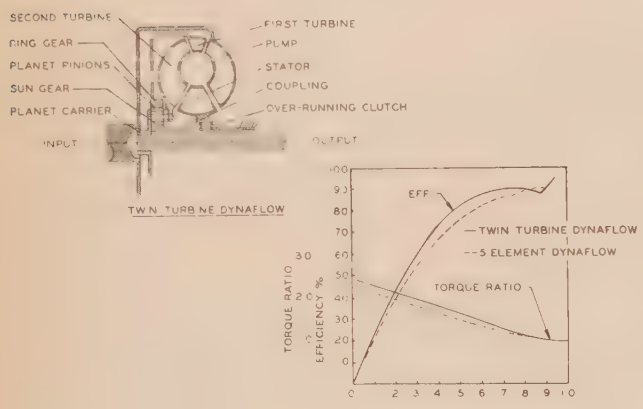


Fig. 4—Twin-turbine Dynaflow introduced in 1953

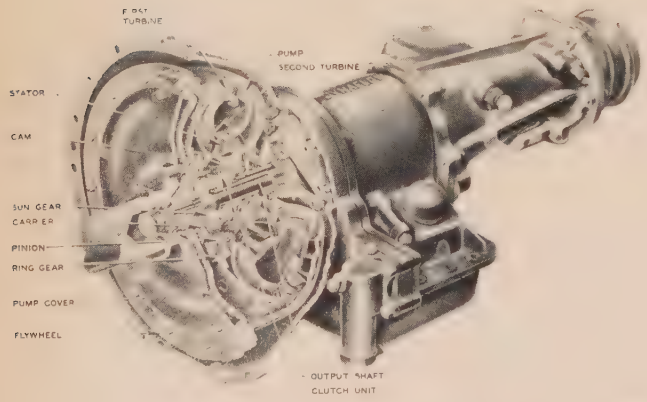


Fig. 5—1954 twin-turbine Dynaflow

at this point, the stator element free wheels, and the unit functions in a manner characteristic of a fluid coupling.

Different converter designs will produce different torque ratio and efficiency curves. The original Dynaflow torque converter, which consisted of a primary pump, a secondary pump (which was mounted on an overrunning clutch), a turbine, and two stators

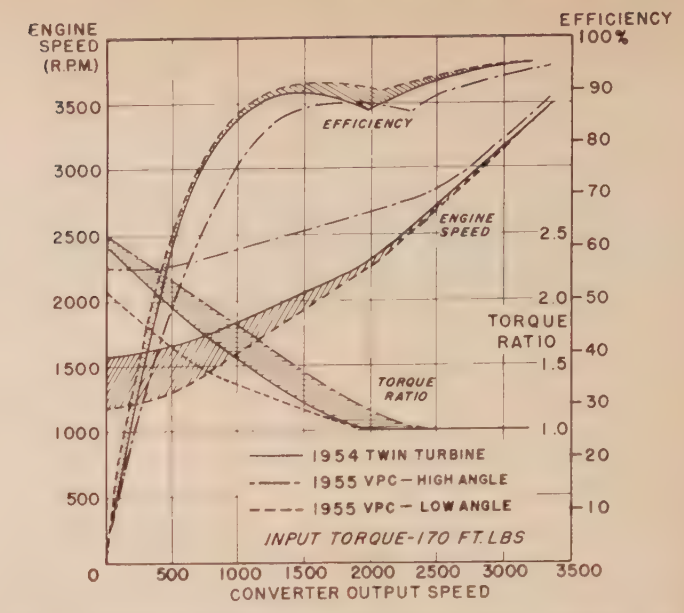


Fig. 6—Comparison between 1954 twin-turbine converter and 1955 variable-pitch converter

(which also were mounted on overrunning clutches) was, in a sense, three torque converters and a fluid coupling combined into a single unit (Fig. 3). This resulted in four sets of curves, the most desirable portions of which were utilized. The break points 1, 2, and 3 in the efficiency curve are the points at which the secondary pump stops overrunning the primary pump, the secondary stator starts to free wheel, and the primary stator starts to free wheel, respectively.

To improve the car performance and converter efficiency, and to reduce the engine speed for a given car speed, the twin-turbine Dynaflow transmission was introduced on our 1953 models (Fig. 4).

This design utilizes the torque-multiplying characteristics of a planetary gear set in conjunction with the torque-multiplying ability of a hydrokinetic torque converter, and retains the smooth, uninterrupted power flow characteristic of a fluid torque converter. The arrangement of the twin-turbine Dynaflow torque converter is unique in that all the power transferred is through the gear set and first turbine at low speeds. The torque gradually and smoothly diminishes as the second turbine torque increases, until it does all the work at higher speeds.

Let us take time out to go over the nomenclature of the twin-turbine parts (Fig. 5). The converter pump is driven by the engine and energizes the converter fluid. The first turbine receives the fluid leaving the pump, and drives the planetary ring gear through a disc member which is riveted to the ring gear. The second turbine is doweled and bolted to the planet carrier, which in turn is splined to the converter output shaft. The stator is mounted on an overrunning clutch which is common to it and to the planetary sun gear.

Torque converter engineers have long dreamed

about variable-pitch blades. For some time, they have been commercially used for airplane and boat propellers. In the 1955 Dynaflow, we introduced a converter having a variable-pitch stator blade. This converter is similar to the one used during the 1953 and 1954 model years. The principal revision was in the stator, changed from a fixed-blade member to a variable-pitch blade assembly. By controlling the pitch of the stator blades, the best angle for cruising and economy can be obtained, as well as the best angle for performance. With the fixed-stator blade, a compromise angle had to be used.

As can be seen from the 1954 versus 1955 converter comparison, the 1955 transmission has two completely different sets of characteristics (Fig. 6). The solid lines show the 1954 twin-turbine converter. The dot-dash lines show the 1955 variable-pitch twin-turbine at high angle, and the dash lines show the 1955 variable-pitch twin-turbine at low angle. The shaded areas show how the engine speed at low angle was reduced, how the efficiency was improved, and how the torque ratio or performance was increased (at high stator-blade angle).

To further increase the low-end car performance, a fixed-blade stator, located between the first turbine and second turbine, has been added to the 1956 twin-turbine Dynaflow (Fig. 7).

A developed view of the converter element blades shows how the changing direction of oil flow through the converter imposes forces on the various elements (Fig. 8). The pump member energizes the oil and causes it to flow to the first turbine member. Here its flow direction changes from the entrance end to the exit end of the blade, causing a driving force to be imposed on this member. This driving torque is multiplied by the converter planetary gear set before it is fed into the converter output shaft. The fixed-blade stator member then redirects the oil flow so that it will enter the second turbine member at an angle which will provide a driving torque to this member. The second turbine is splined directly to the converter output shaft, and transmits torque supplementing the first-turbine torque during acceleration. At higher car speeds and at cruising, the second turbine transmits all of the driving force.

The oil leaving the second turbine then enters the variable-pitch stator, where it is redirected before re-entering the converter pump member to repeat this cycle. The magnitude of directional change is greater at high stator-blade angle than at low stator-blade angle, which results in more converter ratio or car performance.

This diagram shows the flow at a "stall" condition, in which all members except the pump are stationary. The direction of oil flow changes as the power being transmitted and the car speed vary. This causes the turbines to increase in rotational speed and the stators to start overrunning.

This change in oil flow can be illustrated by using vector diagrams (Fig. 9)

With a low member speed, as shown by S_1, and a high vortex flow, as shown by F_1, the vector sum will be *O-A*. With a high member speed, S_3, and a low vortex flow, F_3, the vector sum will be *O-C*.

The length and direction of the vector sum, or absolute velocity of the oil, will change as the conditions vary. The vectors shown are for purposes of

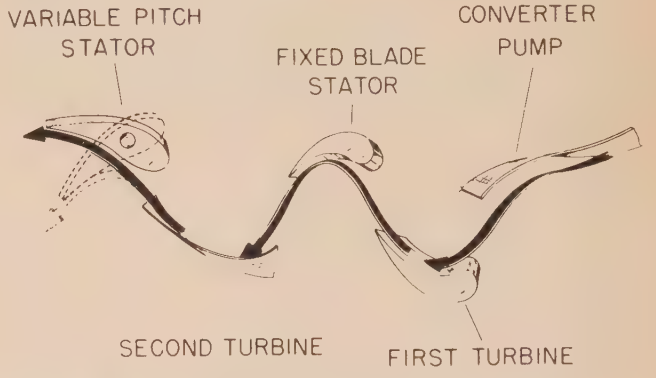


Fig. 8—Developed view of converter oil flow

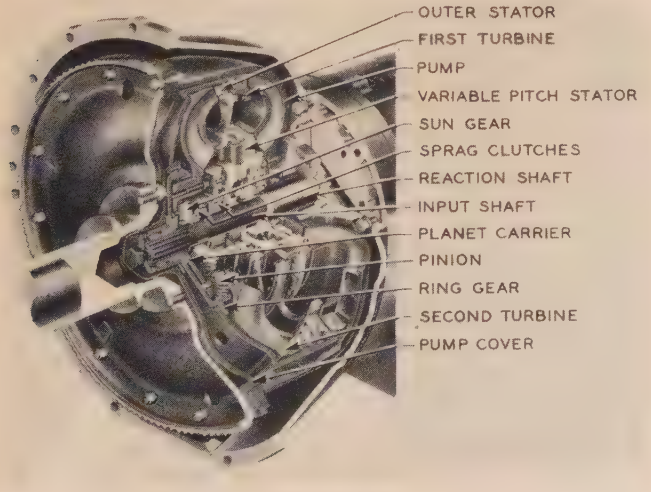


Fig. 7—1956 twin-turbine Dynaflow, wtih fixed-blade stator added between first and second turbine

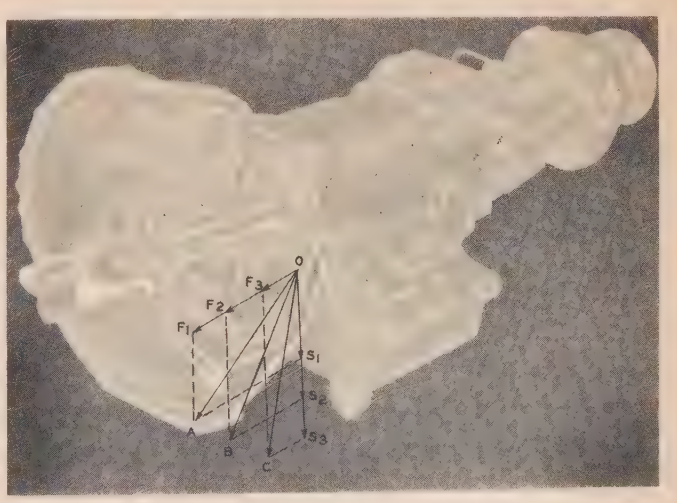


Fig. 9—Pump vector analysis

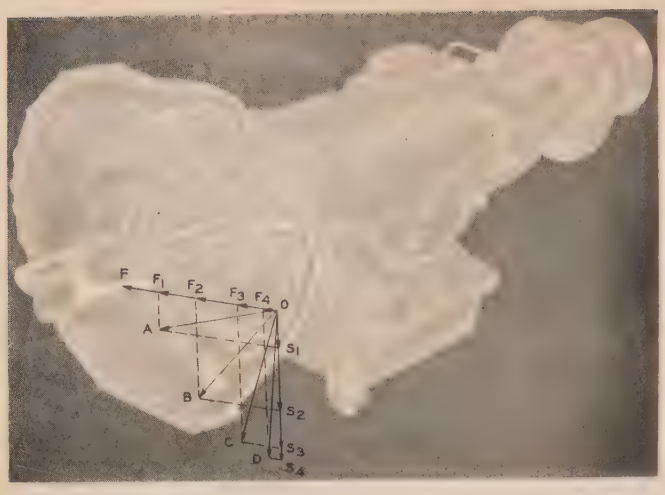


Fig. 10—First turbine vector analysis

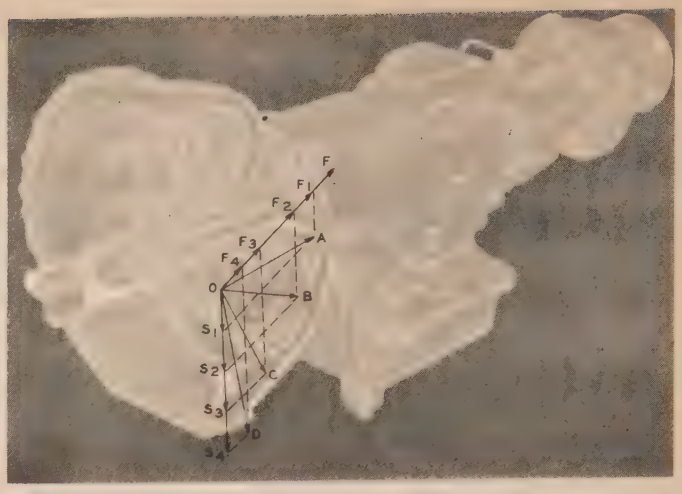


Fig. 12—Second turbine vector analysis

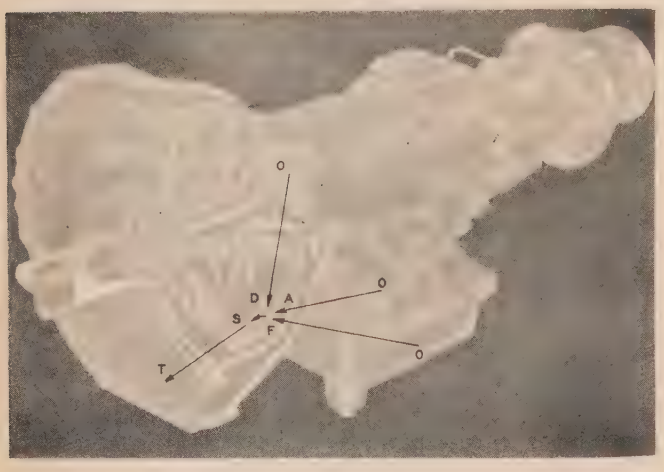


Fig. 11—Fixed-blade stator vector analysis

Fig. 13—Variable pitch stator vector analysis (low pitch shown)

illustration, and do not necessarily represent true values. The absolute velocity of a particle of oil will be the vector sum of the velocity of the converter member it is leaving, which will be a tangential vector, plus the vortex velocity of the oil in relation to the blade it is leaving. The latter will have a direction parallel to the exit end of the blade. If the member is stationary, the absolute velocity will be the same as the vortex velocity.

As previously mentioned, the pump operates like a simple centrifugal pump, and the absolute velocity of the fluid will be the vector sum of the circumferential speed (S_1) of the pump blade which drives the fluid, plus the linear speed (F_1) of the fluid in relation to the blade. As the car speed increases, the fluid flow decreases; the vector sum will change in direction and magnitude as shown by vectors *O-A*, *O-B*, and *O-C*. Because of its mass and velocity, this fluid contains energy, which it obtained from the engine.

When a converter element changes the direction of flow, this member will have a force imparted to it. A vector diagram showing the oil leaving the first turbine blades will show the amount of directional change of the oil flow (between the pump exit and the first turbine exit), and will consequently indicate the force imparted to this member (Fig. 10).

At stall, the turbine speed is zero; the direction of oil flow will be tangent to the exit end of this blade, as shown at *F*. This results in the largest change in direction between the entering oil and the leaving oil; consequently, it produces the greatest driving force on this member.

The difference in directional change decreases as the first turbine speed increases, which is shown by the resultant vectors *O-A*, *O-B*, *O-C*, and *O-D*. At *D*, the high rotational speed of the first turbine causes the oil to flow between its blades without being changed in direction. At this point it is delivering no torque, and the second turbine blades do all the driving.

The direction of oil flow leaving the fixed-blade stator will be tangent to the exit end, which is shown by vector *S-T* as long as the oil enters at an angle such as *O-A* or *O-F* (Fig. 11). When the direction of oil flow leaving the first turbine changes to an angle such as *O-D*, the fluid impinges on the back of the stator blade, and the member starts to overrun. The direction of oil flow leaving the stator will then be the same as that entering the stator, namely *O-D*.

The second turbine has curved blades and has a

smaller diameter at its exit end than its entrance end, which further enables it to extract energy from the flowing oil (Fig. 12). This member functions in the same manner as the turbine in the previously mentioned three-element torque converter. The exit oil flow is shown by vector sums *O-A*, *O-B*, *O-C*, and *O-D*.

By virtue of the converter planetary gear set which multiplies the first turbine torque by a ratio of 1.6/1, the speed of this member will be 1.6/1 faster than that of the second turbine. It is this relationship which enables the first turbine to reduce its load-carrying ability while the second turbine increases its load-carrying ability. The *difference* in their speeds is zero at stall, and increases as the output shaft speed increases. This feature enables the first turbine to move ahead of the second turbine, so that the latter can take the driving force of the fluid and deliver it directly to the converter output shaft.

The direction and magnitude of the oil flow leaving any converter member is the same as that entering the adjacent member. Therefore, the vectors *O-F*, *O-B*, and *O-D* shown entering the stator are the same as those just mentioned as leaving the second turbine (Fig. 13). As the direction of flow changes from *O-F* to *O-B* to *O-D* when the car speed increases, a diminishing torque is imposed on the stator blades. When the oil impinges on the back side of the stator blades, as shown at *D*, the stator free wheels along with the oil, so that it will not obstruct the fluid flow.

The direction of the oil flow leaving the variable-pitch stator will change when the blade is shifted from low to high-angle position (Fig. 14). The high-angle position will result in a greater torque reaction on this member, and consequently a higher converter ratio.

To accomplish the addition of a twin turbine, a variable-pitch stator, and a fixed-blade stator, all of which resulted in additional manufacturing costs, considerable stress was put on the most economical design and on the method of parts manufacture.

A variety of materials and manufacturing methods is used in making the variable-pitch stator and overrunning clutch assembly (Fig. 15). The stator overrunning clutch is of the cam and roller design. Accordion-pleated-type roller energizing springs are used.

The rear stator blade carrier is made of a shell-molded iron casting (Fig. 16). This part is rough machined, removing 0.030–0.045 in. from all surfaces.

The front stator blade carrier is also made of a shell-molded iron casting (Fig. 17). This part contains the support bearings, which are steel-backed bronze bushings.

The front and rear carrier halves are bolted and doweled together (Fig. 18). This matched assembly is then finished-machined, removing 0.015-0.020 in. for finish cuts. The 20 holes for the stator-blade cranks are drilled and reamed to a tolerance of 0.002

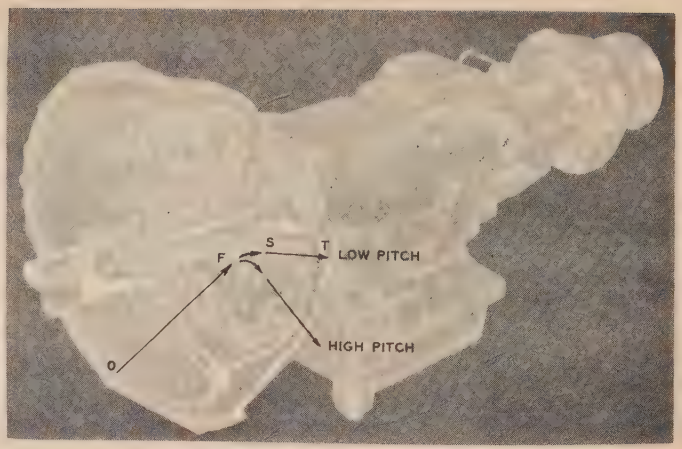


Fig. 14—Variable pitch stator oil flow

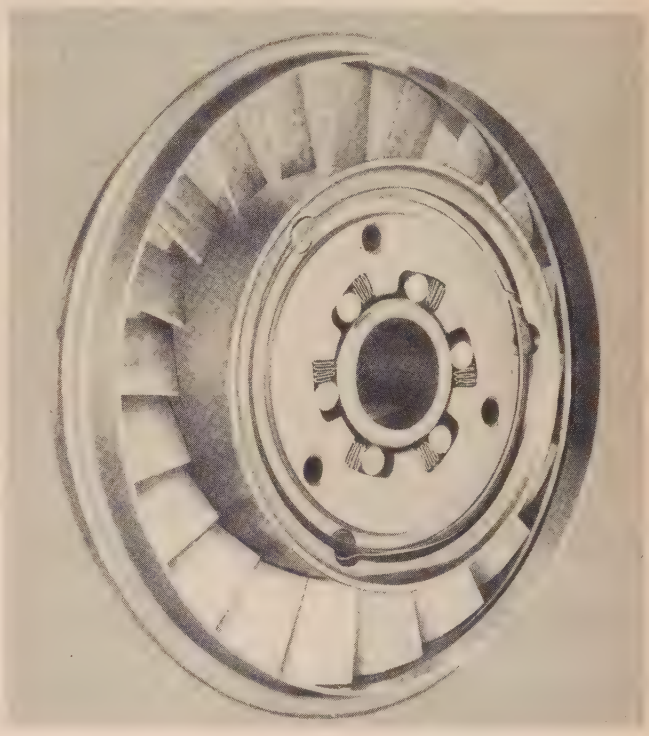


Fig. 15—Stator and overrunning clutch

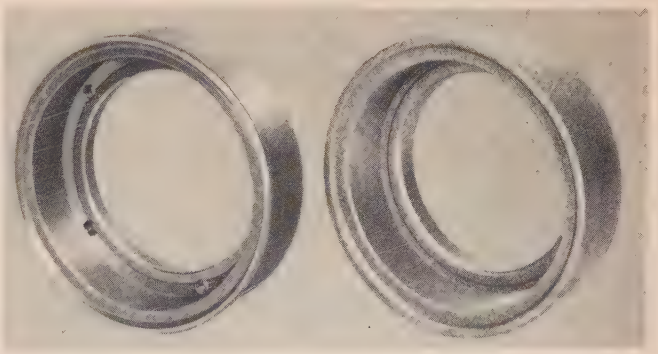


Fig. 16—Variable-pitch stator carrier, rear. Left: casting; right: semifinished

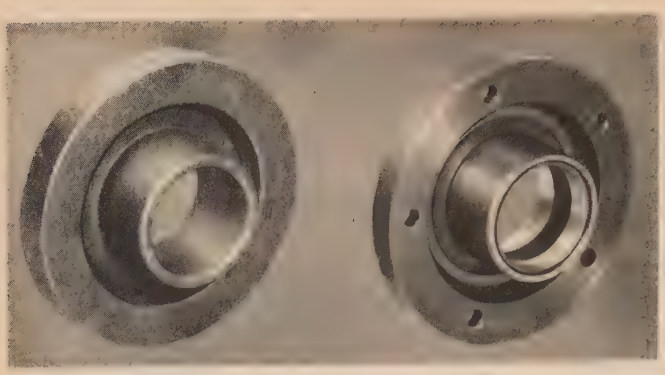


Fig. 17—Variable pitch stator carrier, front. Left: casting; right semifinished

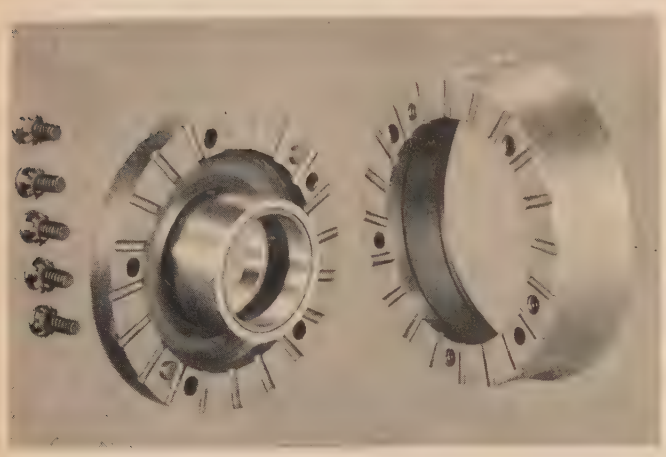


Fig. 18—Stator blade carrier. Left: front; right: rear

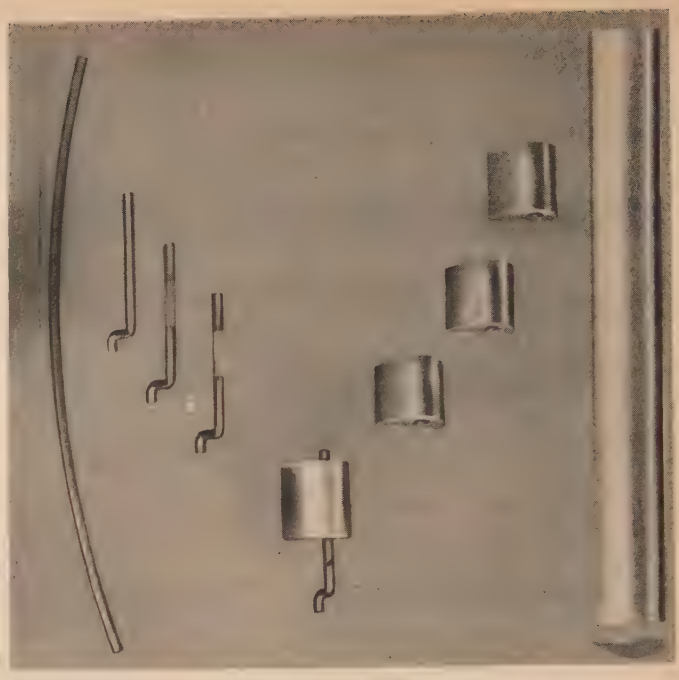


Fig. 19—Stator blade assembly

in. The centerlines of these holes are in the parting-line plane of the two carrier sections.

A close hole tolerance is required to minimize oil leakage between the cranks and the crank bearings. The centerline parting permits the use of a crank and blade subassembly, which facilitates the initial assembly and any service disassembly.

The stator-blade crank is made from close-tolerance SAE 1018 drawn wire of 0.1540–0.1550-in. diameter (Fig. 19). The offset for the crank is made in a four-slide machine. The serrations which lock the stator blade to the crank are formed by rolling. This part is then heat-treated to increase its strength and wear resistance.

Several types of variable-pitch stator blade, including sintered powdered iron, die-cast, formed sheet steel, and extruded aluminum were considered. They all had their shortcomings in regard to cost, durability, lack of uniformity, and performance characteristics. A smooth surface, a thin trailing end, and practically no variation from part to part were required. A hot aluminum extrusion, cold-drawn to maintain uniformity, filled the requirements the best. The cross-section of this cold-drawn stock is being maintained within 0.002 in. of the exact contour shown on Fig. 19. The stator blades are cut from these hot-extruded, cold-drawn bars. The ends are broached to form a small thrust area, and to give a length of 1.061 ± 0.001 in. The crank hole is drilled and reamed to 0.1557–0.1567-in. diameter.

The throw of the crank is positioned in its proper relation to the blade, and the crank is pressed into place. The interference fit between the serration on the crank and the hole in the blade locks these two parts together.

The stator piston or crank actuator is a sintered-powdered iron part (Fig. 20). Labyrinth grooves are machined on the ID to provide oil sealing, and a groove is turned on OD to accommodate a cast-iron oil-sealing ring. The weight decrease of this part during machining amounts to only 23%. Most of this material is removed to form the grooves, which cannot be readily cast without encountering back drafts or an extremely complicated die.

The outer ring or shroud is a sheet-metal stamping (Fig. 21). First, a cup is formed from which the draw flange and the bottom are trimmed off, leaving a cylinder of seamless tubing. This cylinder is then placed over a plug and pressed axially between form-controlling dies. This causes the center section to bulge out and get squeezed together, thereby forming a very rigid "T" section. The holes which accommodate the outer ends of the crank are pierced in a radial outward direction, so that a smooth surface will be retained on the ID for the stator-blade thrust area. No machining is done on this part. A machined die casting was initially considered for this part, until an analysis showed that it would have cost appreciably more than the stamped part.

The rear stator-blade carrier, the stator blade and crank assemblies, the stator piston, and the outer shroud are assembled before the front stator-blade

carrier is bolted on to complete the assembly of this unit (Fig. 22).

Under normal acceleration and cruising, the stator vanes will be in "low angle," or nearly parallel to the centerline of the converter. This provides the largest opening between the vanes for freedom of oil movement through the stator, and a minimum amount of redirecting of the converter oil flow. The "low angle" position gives less engine speed and more efficient converter operation.

Under performance operation, the stator vanes will be in "high angle," or like a venetian blind partially closed. This provides the greatest redirecting of the converter oil flow, and a resultant higher converter torque ratio. It also permits the engine speed to increase, so that more power is developed to feed into the converter.

In this manner, the driver can adjust the converter to better meet his driving conditions. The stator piston moves in response to oil directed from the stator control valve. This control valve is operated mechanically by linkage from the throttle control mechanism. An added resistance at the end of throttle pedal movement is used, so that the driver will realize that further movement will put the transmission into performance range.

Fig. 20—Stator piston

The transition from cruise performance to full-throttle performance is very smooth, because it is accomplished by changing the converter oil flow instead of by shifting clutches and bands. The increase in performance from low vane angle to high vane angle is greatest at low car speeds, and gradually diminishes as the converter coupling range is approached. This provides a safety feature to overtake other vehicles by "switch the pitch" performance.

A sprag-type overrunning clutch is used with the converter planetary sun gear (Fig. 23). To maintain concentricity between the outer and inner races, a bearing is pressed into the front end of the sun gear

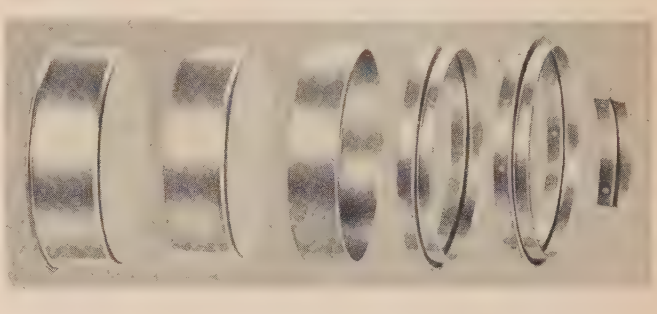


Fig. 21—Stator outer ring

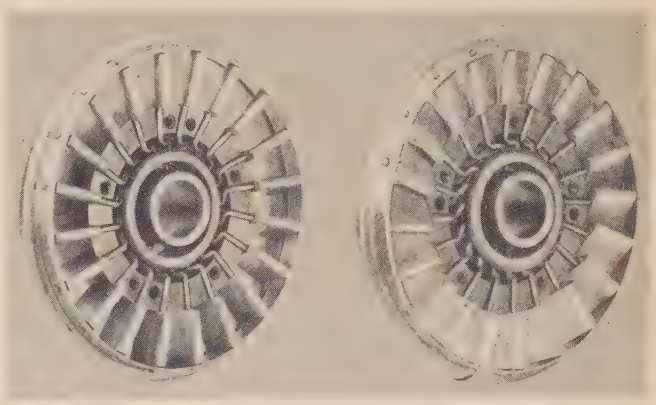


Fig. 22—Variable-pitch stator blade angle. Left: low; right: high

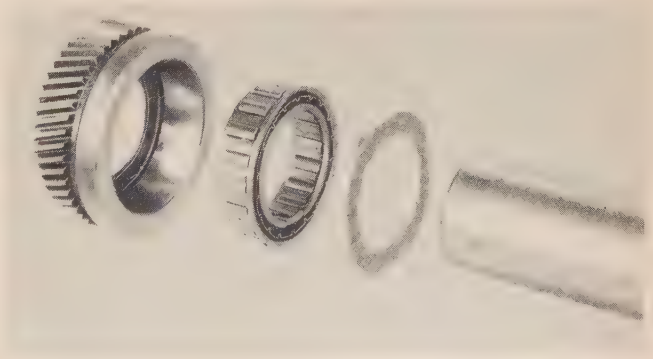


Fig. 23—Sun gear and overrunning clutch

and bored to size, locating from the ID of the sun gear raceway. To the rear is assembled a narrow bronze bearing having very close diametral tolerances and concentricity.

The overrunning unit is a double-cage, full-phasing sprag clutch (Fig. 24). One of the features of this type of overrunning clutch, in comparison to the cam and roller type, is that it requires less space. With five converter elements together with a planetary gear set and a variable-pitch stator, space limitations have become a very serious problem.

The first turbine is a semipermanent-mold aluminum casting (Fig. 25). Early 1955 parts were made using individual plaster cores for each opening between the blades. A flash resulted in the casting, where these cores mated together (Fig. 26). The flash on this cast element proved to be detrimental to converter performance, and no satisfactory way was

Fig. 24—Overrunning clutch

Fig. 26—First turbine, showing flash caused by use of individual plaster cores for each opening between blades

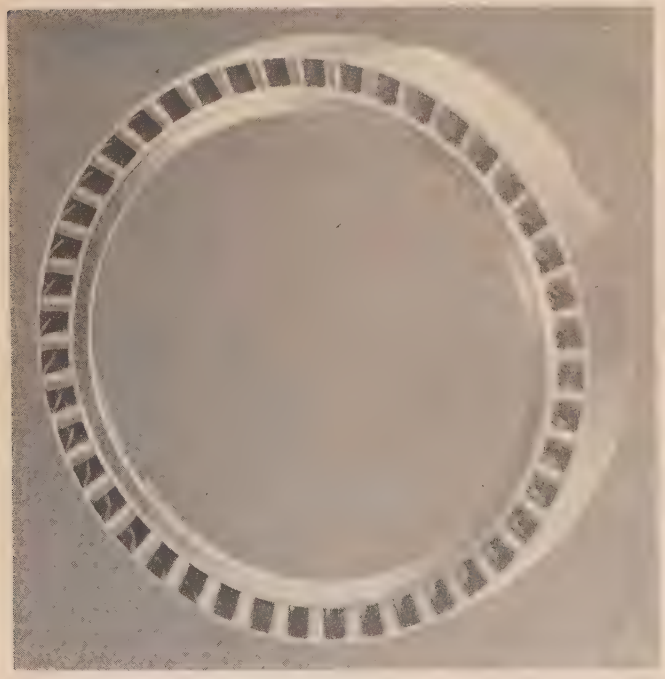


Fig. 25—First turbine

Fig. 27—One-piece core first turbine

found to machine it off. Turning the flash off was not too satisfactory, because any reasonable casting and machining tolerances resulted in either some flash remaining, or a flat surface machined on the nose of the blade, which has also proved to be detrimental to converter performance.

This problem is solved by the development of a one-piece core (Fig. 27). Since there is no parting line between the core sections forming the oil passages, no flash on the rounded entrance end of the blade is encountered (Fig. 28). In this manner, a part is now being produced which gives better converter performance and which is more economical to cast.

The fixed-blade stator is mounted on narrow full-floating bearings between which is the sprag-type stator and overrunning clutch (Fig. 29). This member is a semipermanent-mold aluminum casting. The addition of this element to the 1956 Dynaflow accounts for the improvement in performance (Fig. 30). The stall ratio has been increased from 2.1/1 to 3.2/1 for low stator-blade angle and from 2.4/1 to 3.5/1 for high stator-blade angle.

A die-cast stator member has been produced and is now undergoing durability tests (Fig. 31). The trailing ends of these blades are being die-cast to a thickness of 0.020 in., which many experts said "couldn't be done." The samples submitted for test, and the test results to date, make this part look most promising as a die-cast prospect.

A semipermanent-mold, cast-aluminum second

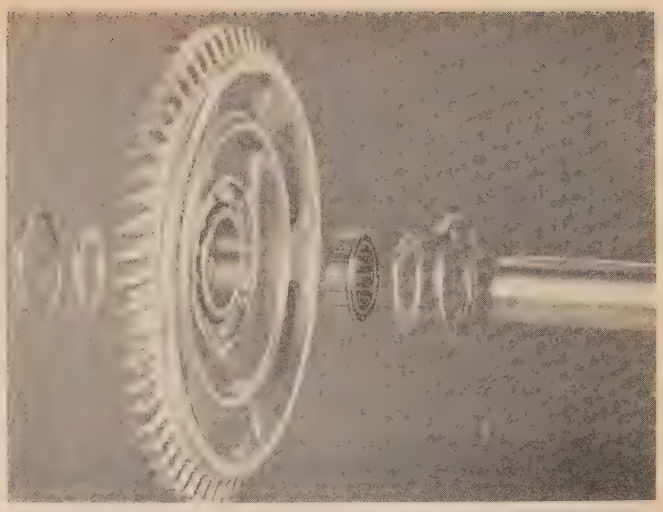


Fig. 29—Stator and overrunning clutch, a semipermanent-mold aluminum casting

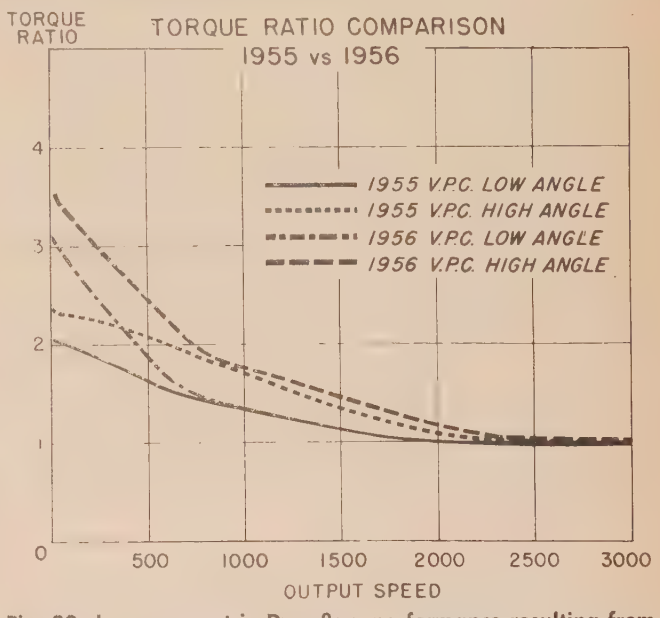


Fig. 30—Improvement in Dynaflow performance resulting from addition of sprag-type stator and overrunning clutch

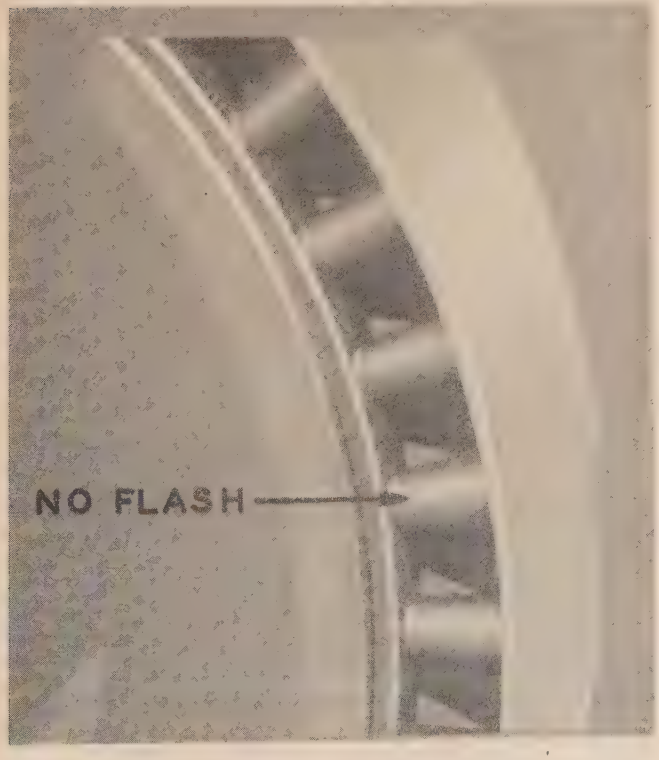


Fig. 28—Section of one-piece core first turbine, showing elimination of flash

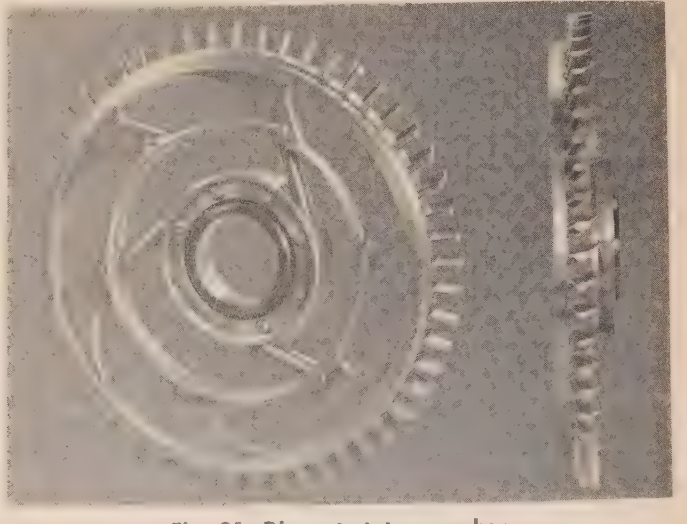


Fig. 31—Die-cast stator member

turbine was used in the 1953-1955 twin-turbine Dynaflows (Fig. 32).

This part is being supplanted by a sheet-metal assembly (Fig. 33). No performance or weight penalty was encountered in making this change. The thinner sheet-steel sections offset the greater specific weight of the material used.

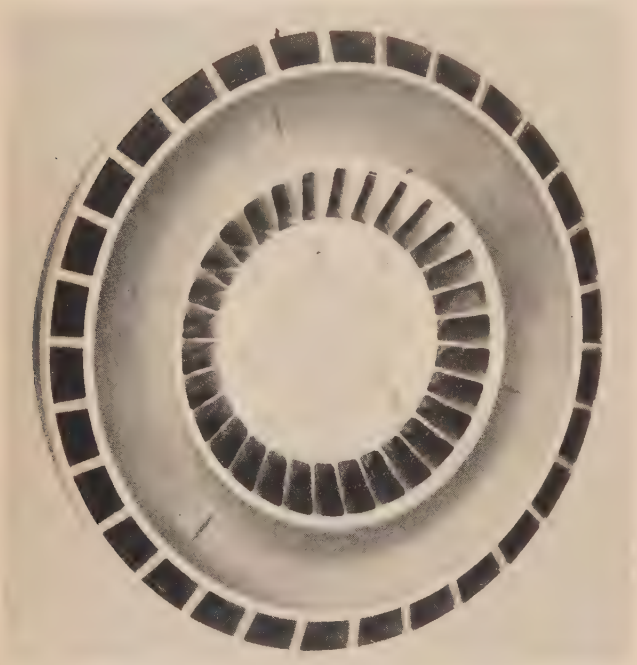


Fig. 32—Cast-aluminum second turbine used in 1953-1955 twin-turbine Dynaflows

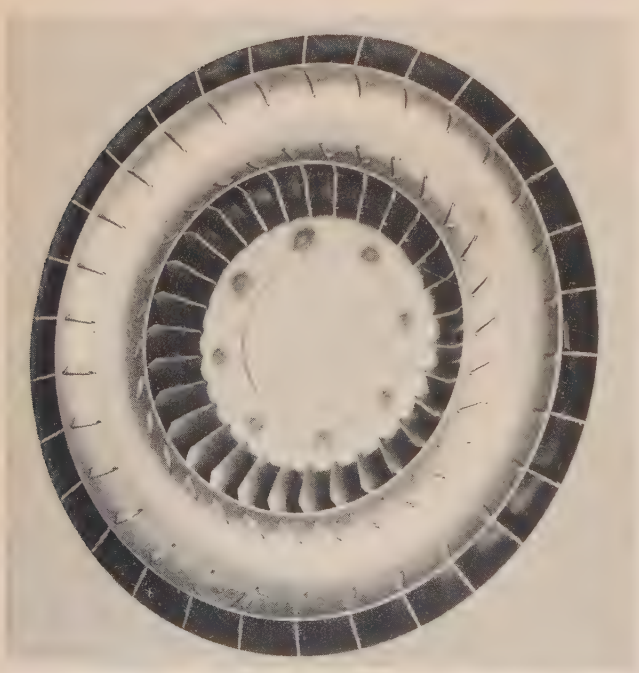


Fig. 33—Sheet metal assembly which replaced aluminum second turbine

The 1955 converter housing was a sand-cast-iron part weighing 24.2 lb (Fig. 34). The heavy ribbing used was not for structural purposes, but for acoustic reasons. Before this number of ribs was added, the resonant frequency of the engine and transmission assembly resulted in some objectional noise periods.

A full permanent-mold cast-aluminum converter housing which weighs 7.3 lb is now being used (Fig. 35). Instead of ribs, this part has convolutions in the conical wall to provide the most rigidity and strength with a minimum amount of material.

The plain bronze thrust washer between the sun gears of the transmission planetary gear unit has been replaced by a needle thrust bearing (Fig. 36). Testing was started on the bearing some time ago. Due to the increase in thrust loads caused by the higher converter torque ratio, a greater capacity bearing was needed. This was accomplished by the needle thrust bearing, in about the same space formerly needed for the plain bronze thrust washer (Fig. 37).

The additions of the twin-turbine in 1953, the variable-pitch stator in 1955, and the fixed-blade stator in 1956 have each contributed to the reduction in engine rpm, to the greatly increased low-end performance, and to the improved efficiency of today's Dynaflow (Fig. 38).

The smooth uninterrupted power flow to the rear

Fig. 34—Sand-cast-iron converter housing used in 1955 (casting weight, 28.9 lb; finished weight, 24.2 lb)

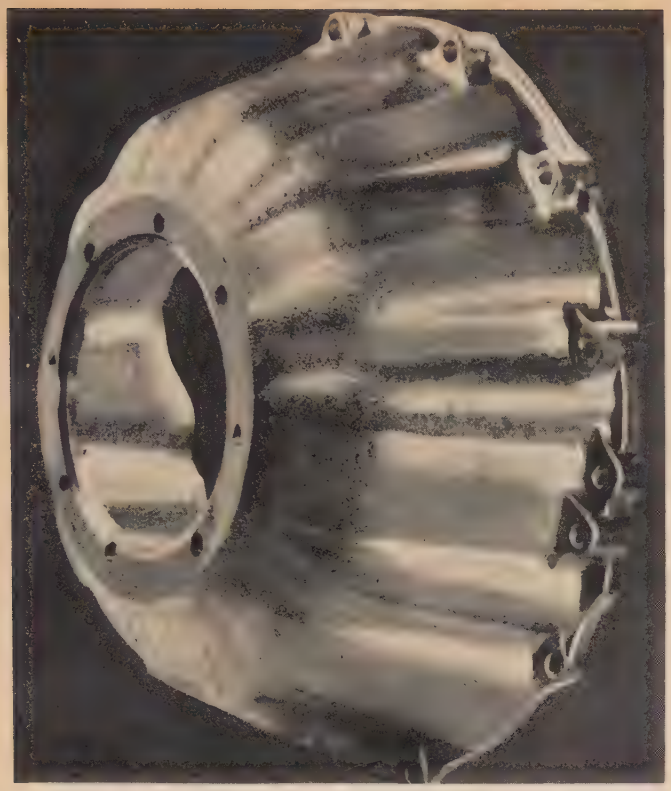


Fig. 35—Full permanent-mold cast-aluminum converter housing, weighting 7.3 lb, used now (casting weight, 8.1 lb; finished weight, 7.3 lb)

wheels that has been characteristic of all Dynaflow transmissions built has been retained. Today, it is the only automatic transmission that has this smoothness of operation, free of automatic shifts, from start to cruising speeds.

Discussion of this paper—as well as of the Gage-Rhoads paper, which follows—is to be found on pp. 132–135.

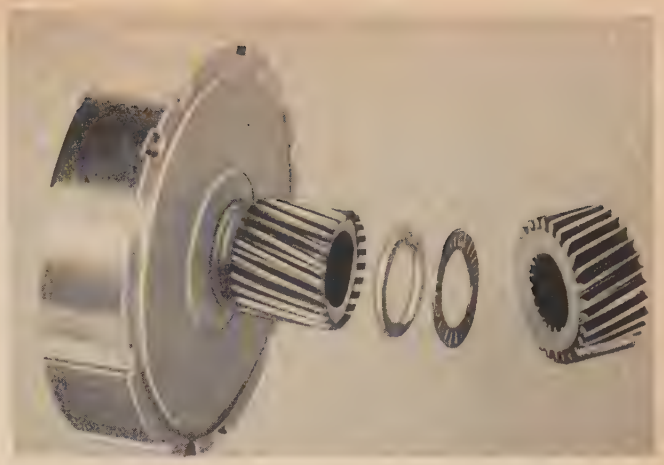


Fig. 36—Sun gears and needle thrust bearing which replaced plain bronze thrust washer between sun gears of transmission planetary gear unit

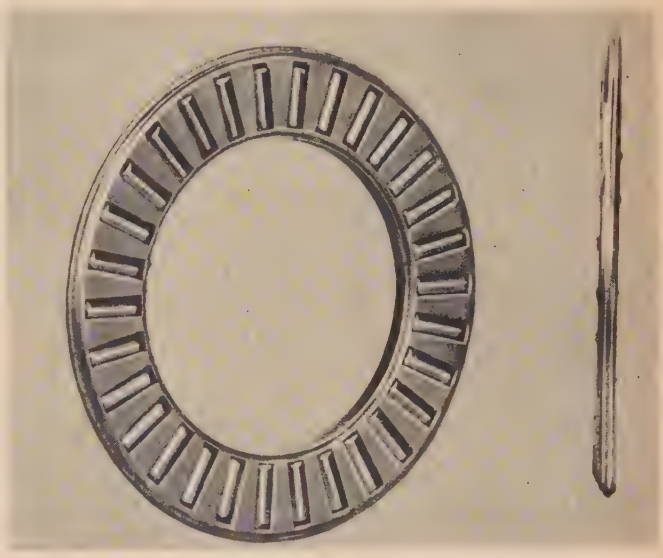


Fig. 37—Needle thrust bearing

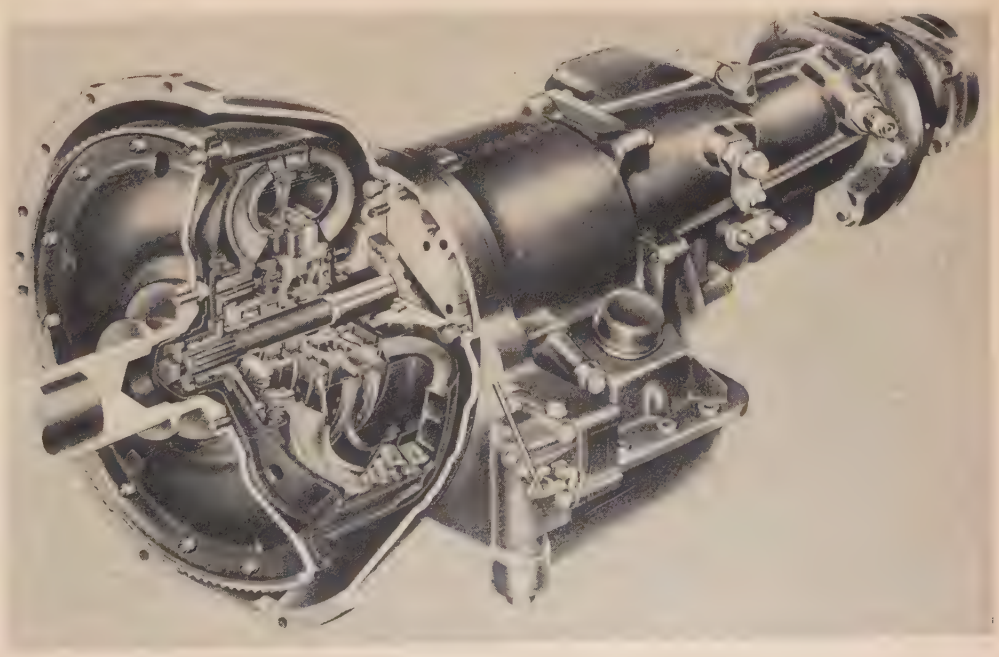


Fig. 38—Cutaway view of 1956 Dynaflow

The New General Motors Hydra-matic Transmission

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Detroit Transmission Division, GMC

This paper was presented at the SAE Annual Meeting, Detroit, Jan. 12, 1956.

SINCE the introduction of the Hydra-matic in September, 1939 on the 1940 Oldsmobile, we have built over 7,000,000 passenger-car transmissions and many thousand Hydra-matics for tanks and military vehicles.

In 1949, the Hydra-matic transmission was introduced into the commercial field in the 2708 General Motors Coach. In 1950, it was put into production on the GMC Military 2½-ton 6 x 6 truck. In 1952, the first entry was made into the commercial truck line: in that year, 0.5% of GMC trucks used the 4-speed Hydra-matic. During the next two years, the Hydra-matic coverage was expanded to encompass from the 5000-lb gross vehicle weight (GVW) to the 70,000-lb gross combination weight (GCW) trucks. Last year, 1955, in the 4-speed class, 30% of our trucks used Hydra-matics; in the 8-speed class, 15% of our trucks used the 8-speed Hydra-matic; and in the 50,000 to 70,000-gcw class, 20% used twin Hydra-matics.

In 1940, we produced 62,889 Hydra-matic transmissions, and in 1955 through October we produced 1,211,564.

With the introduction of the Hydra-matic in the 1948 Pontiac, it became apparent to everyone that the automatic transmission would find ready acceptance even in the lower-priced cars.

Today, 16 years after the introduction of the Hydra-matic, we see the automatic transmission as standard equipment on many passenger-car models. There are many more people driving cars today because of this marvelous contribution to the safety and ease of driving.

In 1938, all passenger cars built in the United States had a manually shifted transmission and foot-operated engine-to-transmission clutch. Today, all passenger cars built in the United States offer an automatic transmission.

SINCE its introduction in 1939, during which time over 7,000,000 units have been produced, Hydra-matic has regularly added such improvements as the cone friction reverse to enable rocking, and the dual-range transmission to improve performance and economy.

The 1956 model, the authors report, is completely redesigned. The basic engineering concept involved, they explain, is substitution of a small fluid coupling for the multiple-plate clutch in conjunction with the front unit gear set.

The new transmission is reputed to retain all of the desirable features of the former Hydra-matic, at the same time offering new driving ease and performance.

* Now with Buick Motor Car Division.

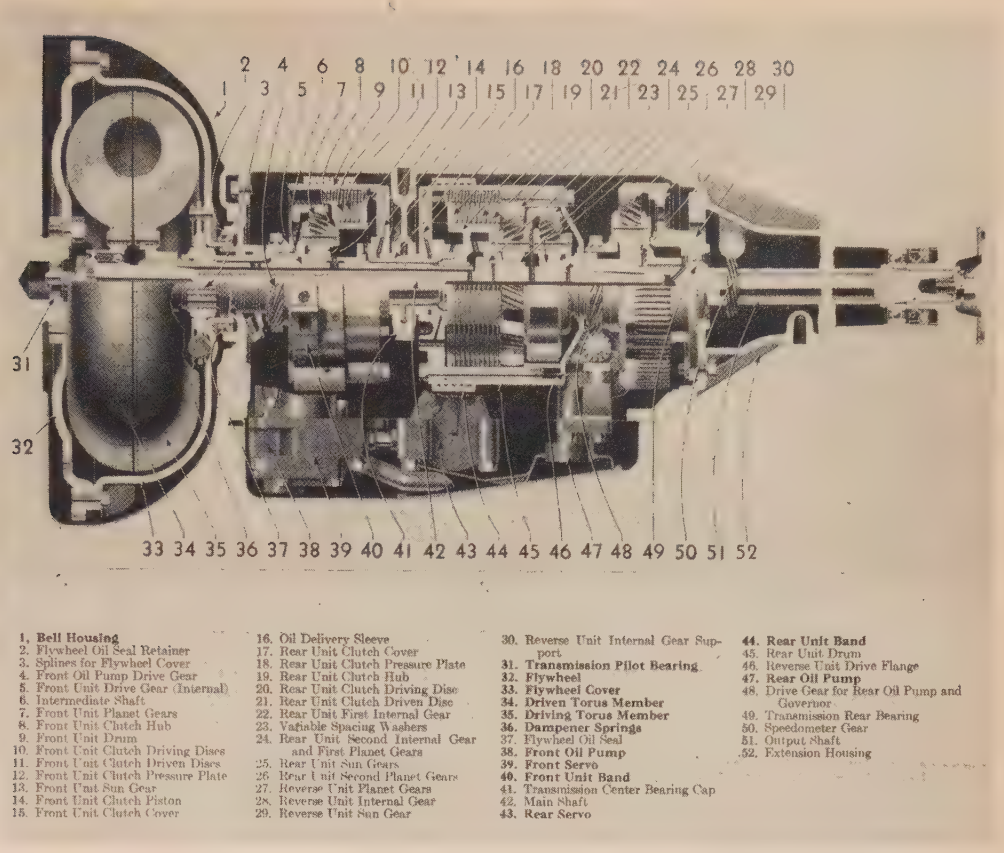


Fig. 1—Cross-section of first production Hydra-matic

In 1955, 68.5% of the passenger cars built in this country were equipped with automatic transmissions.

We feel that the foregoing remarks indicate how right our engineers were in the engineering venture begun in 1932, and how well-placed their faith was in the introduction of the Hydra-matic in 1939.

Our continued adherence to and exploitation of the Hydra-matic principle of stepped gear ratios and split torque with a fluid coupling has, in our opinion, done much to advance the art.

Over the years, it has been apparent that the Hydra-matic principle offers certain advantages. We have been able to accommodate a wide range of engine sizes with excellent overall efficiency and performance.

One of our most outstanding features has been the downshift passing gear on a fourth-to-third gear shift. With a happy combination of low axle ratio and our 1.45 or 1.55 third gear transmission ratio, we have always had flashing passing-gear performance with the engine speed in direct proportion to car speed.

In the 1940 Oldsmobile, the Hydra-matic was used with a 110-hp engine developing 210 lb-ft of torque at 2000 rpm and used with a 3.63 axle. In 1955, the hydra-matic was used with a 250-hp engine developing 345 lb-ft of torque at 2800 rpm and used with a 3.07 axle. This great change in transmission capacity requirement through the years called for continual design improvement.

Fig. 1 shows a cross-section of the first production Hydra-matic. Contrast this with a cross-section of a 1952 Dual-Range Hydra-matic shown in Fig. 2. A careful look at these two transmissions will indicate an almost complete redesign of every part as we have refined the transmission through the years. Most noteworthy of the changes in the evolution of the front pump, as shown in Fig. 3. In 1940, we had a cross-drive gear pump.

In 1946, we introduced the internal gear type pump now accepted as standard design by most of the industry. In 1952, we introduced the variable-capacity pump. The variable-capacity pump overcame the high pumping loss problem at high engine

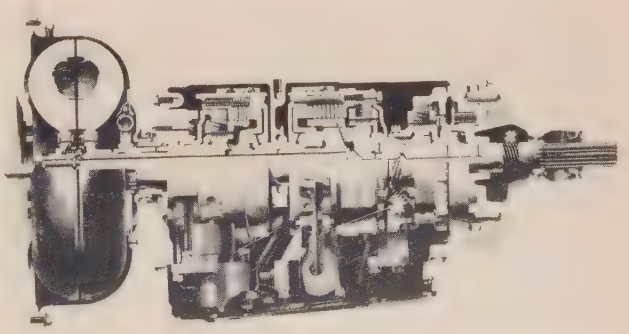


Fig. 2—Cross-section of 1952 dual-range Hydra-matic

Fig. 3—Evolution of the front pump

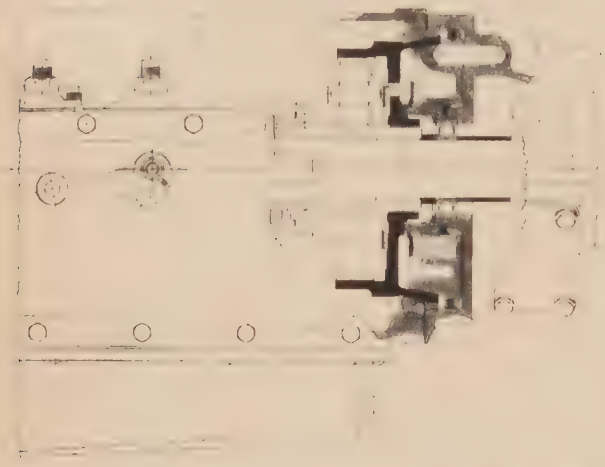


Fig. 4—Cone-type friction reverse clutch introduced in 1951

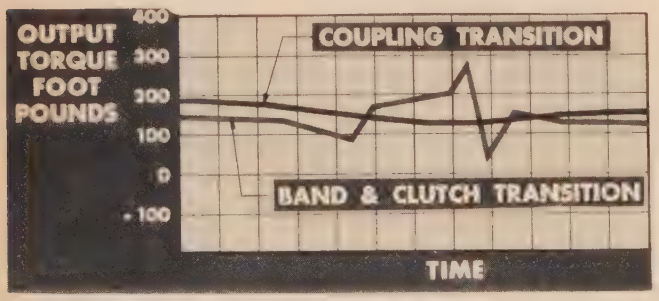


Fig. 5—Output torque characteristics of transmission during 3–4 shift

speeds which is inherent in the constant-displacement pumps prevalent in present automatic transmission design. The new pump permitted large capacity at idle speed where it is required, but permitted the ability to drop off to zero capacity when the rear pump could supply the demand for oil during higher car speeds. This permitted a saving of as much as 2 hp at top speed. The variable-capacity pump provided the means for development of our new Hydra-matic. Several other developments include the cork-and-paper type of friction plate, the large-capacity clutch piston with synthetic rubber lip seals, and the cone-type friction reverse clutch introduced in 1951 and shown in Fig. 4. The friction-type reverse permitted good rocking ability between drive and reverse.

By 1952, the trend to large engines with good torque characteristics spread throughout a wide range of engine speed, indicated that about a 3.96/1 overall reduction ratio as used in the Hydra-matic was more than adequate for general driving requirements. By introducing a performance drive range, we were able to reduce axle ratios on several of our customers' cars to 3.07. By holding in the performance range (third gear), we then had the equivalent of a 4.4 axle. This permitted excellent performance on most grades encountered. It also gave very pleasant downhill braking on general grades without the high-speed engine fuss and slow car speed involved with low gear ratios.

With the introduction of the Dual Range Hydra-matic, we set a new base line for American passenger cars for economy, performance, and flexibility.

During this phase of development, converter transmissions were coming on the market with their advantage of smoothness. While we have over the years made marked improvement in shift smoothness, it was apparent to us that the ability to make fluid-smooth shifts in the Hydra-matic was unobtainable, particularly on our 2–3 shift.

We saw some years ago that, with the trend to higher-torque engines and lower axle ratios, sooner or later we would have to increase the capacity of our transmission. In late 1945, we began to work on various torque converter designs, all of which had certain advantages. Our one difficulty was the fact that during this period we continually improved the Hydra-matic, making it necessary to start with a clean sheet of paper, many times just to catch up with the production transmission.

It is interesting to note that other companies have been faced with the same problem.

Back in 1952, we hit on a new principle for shifting gears which was evolved into our new 1956 Hydra-matic. Before we go into the design and operation of our new transmission, I would like to make a few general remarks about the new design.

In going to a new transmission there were only three requirements which our 1955 Hydra-matic did not meet. These are:

1. It did not have enough capacity for projected engines.
2. The package was not properly arranged for projected body designs.
3. It was not as smooth as some converter transmissions.

There were the following requirements that our customers and ourselves felt were essential in any future design:

1. It must have as a minimum the outstanding performance and efficiency of the Hydra-matic.
2. It must have the passing ability of the Hydra-matic.
3. It must have the low gear braking ability of the Hydra-matic.
4. It should retain the dual-range feature of the Hydra-matic.

In short, except for the three requirements not met by the 1955 Hydra-matic, it indicated to us that we should retain the Hydra-matic principle and incorporate the means for smoother shifts.

In addition to our customers' requirements, we have our own requirements. These are:

1. The transmission design must be flexible enough to accommodate a wide range of engine and car designs.

2. It must be rugged enough to withstand truck and bus operation without penalizing passenger-car weight and size.

3. Our new transmission must have a minimum of compensating adjustments to be made in the field. This objective required the elimination of band adjustments.

We feel that in our new 1956 Hydra-matic we have met every requirement that we set out to meet. It has been designed to provide the smoothest and most efficient means of transmitting power from the engine to the wheels. We feel sure the new Hydra-matic will present to the owner a new thrilling concept of driving ease.

Design Features and Operation

Our engineers have continually been searching for better and smoother ways to accomplish the transition between ratios, and in 1952 started a development which has resulted in the 1956 Hydra-matic.

The fundamental principles of the 1956 Hydra-matic remain the same. The new transmission still consists of a fluid coupling with three planetary gear sets providing four even-stepped forward speeds and reverse. The dual-range principle which found such favor on the 1952 Hydra-matic was retained, along with the fluid coupling split torque hookup which allows good economy characteristics.

While these basic fundamentals remain the same, the methods of acquiring the ratios have been completely revised. Almost every part in the transmission has been redesigned.

The basic engineering concept which made the 1956 Hydra-matic possible was to substitute a small fluid coupling in place of the multiple-plate clutch, and a one-way sprag clutch in place of the reaction band in the front unit. The action of filling a coupling to lock a planetary gear set in the direct drive by picking up a sprag clutch is basically much smoother than that which can be obtained from the action between the band and plate clutch. If a band-to-plate clutch-shift is not timed properly, it is possible to get either a neutral condition or a tieup between the clutch and band during the shift. In the 1956 Hydra-matic, the fluid coupling smoothly picks up the reaction sun gear of the front unit from the one-way sprag clutch without any possibility of either neutral or tieup. Results of this type of shift can be more easily seen from a comparison of the output torque characteristic of the transmission during the shift time (Fig. 5). You will note from the two curves the marked difference in torque change during a front unit upshift transition.

Due to the method of tying the gear set together for direct drive, the coupling is required to hold only 0.356 engine torque as compared to 1.556/1 engine torque for the large coupling. This made the project feasible, as the front unit coupling could be made small enough to be practical.

Fig. 6 is an illustration of the cross-section of the 1956 Hydra-matic. A review of this illustration will serve to point out the main design features of the new transmission.

Since a completely new transmission was required to incorporate all of the new features, it was decided to review the 1955 Hydra-matic with the idea of incorporating such design features as would help our customers as well as our own manufacture. Previous

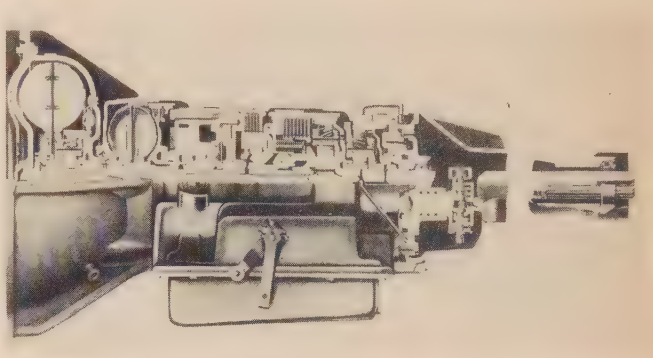


Fig. 6—Cross-section of 1956 Hydra-matic

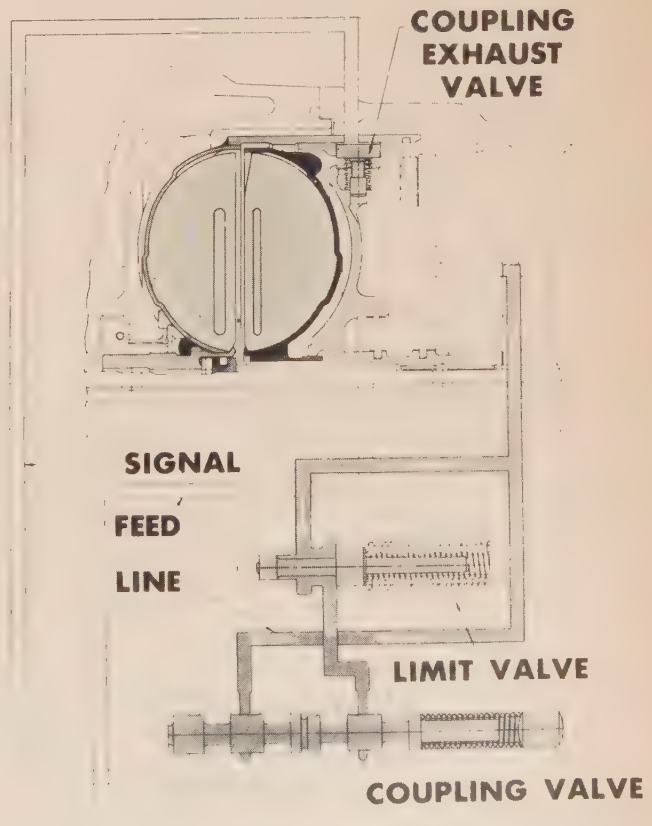


Fig. 7—Controlling valves and small coupling

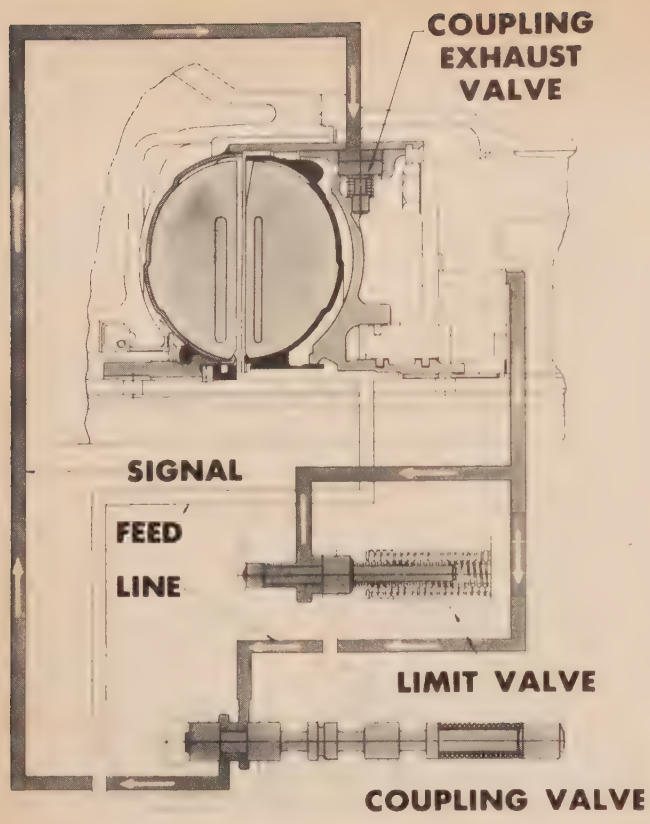


Fig. 8—Coupling valve moved to fill position, following which oil is directed to close coupling exhaust valves and line pressure is directed through the limit valve through the coupling valve to fill small coupling

to the 1956 Hydra-matic, all large fluid coupling parts have been shipped separately. It was decided to design the new transmission as a package unit which would use a flex plate drive to facilitate installation and removal from the car. In order to provide floorboard clearance and facilitate service, the valve body and side cover were removed and the valve body was installed in the bottom pan. Because of our many customers, it was decided to split the bell housing and case to provide a common case and separate bell housings to fit the customers' individual needs.

As stated before, the basic 1956 transmission is the same as previous Hydra-matics. Looking at the cross-section illustration, you will see that the drive from the engine goes into the internal gear of the front gear set which in this transmission has been installed in the large fluid coupling cavity.

The large fluid coupling receives its drive from the carrier of the front unit gear set. Reaction of the front unit is through the sun gear which is locked through the front unit sprag assembly to ground.

As the front unit coupling is filled, the fluid drive picks up the sun gear through the driven member of this coupling to take the torque reaction of the front unit as a direct ratio. The action of the coupling fill is illustrated by Figs. 7, 8, and 9.

Fig. 7 shows the environment of the controlling valves and the small coupling. Signal for fill or dump of this coupling is received from the coupling valve. This figure shows the coupling valve in the exhausted position. When the coupling valve moves to the fill position (Fig. 8), oil is first directed to close the coupling exhaust valves. At the same time, line pressure is directed through the limit valve through the coupling valve to fill the small coupling, Fig. 9. The limit valve insures that the coupling exhaust valves are closed before the coupling itself is filled. When the coupling is dumped, the coupling valve moves to the exhaust position, which closes off the flow to the coupling and exhausts the oil from the top of the coupling exhaust valve. This allows the coupling exhaust valves to move out, dumping the coupling directly.

Referring back to Fig. 6, the next item for discussion is the front pump. The same type variable capacity pump, with a greater capacity, is used in the 1956 transmission as has been used since 1952. This pump is of the variable-capacity vane type. Due to the large demand, during time of filling the coupling, the variable-vane type pump is the only practical pump to use in this application. A constant displacement pump, if used in this application, would pump too much oil when not needed to fill the coupling. This would create larger pumping losses than necessary with the problem of handling the excess oil.

Attached to the inner sprag race of the front unit

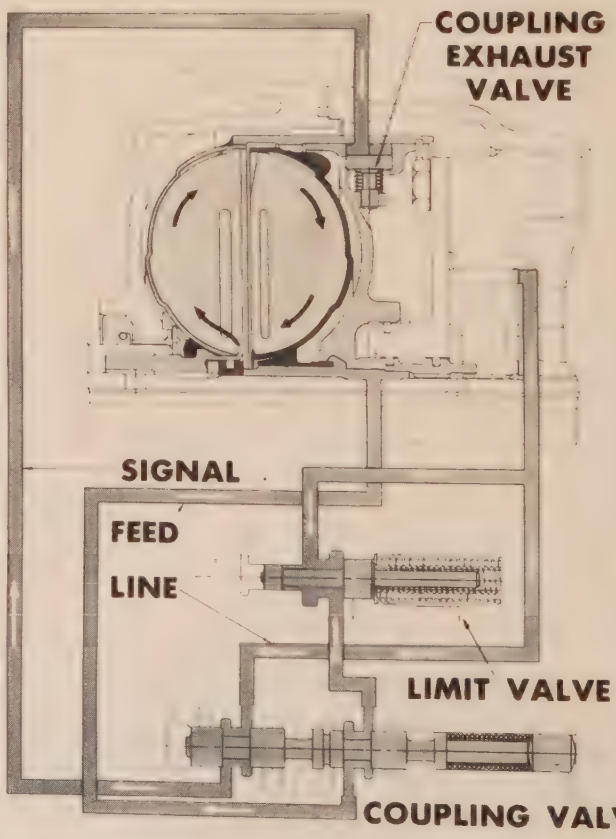


Fig. 9—Small coupling. Before filling, the limit valve insures that coupling exhaust valves are closed

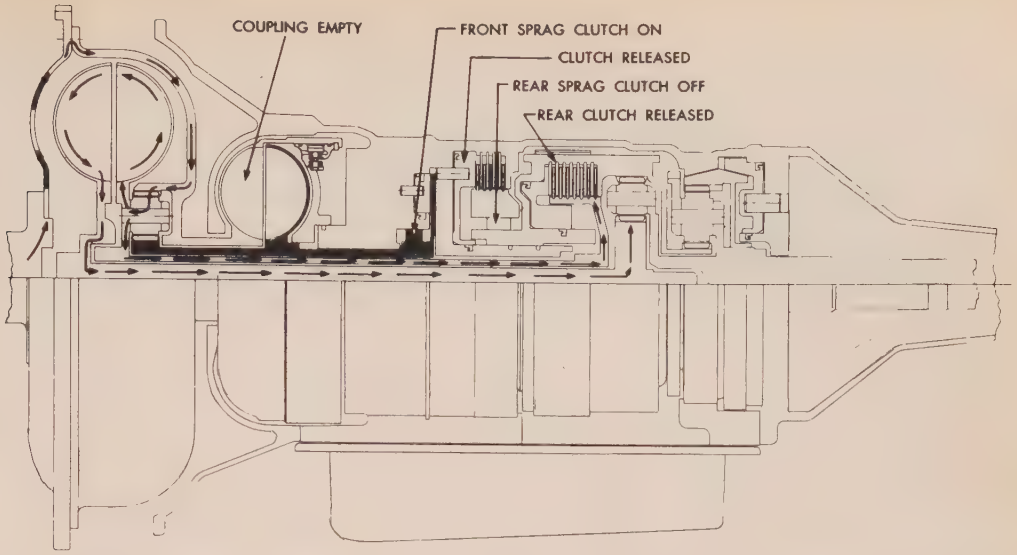


Fig. 10—Schematic of transmission in neutral—engine running

is a single plate clutch which is used only in Drive 3, low, and reverse to take overrun reaction of the front unit sprag assembly. This plate is not in the picture in any ratio in Drive 4 Range.

In the rear unit assembly, reaction for the reduction ratio is taken by the internal gear through another one-way sprag clutch to ground. Neutral is obtained in the transmission by making and breaking the reaction of the rear unit sprag assembly by means of a multiple plate clutch. A band is used in conjunction with the rear unit only for overrun braking in low gear.

It can be seen from this description that three out of four friction elements, as present in the 1955 Hydra-matic, have been replaced with one-way clutches and a fluid coupling in the 1956 Hydra-matic. Since the front unit coupling is either exhausted or filled on every shift, it allows inherently smoother timing of the shifts than was possible before. The third gear set as seen in Fig. 6 is used for reverse only, and reaction for the internal gear is taken through a cone clutch.

In the 1955 Hydra-matic, a cross-drive rear pump and governor assembly is used. In the 1956 transmission, the drive gear of the rear pump is coaxial with the output shaft and drives the driven gear of the pump, which in turn drives the governor. Accessibility to the governor is made easy by removing or backing off the rear bearing retainer.

Fig. 10 is the schematic diagram of the transmission in neutral. Power to the rear wheels is broken in the transmission because the neutral clutch is released allowing no drive through the rear unit.

Fig. 11 illustrates first gear. In the 1955 Hydra-matic, both bands had to be applied to get first-gear reaction. In the 1956 Hydra-matic, the neutral clutch

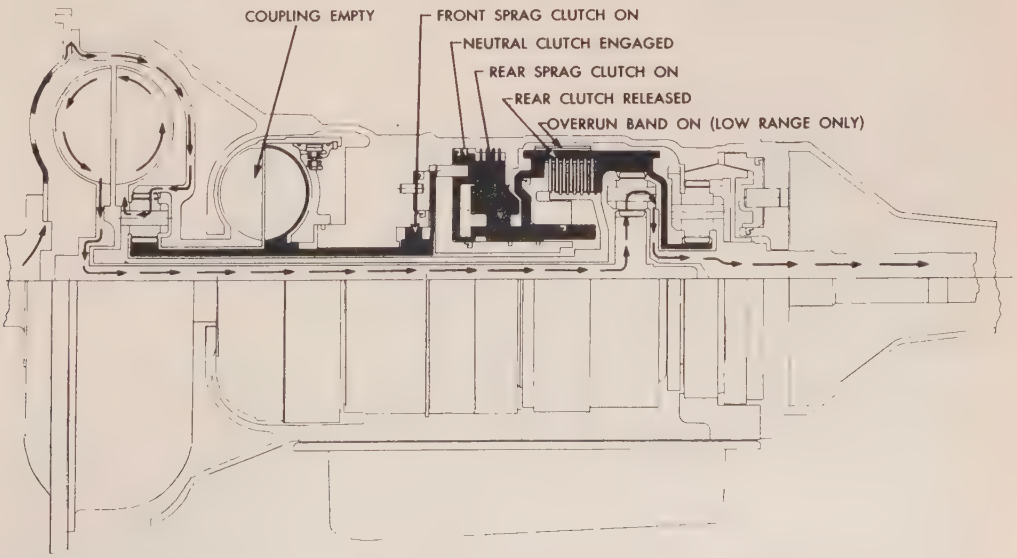


Fig. 11—First gear

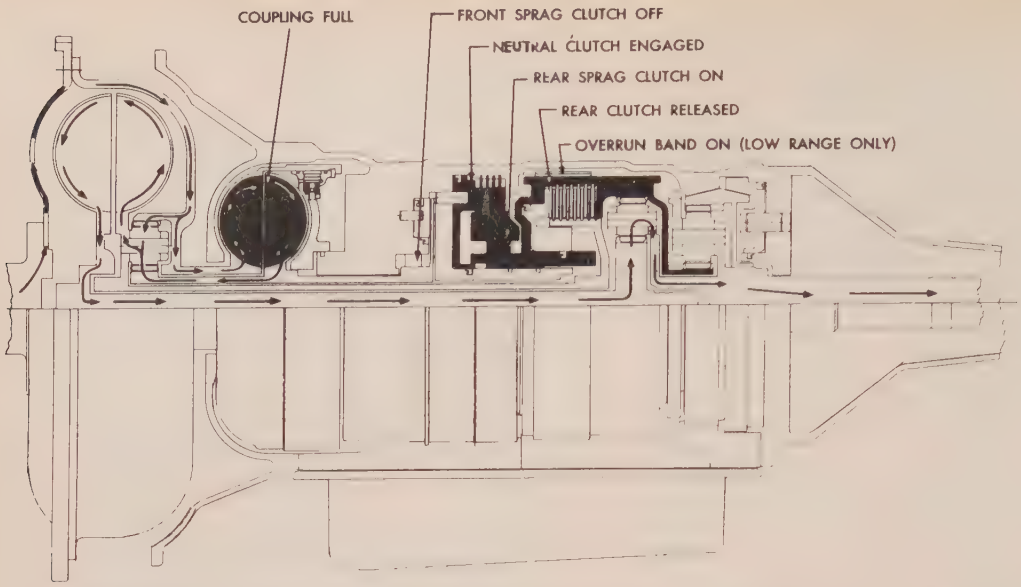


Fig. 12—Second gear

only has to be applied. The front unit is in reduction with the sprag grounded and coupling empty. The rear unit is in reduction with the rear sprag on and rear clutch off.

The power flows mechanically from the flywheel to the torus cover and to the front unit internal gear which is attached to the torus cover. Power is directed through the internal gear to the front unit carrier assembly. The sun gear is held by the front sprag assembly, and the carrier rotates around the sun gear at a slower rate or 1.55/1 reduction. The front unit carrier is attached to the large drive torus member. In turn, the driven torus member drives the connected main shaft and rear unit sun gear. Since the internal gear is grounded by the rear unit sprag assembly, the rear unit sun gear drives the rear unit carrier at a reduced speed, or 2.55 reduction. The carrier of the rear unit is attached to the output shaft, and power will be directed from the carrier through the output shaft. The transmission is now in first speed. The rear unit reduction 2.5532/1 times the front unit reduction 1.5536/1 = 3.966/1 reduction.

Fig. 12 illustrates second gear. The coupling is filled hydraulically picking up the reaction of the front unit sun gear. Now the front unit is in direct drive with the sprag off and coupling full. The rear unit is still in reduction with the sprag on and the rear clutch off. Power flows mechanically from the flywheel to the torus cover and to the front unit internal gear which is attached to the torus cover. It also flows hydraulically through the front unit torus assembly to the sun gear. Thus, through the split torque principle, the planetary gear set is locked together (internal gear and sun gear), through the front unit torus assembly, which creates a 1/1 ratio in the front unit. The ratio of split torque drive is 0.356 hydraulic and 0.644 mechanical.

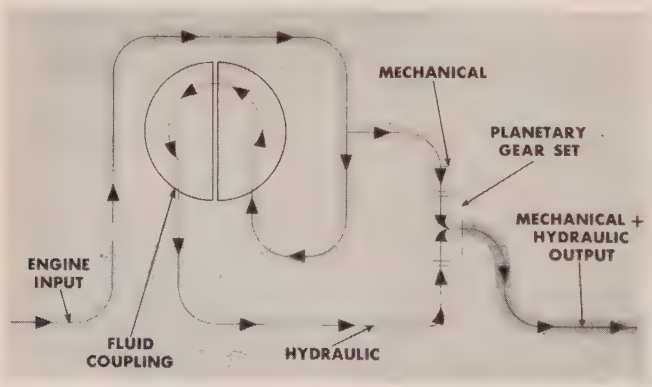


Fig. 13—Split torque principle used in front unit assembly and in conjunction with large torus assembly and rear unit gear set in third and fourth speeds

The front unit planet carrier, attached to the large drive torus member, transmits torque through the oil to the driven torus member. In turn, the large driven torus member drives a connected main shaft and rear unit sun gear. The rear unit is in reduction, as was described in first gear. The transmission is now in second speed, with reduction ratio of 2.5532/1.

A schematic diagram (Fig. 13) illustrates the split-torque principle as used in the front unit assembly and in conjunction with the large torus assembly and rear unit gear set in third and fourth speeds. As you can see from the diagram, the input splits, part mechanically through the ring gear, and part hydraulically through the coupling to the sun gear. Both combine to give a 1/1 ratio of input to output torque while allowing a cushioning effect through the torus assembly.

This is the same split-torque principle as used in the Hydra-matic. It allows a maximum of economy with a minimum of slips; at the same time it gives a cushioning effect to all shifts.

Fig. 14 illustrates third gear. To obtain third gear, we use the front unit in reduction and rear unit in direct drive. This is accomplished by applying the rear clutch and exhausting the front unit coupling,

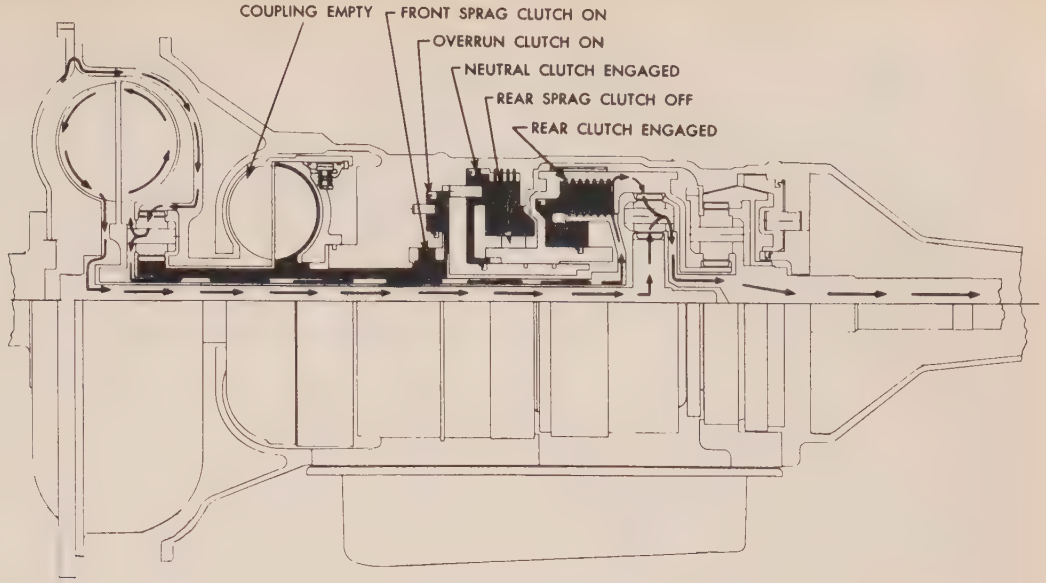


Fig. 14—Third gear

which means the front unit sprag is grounded and the coupling is empty; the rear unit sprag is off and the rear clutch is on. Power flows mechanically from the flywheel to the torus cover and to the front unit internal gear which is attached to the torus cover. Power is directed from the internal gear to the front unit carrier assembly as the front unit sun gear is grounded by the front sprag assembly.

When the rear clutch is on, power divides to the front unit carrier assembly and is directed to the rear unit in two ways:

1. Fluid—The front unit carrier is attached to the drive torus member and will, through oil, drive the driven torus member. The driven torus member drives the connected main shaft and rear unit sun gear.

2. Mechanical—Power is directed mechanically from the carrier of the front unit to the rear unit through the intermediate shaft which is splined to the large drive torus member. The intermediate shaft is splined to the rear clutch hub.

The rear clutch is applied and power is transmitted through the clutch plates, drum, and rear internal gear to the planet pinions where it combines with the power from the rear unit sun gear. The split torque ratio under these conditions is 60% mechanical and 40% hydraulic.

The carrier of the rear unit is attached to the output shaft, and directs power to the rear wheels. The transmission is now in third speed, with reduction in the front unit 1.5536/1.

Fig. 15 illustrates fourth gear. During the shift from third to fourth gear, it is only necessary to fill the front unit coupling. The rear unit is already in direct drive, and filling the front unit coupling puts it into direct drive, giving a 1/1 transmission ratio.

Fig. 16 illustrates reverse power flow. In reverse,

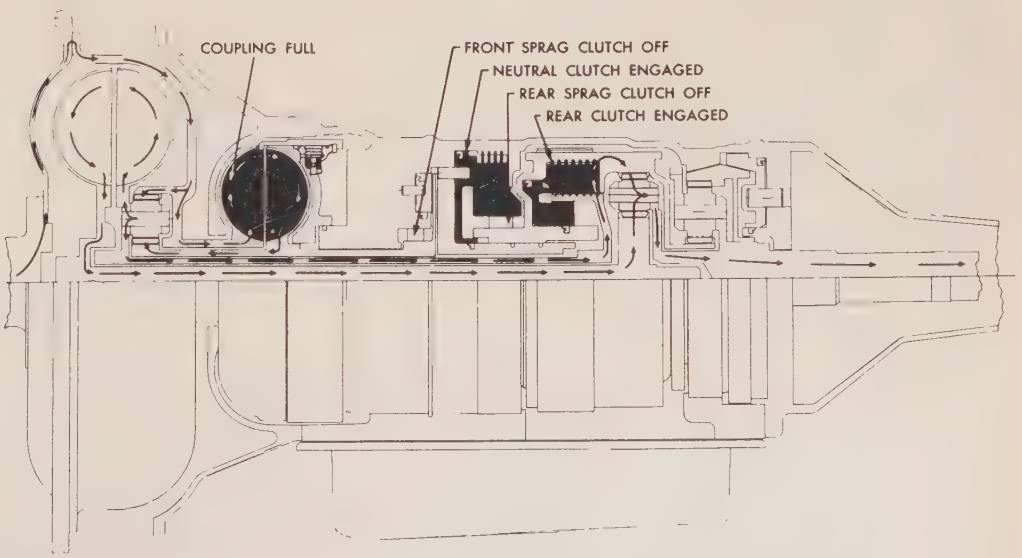


Fig. 15—Fourth gear

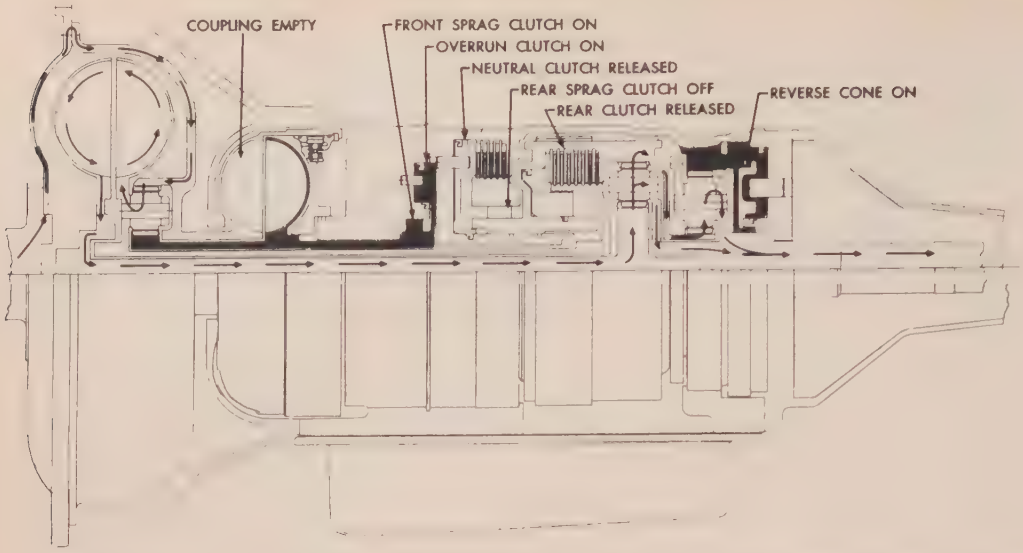


Fig. 16—Reverse power flow

the front unit is in reduction with the sprag on and coupling empty. The reverse gear set is in reduction with the internal gear grounded by means of a cone clutch, and the rear unit gear set is compounded with the reverse gear set.

Drive is through the reduction of the front unit through the large torus assembly to the rear unit sun gear. Since the rear sprag is released, the sun gear of the rear unit will drive the rear internal gear counterclockwise or reverse. The internal gear through a flange, will drive the center gear of the reverse unit counterclockwise. Power then travels through the reverse planetary carrier to the output shaft in reduction, because the internal gear is grounded by the reverse cone clutch.

The transmission is now in reverse with a 4.307 reduction.

Now that the functions of the 1956 Hydra-matic have been described, some of the driving features will be discussed.

In addition to the 1955 shift sequence of Neutral, Drive 4, Drive 3 or "S," "Lo," and "Reverse," a new position of "Park" has been added to the left of the neutral position. This allows the car to be started in either "Park" or "Neutral," and is a positive "Park" with the engine running.

In addition to having excellent control of all forward driving conditions through throttle control of the shift points in "Drive 4" range, additional dual range controls have been retained. The "Drive 3" or "Super" range with its top third gear and "Lo" with a top second gear give excellent over all control of the vehicle. The third gear range can be used not only for mountain highway running, but also as an additional braking feature when the vehicle needs to be slowed down from high speed. For example, when the driver approaches a car going in the same direction and has to slow down prior to passing, "Drive 3" range is excellent as a braking range as well as for passing when the road is clear.

Both "Drive 3" and "Lo" ranges as well as "Reverse" are inhibited. "Drive 3" is inhibited above 65 to 70 mph, and "Lo" above 40 mph.

The inherent smoothness of the front unit coupling control has led to more performance under city driving conditions. Because of this smoothness, it was possible to raise the part-throttle 4–3 downshift to 30–40 mph, thus giving better performance under part-throttle conditions. Full-throttle 4–3 detent downshifts to 65–70 mph with 3–4 upshifts at 70–80 mph retain the excellent high-speed passing ability of the Hydra-matic with the additional feature of imperceptible shift smoothness.

This is our 1956 Hydra-matic, which combines the recognized features of economy and performance with a new fluid smoothness of operation not before possible.

DISCUSSION

of Gorsky and Gage-Rhoads papers

Lower Stall Speeds Preferred Despite Performance

—Forest McFarland
Studebaker-Packard Corp.

THE introduction of the 1956 models of the new Hydra-matic and new Dynaflow transmissions marks another step in each of these paths of automatic transmission development.

In the new Hydra-matic, the use of the small coupling is a move toward greater use of fluid in moving from one ratio to another. Filling and dumping of fluid couplings to connect and disconnect shafts has been known of for a long time for a variety of uses, but we must give Detroit Transmission credit for doing this fast enough so that it can be used as a speed-changing unit in automobiles. I believe a little more information on the valving problem to accomplish this change, and more discussion on the small fluid coupling, would be of interest. Fig. 13 could probably be better understood if it were expanded schematically to include

the balance of the transmission with the units located as in the actual design.

The 1956 unit appears more complicated in general structure than its predecessor. Despite this additional complication, I am sure Detroit Transmission are expecting this unit to have a better service life than the former, due to reduction of use of friction elements during shifting and reduction of sensitiveness in the shifting. Comments by the authors on the two-to-three shift should be interesting, as this appears to be the only shift where timing can still be somewhat critical. My own driving experience leads me to say this transmission is greatly improved in reduction of harshness over the last model.

I don't know if predictions are in order, but if so, I would expect to see in the future elimination of more frictional members in this unit. I am saying this as it appears to be simply a part of the trend.

The new Buick Dynaflow with the added first reactor between the first and second turbine has definitely improved the performance of this unit in the range from stall to approximately 20 mph. Driving experience confirms this fact. When one considers the negative torque fed into the second turbine which our own tests on the 1955 unit show to be present up to approximately 700 rpm converter output shaft, it is understandable why the reactor was inserted between the first and second turbine. We have shown a certain amount of efficiency loss when inserting struts between impeller members. Any figures the author wishes to quote on this condition would be interesting. This condition may not be as severe as the one we tested.

The figures for performance of the 1955 unit show fairly good agreement with our test results. We have not completed tests on the 1956 unit, so cannot comment on these figures. After scanning the advertised engine horsepowers and then comparing them with the actual engine horsepowers delivered at the flywheel, I can truthfully say the torque converter people are really quite honest. Apparently they are not subjected to as much pressure from the sales departments, or else are more rugged individuals.

We can subscribe to the use of the needle bearing between the sun gears. We have used a needle bearing in this location through our last model, and have found it has capacity comparable to the ball bearing we originally used. To insure against retainer breakage, we requested the bearing people to join the two stamped discs at their outside diameter. Buick appears to have the same type of construction.

The author states that the first turbine is 1.6 faster than the second turbine. I am sure he means 1.6 times as fast. However, this is quarreling about details.

Casting the first turbine in one piece, eliminating the fins at entrance, is a worthwhile achievement. We have found machining or otherwise removing these projections satisfactorily is difficult. Our men thought they could see core prints in Fig. 27, where individual cores were assembled. I assume these marks are of slides that withdraw radially. Could the author clear up this point?

In Fig. 30, I see no engine speeds. How do the stall figures compare with the 1955 figures shown in Fig. 6?

We prefer stall speeds below the 2250 rpm shown in Fig. 6 in high range for 170 ft-lb engine torque. We have, however, resorted to a stall speed of 2100 rpm at actual engine torque in the converters of our Caribbean and other Packard and Clipper sport models, where we went all out to get a stall torque ratio of 3.25 in the converter without benefit of gearing. The temptation to resort to higher stall speeds to get additional performance is great.

The subject of stall speeds of converters reminds me of an actual occurrence in the early days of the torque converter. A golfing companion of mine, an insurance man, had just purchased a new Buick Roadmaster with a Dynaflow torque converter. I said to him, "John, how do you like it?". He replied, "I like it fine, but the transmission slipped too much when I started up from a stop. I took the car into the service station and they tightened it up. It's better now." So if any of you torque converter men have trouble reducing stall speeds, I know of a good garage that can help you.

We are interested in Buick's going to the aluminum bell housing. We went to an aluminum bell housing and tail housing a year ago, and a main transmission case this year. We were also concerned about greater sound transfer or radiation, but to date have not found anything uncommercial. We have a straight coned section bell housing which we believe gives us more stiffness than our previous curved section. Any further experience Buick has had soundwise would be interesting.

In summing up, we see Hydra-matic with very good performance taking another step to obtain more smoothness, and Buick with adequate smoothness obtaining more performance, which has been characteristic of both of these paths of automatic transmission development since their beginning.

I wish to compliment the authors of both papers and their associates for the forward steps they have taken in their respective designs.

Sees Progress Toward Performance-Smoothness Goal

—H. C. English
Ford Motor Co.

RATHER than discuss each paper individually, I feel that it might be interesting to review a little of the past history of these two fundamentally different transmissions.

The Hydra-matic transmission as introduced in 1939 represented a rather substantial improvement over its predecessor, the semiautomatic four-speed planetary transmission first offered by Oldsmobile on its 1937 models. This transmisison, which many of you may have forgotten by now, required a conventional clutch, and employed a countershaft gear set to provide neutral, forward, and reverse. All this was in addition to the four-speed planetary gear set. The Hydra-matic, with its fluid coupling, eliminated the necessity for a conventional clutch, and made possible completely automatic operation. What seemed then to be an unreasonably complex hydraulic control system made full use of the available four gear ratios.

As we look back, I suppose we might say that the Hydra-matic followed a rather natural evolution from the hand-shifted countershaft transmission, to the semiautomatic planetary transmission with its conventional clutch, to the full automatic planetary transmission with fluid coupling.

In those days the hydraulic torque converter was given very little consideration as a transmission for passenger-car use. In fact, the sum total of knowledge of torque converter design in this country in the 1930's was far from impressive. Only a few individuals knew enough about the device to have faith in its ultimate possibilities for widespread automotive application.

The Hydra-matic enjoyed a lone position in the automotive field from 1939 to 1948, before the torque converter transmission made its appearance. These first converter transmissions, with torque ratios of 2.2 to 2.4, introduced the new concept of no-shift driving. Performance was far from impressive; however, smoothness of operation was outstanding.

Since the first torque converter transmission made its appearance in passenger vehicles, a number of new automatics have been introduced. All have been of the converter type, and most have employed automatic geared ratios to supplement the converter in the interest of performance and economy.

Dynaflow is the only transmission to ahdere rigidly to its basic concept of no-shift operation. Improvements in this transmission, including the "twin-turbine," variable-pitch stator, and finally the double-stator converter in 1956, have all been directed at improvement in performance within the framework of the torque converter without resorting to automatic operation of the supplemental low gear set.

Continued success of the converter transmissions with their outstanding smoothness of operation and steadily improving performance set a new standard which was very difficult to attain in a friction-controlled, step-geared transmission such as the Hydra-matic. Introduction of the controlled fluid coupling in the 1956 Hydra-matic to eliminate the front unit friction clutch has resulted in a substantial improvement in the quality of all automatic shifts, and has moved the Hydra-matic one step closer to the converter transmission in terms of smoothness of operation.

Thus we see that these two transmissions, representing basically different concepts of operation, continue to be improved in the direction of a common goal of ultimate performance and smoothness of operation which the motoring public is being gradually conditioned to expect.

There are a few points which have occurred to me in reviewing

these papers, and which might be of interest.

With regard to the Hydra-matic transmission:

1. What is the effect on efficiency of submerging the front unit gear set?

2. What are the road-load and full-throttle slip characteristics of the controlled fluid coupling?

3. What is the normal fill time of the fluid coupling?

Regarding the Dynaflow transmission:

1. How does efficiency of the new torque converter, with its five wheels and second stator having its web section in the fluid path, compare with the 1955 Dynaflow converter?

Current Efforts Called "Period of Refinement"

—W. R. Rodger
Chrysler Corp.

OVER the interval through 1953 at these transmission "graduate school" meetings, we have listened as each proud engineer told of his company's choice of an automatic transmission with which to enter the field. Now we are well into the period of refinement. Our earliest contender for honors in this field has seen his superiority challenged by a newcomer, the torque converter, and he is putting in new life with a redesign. He has done it by component refinement, using different principles to accomplish the same purpose better.

The other fellow who isn't going to put an automatic shift in, come hell or high water, is showing amazing ingenuity in cramming some new element into that crowded fluid circuit almost every year.

As I was studying over these two papers, I was imagining what a practical English engineer and a stranger in American ways and motors cars would conclude. He almost certainly would say, "That first transmission, running on 100% fluid multiplication, must go in the luxury car; their Rolls Royce, where operational economy is no problem. The other one with four geared ratios and split torque coupling must be for the cheap family car without too much power; a car about like the Austin, where they are trying to save petrol and still get fair performance."

Now imagine his wonderment when he finds out the truth. And imagine his utter amazement when he finds that the same company makes and uses both of them. Isn't America wonderful!

Mr. Gorsky has shown us what can be accomplished to overcome the apparent fundamental deficiencies of a transmission type without sacrificing the initial advantage or goal. As usual, he has shown some unique manufacturing processes. We have run quite extensive dynamometer tests on this torque converter, and have found that the increase in torque ratio was obtained at considerable expense in loss of efficiency at the higher vehicle speeds. However, this is no doubt difficult to measure in terms of tank mileage.

The Hydra-matic transmission has certainly reached a new high in compleixty. I think the next years are going to be very interesting to watch. The question of how long this type can withstand the onslaught of the torque converter will be paramount in our bull sessions. We can already see control compromises in the form of light throttle kickdowns at fairly high car speeds in an effort to match the flexibility of the converter. I think I'll venture no higher than the two-dollar window and put my money on the converter.

Shows Schematics of Operation and Torque Paths

—Howard W. Simpson
Consulting Engineer

THE authors have shown that the practical solution of problems which at first seem difficult can become almost routine procedure, when imagination and initiative are applied.

The achievement of smoother shifting, greater accessibility, and adaptability in a package of the same length and weight as formerly, is an outstanding accomplishment.

Use of the small coupling as a clutch is feasible because it only carries 35% of engine torque. It should require little or no service, since there are no parts to wear.

Likewise, the use of the two sprag clutches should reduce service, not to mention elimination of band adjustments. The sprags pick up all reaction torques in both up and down shifting. Thus the brake band and the single disc clutch are only subjected to overrun torque, which is comparatively infrequent and low in value, and also is more gradual in application.

It is gratifying to see that although almost everything else is changed, simple planetary gear sets are retained. The writer has long recommended and used this type exclusively, because they are more efficient, more adaptable, and cheaper than other types having long and short planets, stepped planets, or sets in which each planet does not mesh with both the sun and ring gear. Besides, simple planetaries are extremely versatile in producing desired gear ratios, and load conditions are good.

An improvement in the presentation can be made, in my opinion, by use of simpler schematics to show the operation and torque paths in the various speeds. (See Fig. A.) This also gives a clear picture of the split torque conditions which occur in three of the speeds.

These splits occur in two distinct phases, in both of which sun-gear torque is transmitted hydraulically, and ring-gear torque mechanically.

The planetary sets involved act as differential torque dividers, with sun and ring-gear torque split in proportion to the number of teeth in these two gears.

Thus in second and fourth speeds, the front gear set acts as the divider, with sun-gear torque (35.6% of input) first passing through the small fluid coupling and then through the gear set to the large fluid coupling. The remainder, 64.4%, is transmitted by the ring gear directly through the gear set to the large coupling.

In third and fourth speed, the torque coming out of the carrier of the front set is divided by the middle gear set, with 60.8% (middle set ring-gear torque) being tapped off before it gets to the large coupling. The remainder, 39.2%, meanwhile passes through the large coupling before reaching the sun gear of the middle gear set.

The difference between third and fourth speed is that in third the torque coming out of the front set carrier has already been multiplied by 1.55 in that set, while in fourth this set is in 1/1 ratio, and carrier torque is equal to engine torque.

Therefore, the 39.2% transmitted by the large coupling is greater in third than in fourth, being $1.55 \times 39.2 = 60.76\%$ of engine torque, as shown in Table A.

The small coupling, being located in the drive line ahead of the front gear set is never subjected to multiplied torque, and the 35% it transmits in second and fourth is always 35% of engine torque.

It should be stated that since both front and middle gear sets split the torque before it reaches the gear set itself, the terms "divider" and "splitter" may not be entirely correct, but are used for want of more accurate terms.

To summarize, the front and middle gear sets are both made use of in all four speeds, each functioning as reduction gear in two speeds, and a splitter device in the other two.

In Fig. A, the parts that are working in each speed are shaded, and the others either unshaded or shown as dotted lines.

Gorsky Closure To Discussion

IN answer to Mr. McFarland's question regarding core prints on the one-piece first turbine, as shown in Fig. 27, these marks were caused by the slides that were withdrawn radially.

Regarding the question of stall-speed comparison of the 1956 Dynaflow with the 1955 Dynaflow, as shown in Fig. 6, the addition of the new stator has increased the stall speed about 200 rpm with the variable pitch stator at high angle, and a lower stall speed is obtained when the variable-pitch stator is at low angle.

The spokes in the fixed-blade stator produce some obstruction to fluid flow. An improvement in efficiency is obtained at the low-speed end of the curve, with a slightly lower efficiency in the cruising range. The economy and efficiency in the cruising range

Table A—Per Cent Engine Torque Transmitted by Fluid Couplings

	1	2	3	4	Reverse
Large Coupling	100	100	60.76	39.2	100
Small Coupling	0	35.6	0	35.6	0

are about the same as in the 1955 Dynaflow, with improved fuel economy for city driving because of the better efficiency at low vehicle speeds.

Regarding the aluminum bell housing, the heavy ribbing convolutions were designed around the bolt-hole mountings. Tests prove that the aluminum housings compare favorably with cast iron for acoustical qualities.

Gage-Rhoads Closure To Discussion

MR. ENGLISH asked about the effect on efficiency of submerging the front gear set. In second and fourth speeds, the elements are rotating at practically the same speed, with no measurable loss in efficiency. Loss in first and third is very slight, measuring ½ to ¾ hp at 4000 engine rpm.

Regarding the slip characteristics of the controlled fluid coupling, the slip amounts to about 200 to 300 rpm. The fourth-to-third speed downshift removes the slip objection. On controlled fuel economy tests in the vehicle, no differences were noted for the two types of Hydra-matic transmission.

ORAL DISCUSSION

Reported by Myron M. Schall

Dana Corp.

O. K. Kelley, General Motors Corp.: A transmission should be efficient, economical, and smooth. Compromises in engine noise were originally made in the Dynaflow, whereas a minimum amount of engine noise was obtained in the Hydra-matic with improved performance.

Dynaflow is adhering to its original task of obtaining smoothness, and Hydra-matic is coming into its own with refinements to obtain smoothness.

Drivers with automatic transmissions often say, "I never mind stopping because it is so easy to start again."

Julian Soukey, Caterpillar Tractor Co.: What about the large slip at low vehicle speeds? Will you clarify the amount of slip at about 35 mph cruising speed?

Mr. Gorsky: The performance curves, shown in Fig. 6, are plotted for 150 lb-ft for comparison purposes. This torque is greater than road-load requirements, and the slip is greater at full throttle than for road-load conditions. Some penalty in slip is obtained for smoothness.

Oscar Banker, New Products Corp.: Why does Hydra-matic not change its shift quadrant for the sake of standardization?

Mr. Rhoads: The shift pattern is maintained to enable the vehicle to be rocked between low and reverse.

R. P. Lewis, Dana Corp.: The SAE Hydrodynamic Drive and Transmission Committee has considered this matter for standardization, and has recently decided to drop it since, with the introduction of push-button shifting devices, the shift selector may eventually be replaced.

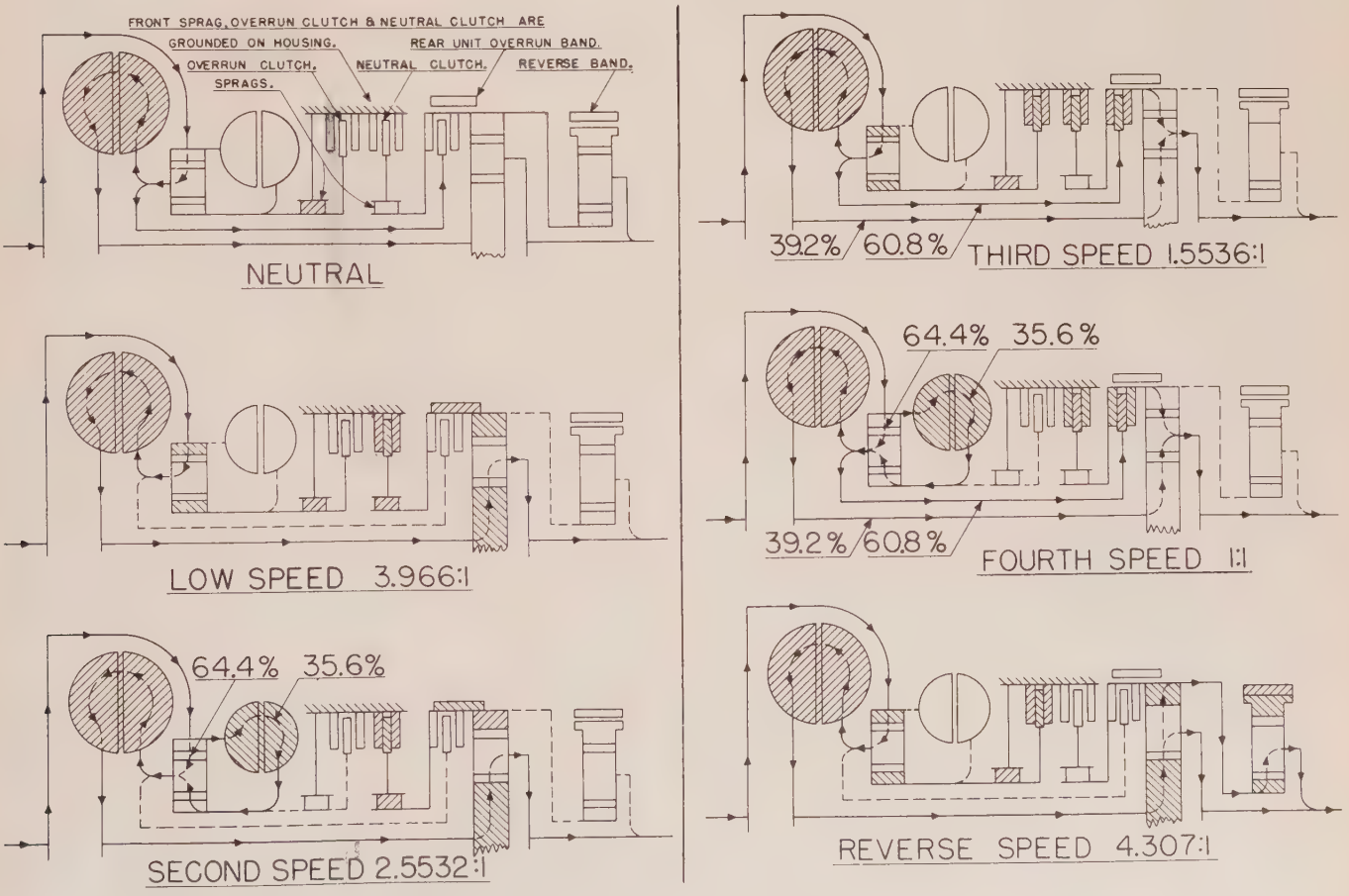


Fig. A—Operation and torque paths in various speeds

THE NEW AMERICAN

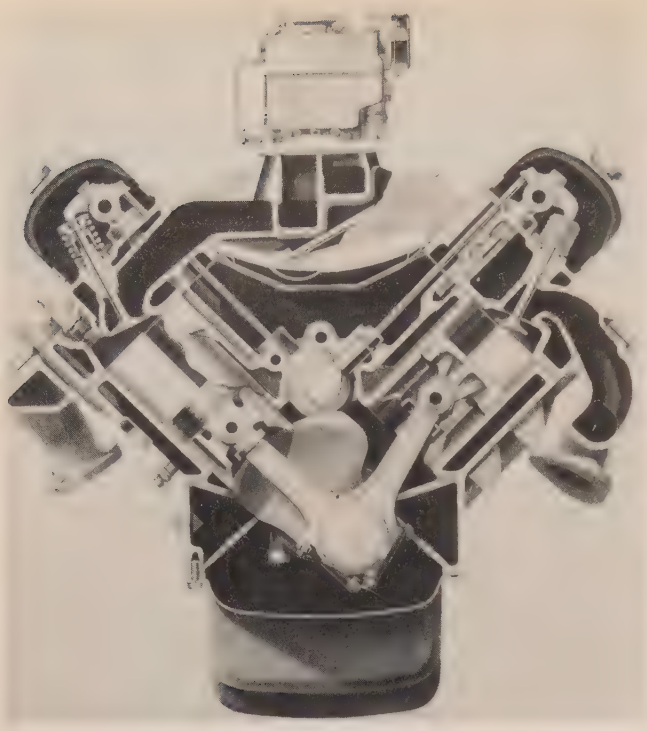


Fig. 1—Transverse cross-section of American Motors V-8 engine

ENOUGH has been said about V-8 engines, to preclude any discussion in this paper about the inherent advantages and disadvantages of this type of design. Rather, we shall attempt to set forth, not only the end results of our design and development efforts, but some of the more interesting problems involved in bringing the new American Motors V-8 engine to the automotive market.

Many of the desirable design features pioneered by Nash and Hudson on our 6-cyl powerplants, have been incorporated into this new V-8. We have drawn from a wealth of experience that includes 39 years of overhead-valve production at Nash, and the well-known ability of Hudson to develop high-performance engines. These features have been retained, in addition to ideas resulting from a very careful and thorough analysis of the newest industry-wide practices and methods. The above, together with a great deal of forward thinking, has produced an engine that will power a new series of cars—the Nash Statesman V-8, and the Hudson Hornet Special V-8 for 1956.

END results of development efforts and problems on the new American Motors V-8 engine are discussed here.

The problem of constructing an engine that is economical, low in weight, durable, and flexible is approached through redesign of the complete engine.

Basic Design and Flexibility

In addition to producing an engine with high performance characteristics, we had basically four other prime objectives in mind when we started to put together the production version of our various experimental projects, engineering studies, and previously approved designs. We have through the years built a reputation for economical and dependable engines, and it was not our intention to sacrifice this or any other past objective in the new design. In short, we also wanted:

1. An engine that would be flexible enough to be readily adaptable to future displacement requirements, compression-ratio changes, and any of the other forward reaching revisions of the automotive industry today.

2. An engine that could be easily installed in our present and future bodies, and would also lend itself to our methods of production assembly. At the same time, we wanted an installed engine that would be readily accessible for service.

MOTORS

V-8 ENGINE

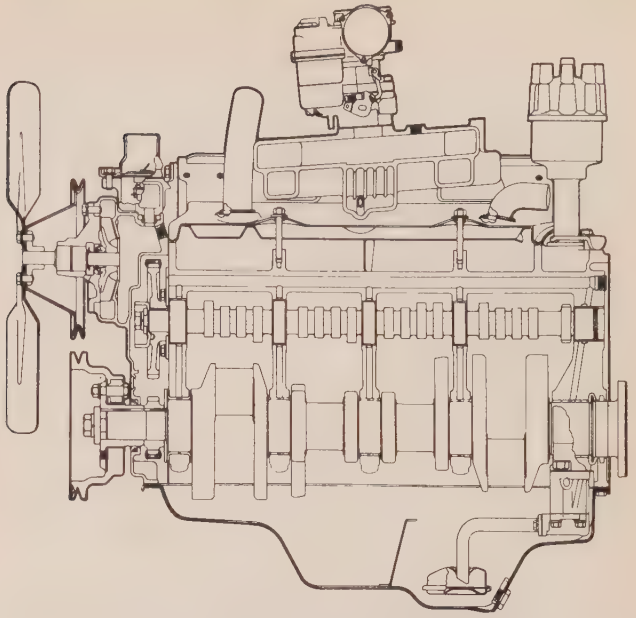


Fig. 2—Longitudinal cross-section

John F. Adamson, Carl E. Burke, and David B. Potter,
American Motors Corp.

This paper was presented at the SAE National Passenger-Car, Body, and Materials Meeting, Detroit, March 7, 1956.

3. An engine that incorporated the latest and most economical methods of manufacturing processes.

4. An engine with the lowest possible weight, without sacrificing durability.

The new 3½-in. bore by 3¼-in. stroke, 250-cu in. V-8 (Figs. 1 and 2), was designed to accommodate greater displacement than is actually being produced this year. As examples, the valve sizes were determined by future breathing requirements, as were both the intake and exhaust port sytems. The crankshaft forging was designed to give 100% balance for much heavier reciprocating and rotating weights than now carried, and the water and oil pumps have a greater capacity than presently required. The cylinder center distance of 4¾ in. is considerably larger than that required for a 3½-in. bore, and allows for future increases without sacrificing coolant flow around the cylinder walls. Main bearing caps and bolts, and connecting rods, have likewise been designed for a greater load-carrying capacity.

The above features, together with other factors, will allow for increased engine displacement without extensive tooling changes. The crankshaft balancing equipment would be revised only with regard to "bobweights," while no changes are needed on the block other than the pattern and machining revisions for bore size. As mentioned previously, the valves have been actually designed for a greater displacement, and consequently, the only change required in the cylinder head for shifting to a bigger engine would be the revising of the cast combustion chamber.

Engine Size

With regard to engine size, we set out to produce an engine that could be easily installed in not only our present line of cars, but also those projected for the future. From the styling standpoint, it had to be a low engine, and from the assembly installation standpoint, it was desirable to minimize width and length.

These objectives were obtained by making the term "compact" the byword of the entire project. A short stroke of 3¼ in. was chosen to reduce height and width, and the crankshaft counterweights are contoured to further shorten the distance from the crankshaft centerline to the top of the block. Exhaust manifolds are carried below the port openings in the head, and the intake port facings are held parallel to the cross-sectional centerline of the banks to further reduce the height of the carburetor flange.

As shown in Fig. 3, the above items allowed us to design an engine that has a width of 24⅛ in. across the exhaust manifolds, and a height from the crank-

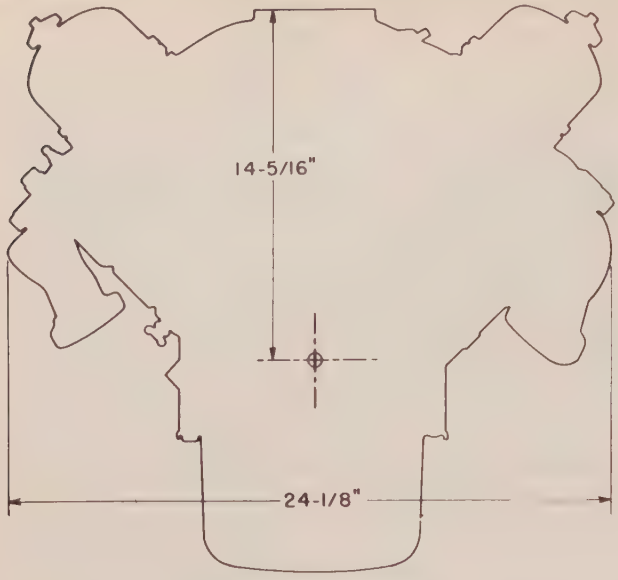


Fig. 3—Engine outline

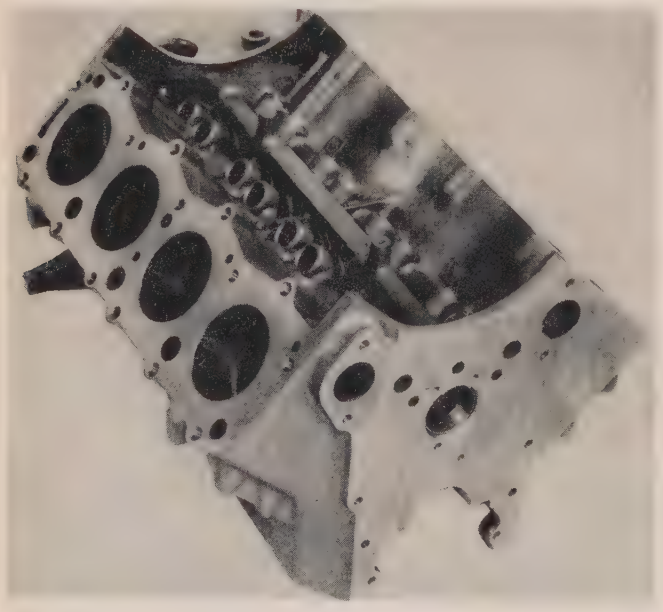


Fig. 4—Cylinder block

shaft centerline to the carburetor flange of $14\frac{5}{16}$ in. The overall length from the back face of the engine to the center of the fan pulley is $27\frac{23}{32}$ in. It must also be borne in mind that these dimensions will remain unchanged for an engine of greater displacement than the present 250 cu in.

Although servicing an installed engine is one of the problems of V-8 design, a number of features were adopted to ease this problem. Spark plugs are located well above the exhaust manifolds, and are in a nearly vertical position for easier removal and installation. Service consideration was also given to the generator, as it is located at the upper right-hand side of the engine and outside of the tappet cover for easier accessibility. In addition, the distributor has a tang length,

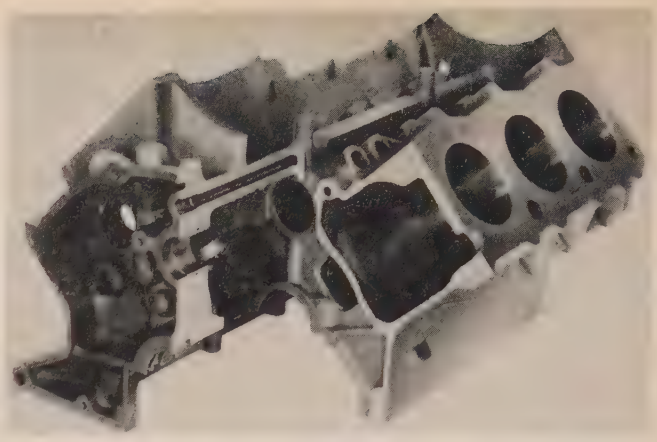


Fig. 5—Cutaway of cylinder block showing design for rigidity

for mating with the oil pump shaft, that allows this engagement to be made prior to engagement of the distributor gear with the camshaft drive gear.

Manufacturing

Exceptionally close liaison was maintained with our manufacturing personnel during the design and development stages of the engine. As a result, many economies in fabricating and tooling processes were built into the original design and have since been carried through into production.

Tooling facilities for the V-8 are completely new, and are based on what we call "segmented automation." In this type of manufacturing, each basic section of tooling, although completely automatic, is not fully integrated with other sections. For our purposes, this type of tooling means increased flexibility as each portion of the line can be utilized independently of other operations.

Of particular interest is the cylinder-block boring equipment, which has been designed to finish simultaneously blocks of two different bore dimensions. It contains two sets of roughing, finishing, and chamfering tools, and blocks of either bore size can enter the equipment in any mixed sequence. Each station is set to tool one size bore, and when a block enters that station, a probe automatically determines whether or not to cycle the cutting heads.

Engine Weight

The many features incorporated to keep engine height and length to an absolute minimum have aided tremendously in reducing engine weight. In addition, every possible design detail has been very carefully investigated for possible further weight reduction. Loading and stressing of all parts was painstakingly calculated to make sure that we were not guilty of overdesigning.

Regarding the loading and stressing of parts, all components, where applicable, were considered for use with the larger displacement versions of this en-

gine. Consequently, even though the weight of the present production engines compares very favorably with competitive powerplants, the advantages of our weight-saving program will be much more apparent in future versions of this engine.

Actual weights of the various major component parts are listed in Table 1.

Cylinder Block

As any engine is basically built in and around the cylinder block (Fig. 4), this component is the biggest factor in regard to powerplant size, weight, and flexibility. Even engine harshness and durability of the drive-train parts depend on the rigidity and design of the block. In the American Motors' design, every effort has been made to provide a satisfactory base upon which to build the engine.

The crankcase flange has been carried 2¾ in. below the crankshaft centerline to provide inherent stiffness and a good oil pan sealing flange. The flywheel housing mounting surface provides a wide and deep base for drive-train mounting, and the 30 cylinder-head bolts give a pattern that carries the gas pressure load evenly into the water jacket walls rather than into the cylinder bores. This arrangement reduces distortion and consequent abnormal wear of the bores, pistons, and piston rings. (See Fig. 5.)

Due to the fact that the block comprises the greatest weight of any component in the engine, it provides a lucrative field in which to effect weight reduction. We have taken advantage of this fact wherever our tests and previous experience has shown it to be possible. Working from the bore-to-bore distance of 4¾ in. established by future displacement needs, the overall length of the block has been held to 23.02 in. by careful consideration of coolant and loading needs, as shown in Fig. 6. Windows have been cast in the main bearing webs where strength requirements showed it possible, and as can be seen in Fig. 7, pockets have been cast on either side of the rear flange for further weight reduction. The right pocket nestles the starter close to the crankcase and allows the starter to be fastened to a light-weight die-cast aluminum flywheel housing. It is obvious that we cancel several machining operations for a starter pad normally found on the block, and in addition omit a considerable amount of cast iron. We estimate that a saving of approximately 15 lb of metal has been realized on the rear of the block alone.

You will notice in Fig. 8 that manufacturing savings have been obtained by eliminating hard sand cores wherever possible. Green sand has been provided for the front, side, top, and tappet chamber sections, and hard sand cores for only the crankcase, bores, water jackets, and rear face. All cores are machine set to minimize breakage.

Machining operations were also taken into account during the design stages of the engine, which resulted in the complete lack of any machining setup carried out for one drilling or surfacing operation only.

Combustion Chamber

We have chosen for this engine, a type of chamber that gives us what we consider the more important of

Table 1—Engine Component Weights

Components	Weight, lb
Engine complete (less transmission)	601
Cylinder block	160.9
Crankshaft	62.8
Cylinder head (each)	49.5
Exhaust manifold (including crossover)	27.9
Intake manifold	27.8
Front cover	14.7
Camshaft	10.2
Connecting-rod assembly	1.72
Piston	1.13
Valve train (including lifter, push-rod, rocker arm, intake valve, valve spring, retainers)	1.09

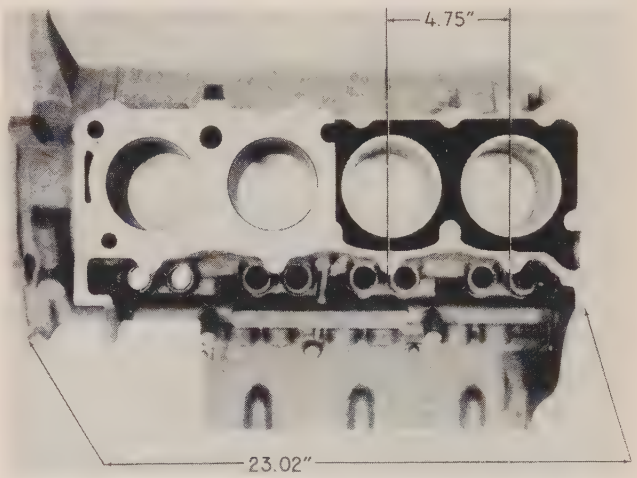


Fig. 6—Cylinder-block dimensions

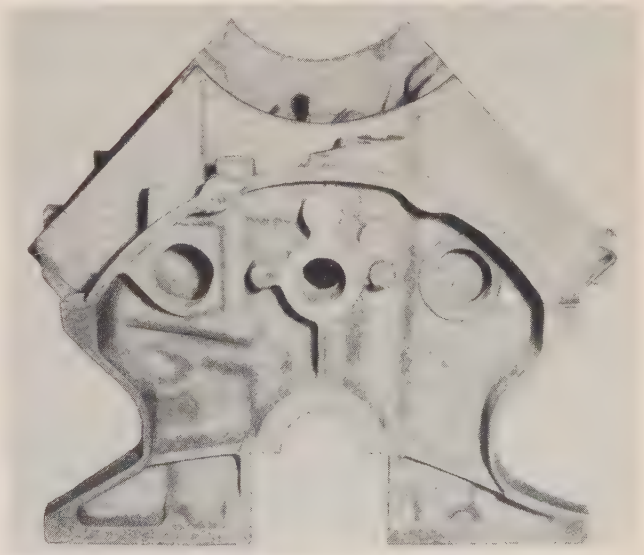


Fig. 7—Cylinder-block rear face

Fig. 8—Cylinder-block core assembly

Fig. 9—Cast combustion chamber

the various characteristics obtainable for a good, efficient combustion chamber.

The design used can best be described as a kidney-shaped, wedge-type, cast chamber. Excellent results have been obtained on our recent 6-cyl engines with this type of configuration, and it inherently contains a number of the factors we desired. Being cast, it requires a minimum of machining, and consequently, volume can be placed just where we want it. The kidney shape gives a swirling action to the intake gas for better turbulence, and spark-voltage requirements are quite low. As shown in Fig. 9, there is no shrouding of the valves and, therefore, a high volumetric

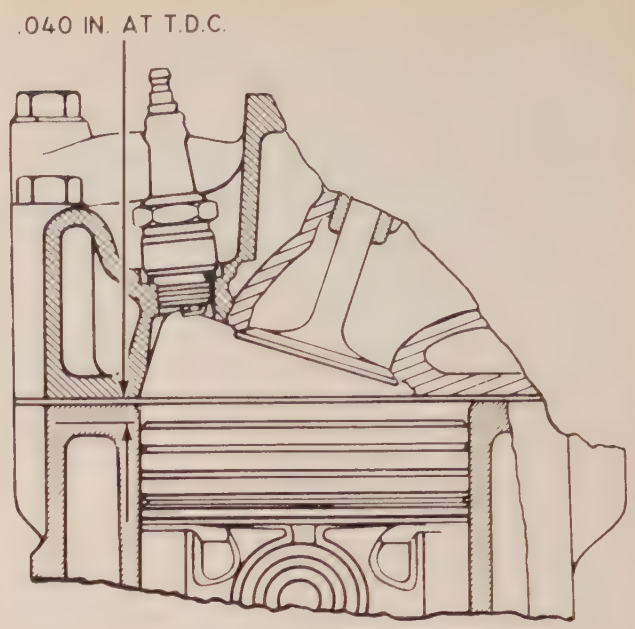


Fig. 10—Combustion chamber

efficiency is obtainable. Combustion characteristics are such that chamber shape controls the rate of pressure rise to minimize engine harshness. A compression ratio of 8.01 has been chosen to allow this engine to burn regular fuel and, therefore, further effect economy of operation.

As can be seen from Fig. 10, the spark plugs are effectively cooled by the considerable volume of water surrounding them. These plugs are located in the chamber in such a manner as to minimize the "drowning effects" of unvaporized fuel during cold starts. Fig. 11 shows the location of the plugs in the head.

Ports and Manifolds

The purpose of induction and exhaust systems is efficiently to feed and scavenge an engine under all driving conditions. The intake manifold must be much more than just an inlet to a high-speed air pump, and the exhaust system must be scientifically balanced for the load it carries.

Many major and minor changes were made in the intake manifold for this engine before the production version was agreed upon. Numerous cold and hot starting tests, plus runs under all operating conditions, were made in order to provide a manifold that would give the average driver a "satisfactory feeling" engine. Fig. 12 plots the air-fuel distribution by this manifold to the various cylinders under the conditions shown.

Fig. 13 shows the basic configuration of the intake and exhaust port designs. Starting with the intake port opening in the head, it can be seen that a smoothly balanced transition is obtained in relation to the valve opening. The slight restriction of flow and consequent increase in gas velocity tends to develop a ram effect at the valve, with subsequent increase in engine

torque. Valve size is such that the theoretical gas velocity through the valve is lower than that of any automotive engine on the market today. The full-throttle variation of volumetric efficiency with speed is shown in Fig. 14.

It can be seen that exhaust ports have also been designed to give balanced areas. Port size is increased as gases move away from the valve to give efficient scavenging and minimize back pressure. Exhaust manifolds are ample in area and provide for a smooth, continuous flow of exhaust gases.

Crankshaft

The crankshaft is a steel forging with five main bearings and six counterweights, as shown in Fig. 15. Shaft length from the flywheel flange to the front edge of the sprocket shaft is 27.2 in., and a journal overlap of ¾ in. provides increased stiffness.

As previously mentioned, all counterweights are cam-turned to realize fully every possible reduction in engine height. In addition, the checks are chamfered to allow the contoured skirts of the pistons to nest closer to their respective crankpins when at bottom dead-center. This in turn, shortens connecting-rod length, and further reduces engine height. Depressions are cast in the bottom jacket walls of the cylinder block to allow for clearance with the counterweights.

Fig. 16 shows the torsional vibration characteristics of this engine both with and without a harmonic damper. As can be seen, the rubber and friction type of damper used reduces all orders of frequency within the engine speed range to negligible amplitudes.

A particularly interesting problem in the design of the engine centered around the balancing of the crankshaft. The crank forging itself had to contain enough inherent balance control to take care of future displacement requirements. However, at the same

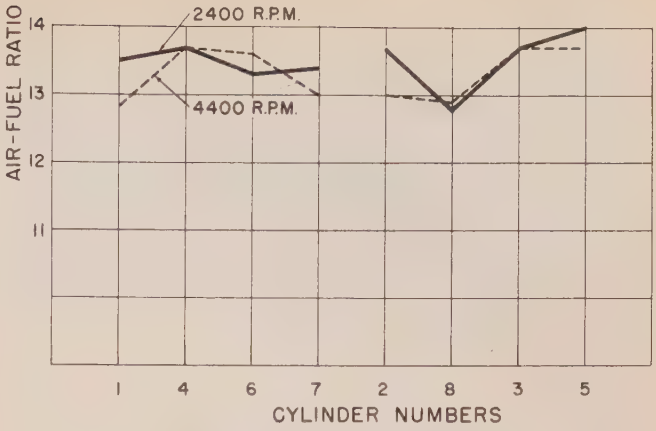


Fig. 12—Fuel/air ratios

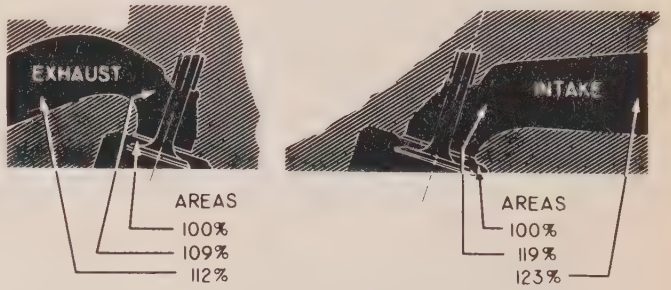


Fig. 13—Intake and exhaust port areas

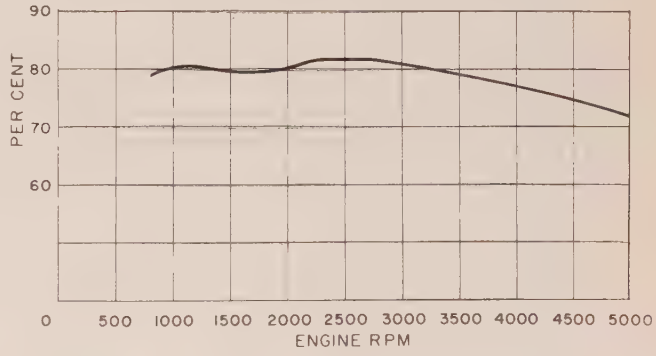


Fig. 14—Volumetric efficiency at full throttle

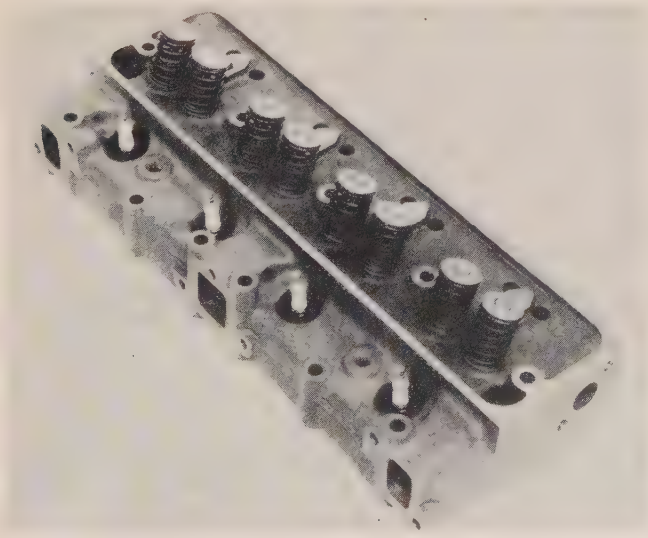


Fig. 11—Cylinder head showing location of spark plugs

Fig. 15—Crankshaft

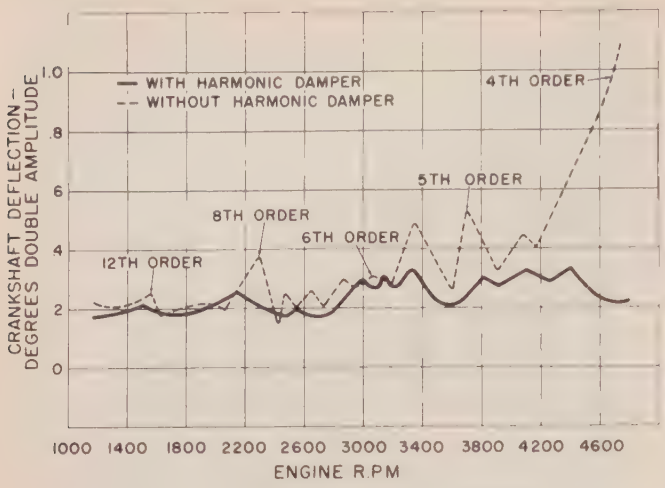


Fig. 16—Crankshaft torsional vibrations

time it could not have more overbalance when used with the engine under discussion, than our balancing equipment could remove. Reciprocating and rotating weights, or what are more commonly referred to as "bob-weights," were carefully calculated for all contemplated displacements prior to the final approval of the crankshaft design. This design was made in such a manner that no tooling changes will be needed prior to the initial balancing of the crankshaft itself. For engines with displacements of over 250 cu in., a very minor resetting of the balancing equipment is all that is needed.

Fig. 17 shows the vectors involved in the balancing operations on the crankshaft, with the right-hand portion representing the initial balancing on the machined crank. This operation consists of drilling two 1⅝-in. diameter holes into the faces of both No. 1 and No. 8 cheeks, and reduces the overbalance of the crankshaft to a value of 7.5 oz-in. in a predetermined plane. This figure shows the maximum correction factor of 31.1 oz-in. obtainable with each drilled hole, plotted as a vector in both magnitude and direction. The ensuing parallelogram contains the area in which balancing correction can be made to the 7.5 oz-in. value.

The designed balance of the machined crankshaft, including "bob-weight" equivalents, is represented by the vector shown. From this vector, there is a permissible variation of 9.4 oz-in. due to possible forging and machining variations. This vector must fall within the correction capacity of the two 1⅝-in. diameter balancing holes. The depth of the drilled holes, will, of course, vary with the correction needed.

As engine displacement is increased, "bob-weights" obviously become heavier, and the balance control of the crankshaft is reduced below 32.2 oz-in. However, as long as the permissible variation circle struck from the end of the crankshaft balance-control vector falls within the parallelogram formed by available correction, the same finished crankshaft can be used.

The crankshaft is then assembled in the engine with connecting rods, pistons, piston pins, piston rings, flywheel, and crankshaft pulley attached. This assembly is then corrected for the 7.5 oz-in. left by the initial balancing, plus variations due to weight and balance tolerances of the attaching parts. The correcting is done by drilling two $^{15}/_{16}$-in. diameter holes radially into the periphery of the No. 1 and No. 8 cheeks, with the location and depth of the holes varying with the amount of correction needed. The left portion of Fig. 17 plots the vectors representing the various corrections obtainable due to location and depth of drilling,

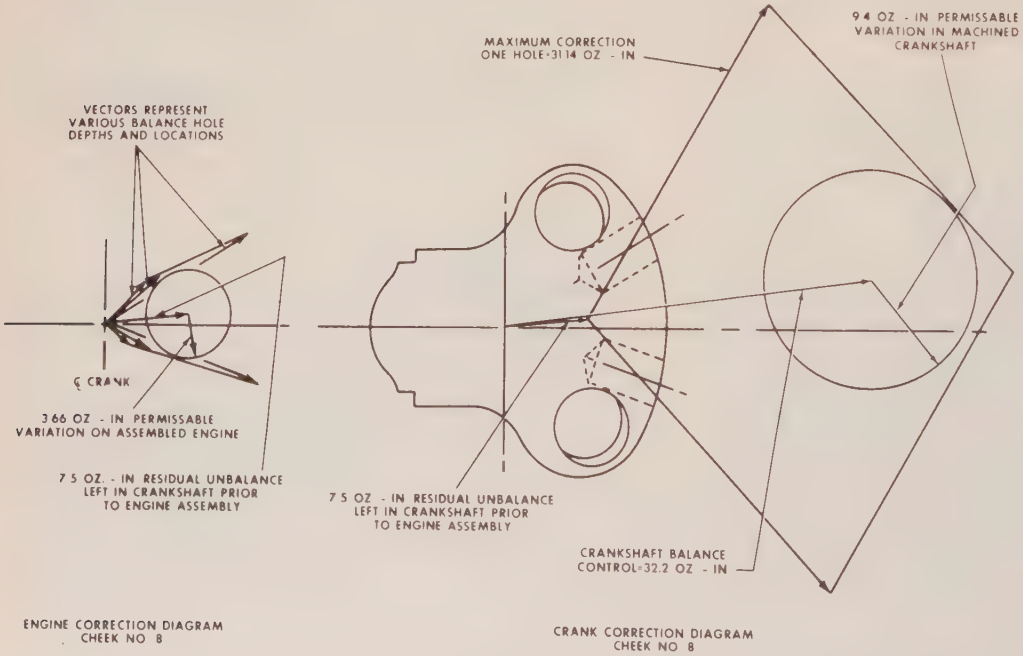


Fig. 17—Crankshaft balancing

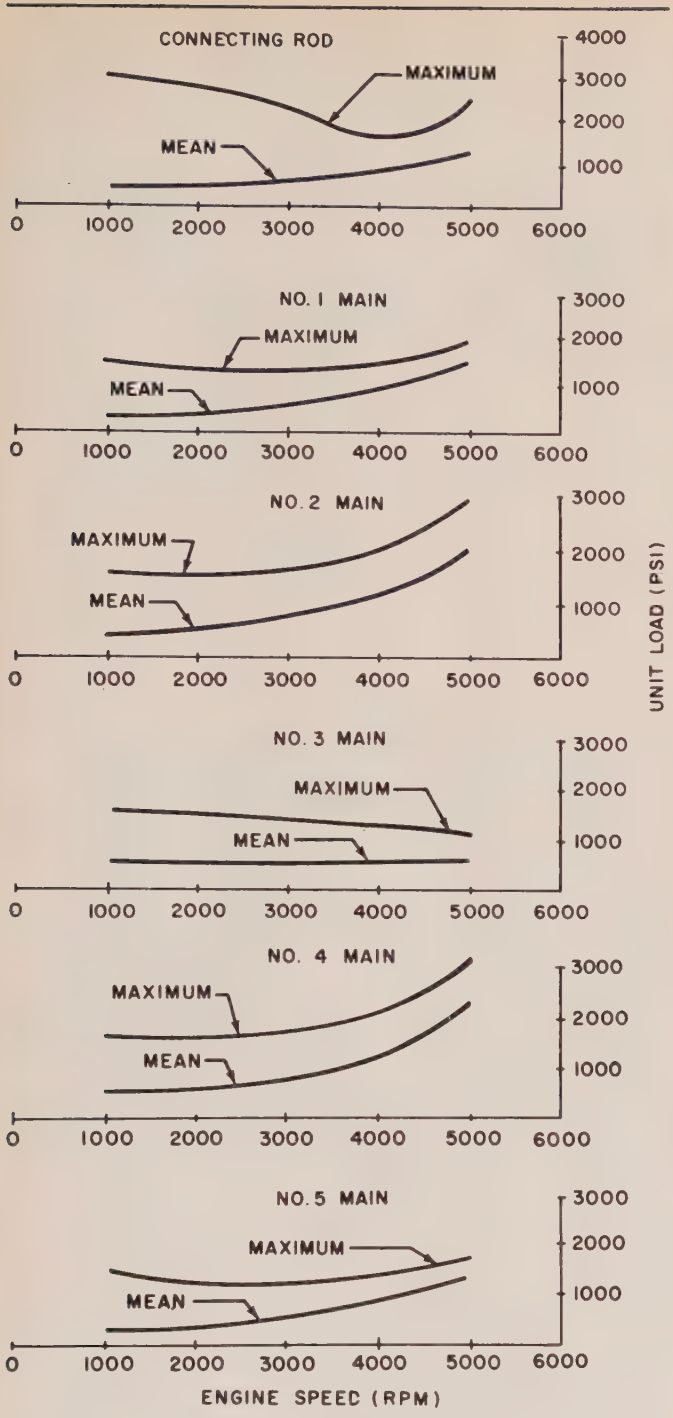


Fig. 18—Bearing loads

and a 3.66-oz-in. circle to cover the permissible component variations. This final operation balances the engine assembly to within a maximum tolerance of plus or minus ½ oz-in.

Bearings

Further economy and assembly simplicity have been designed into the engine through the use of bearing interchangeability. All five main bearing diameters are 2.499 in., and, with the exception of the front which is flanged to take crankshaft thrust, are

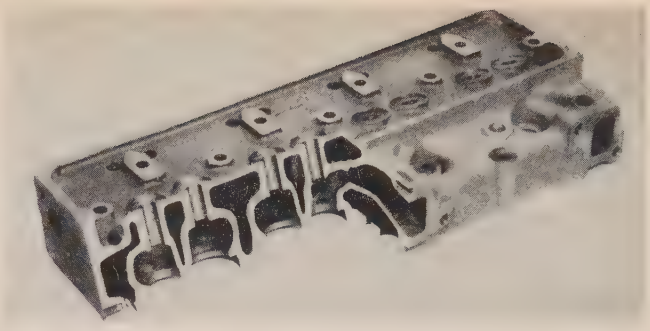


Fig. 19—Cylinder-head water passages

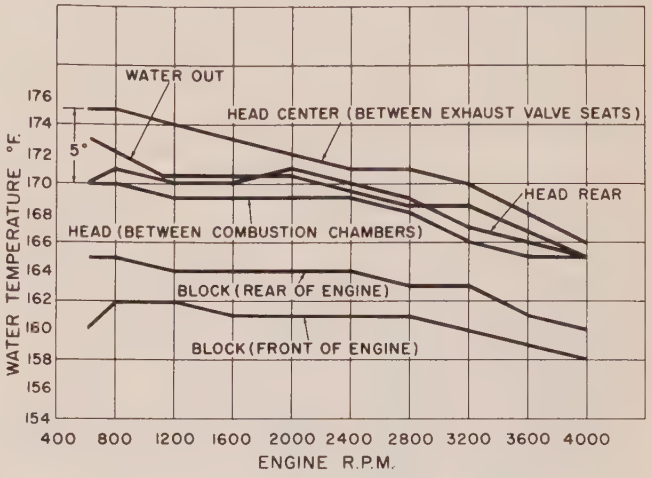


Fig. 20—Coolant temperature variations

interchangeable. Effective bearing area for the intermediate and rear mains is 2.20 sq in., while the front is 2.08 sq in. Full 360-deg grooving is used to make the bearing halves similar, and also to insure adequate flow of oil to the connecting rod at the time of registration for cylinder-wall lubrication.

Connecting-rod-bearing halves are also similar throughout. Bearing diameter is 2.249 in., and effective area is 1.935 sq in.

All bearings are micro-babbitt with steel backs. Mean and maximum loads carried are plotted for the engine speed range in Fig. 18.

Cooling

Engine durability is dependent in many ways on an efficient cooling system. Fig. 19 is a cutaway section through the valve centers in the head, and shows the results of the attention given to getting water around the valve seats. With this penetration of coolant, we obtain longer life and more trouble-free operation of the valves. Complete jacketing of the cylinders provides cooling for the entire length and circumference of the bores. Water flow has been designed to give complete coverage of the various portions of the engine, and as shown in Fig. 20, temperature variations of the coolant have been kept to a minimum.

A single, high-capacity pump provides the flow

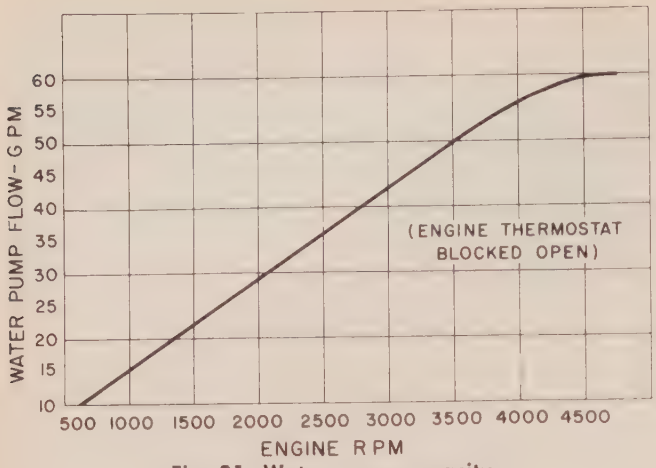


Fig. 21—Water pump capacity

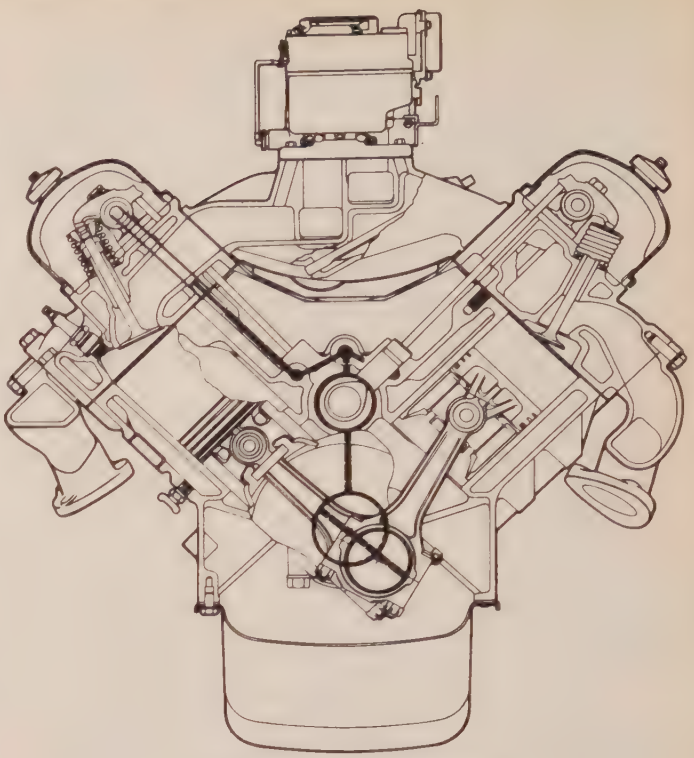


Fig. 22—Lubrication system

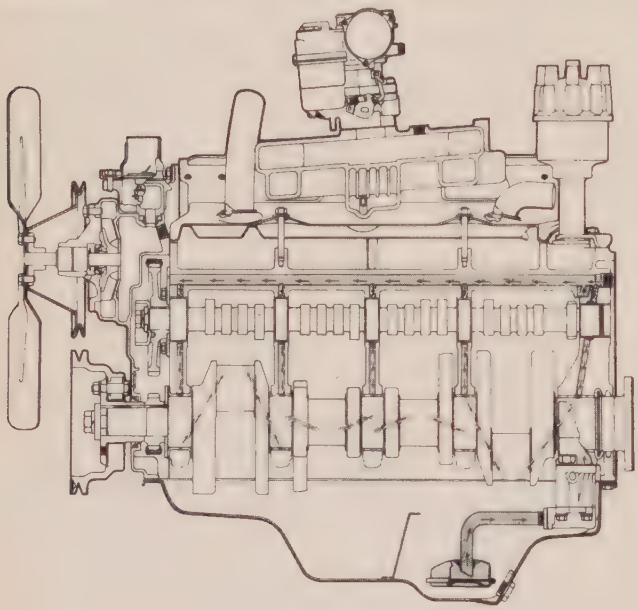


Fig. 23—Lubrication system

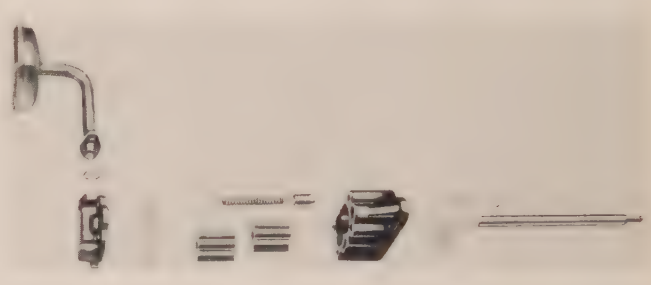


Fig. 24—Oil pump

plotted in Fig. 21. This pump contains a 4-in. diameter plastic impeller, and is centrally mounted at the front of the engine. A drilled bypass is provided in the water manifold for coolant circulation during cold starts, and prior to the opening of the thermostat.

The attention paid to the various cooling problems involved has resulted in relatively low heat-rejection characteristics. This, in turn, means high thermal efficiency and reduced required radiator capacity.

Lubrication

The basic flow of oil for the engine is shown in Figs. 22 and 23. It can be seen that the lubricant is picked up by the fixed-screen inlet and drawn into the pump. This pump, shown in Fig. 24, contains sintered iron gears and is an integral part of the rear main bearing cap. This type of design gives us a pump located well up towards the crankshaft centerline, and allows placing the oil pan sump in a variety of fore-and-aft locations. This, in turn, gives a great amount of flexibility with regard to the location of the steering linkage. The pump is driven through a tongue and groove connection with the distributor shaft, and is provided with an oil pressure relief valve.

From the pump, the oil is forced through the filter and into the main oil gallery. Flow is then down to the $\frac{7}{64}$ x $\frac{5}{16}$-in. annular grooves machined in the camshaft bearing webs, and thence to the main bearings. Except for oil fed to the camshaft bushings, it can be seen that the main bearings are fed prior to any oil being bled off to another location. This assures that the bearings farthest from the pump receive adequate lubrication.

It might be interesting to note Fig. 25, which shows the intermittent lubrication of the chrome-plated fuel pump eccentric. This timed squirting of the eccentric is accomplished when the holes drilled in the front

camshaft journal register with the camshaft bushing oil hole. Lubricant under pressure is then forced through a $^{3}/_{32}$-in. diameter hole in the camshaft sprocket, and onto the pump eccentric. Aside from this same spray lubricating the timing chain, oil is picked up by the cast-in dam on the front cover and directed to the crankshaft sprocket.

Fig. 26 contains actual photographs taken from a series of high-speed motion pictures showing cylinder-wall lubrication at 0 F, 40 psi, and 700 rpm. From these movies of various experimental designs, we were able to get a squirt pattern of the correct timing and intensity to provide adequate cylinder-wall and piston-pin lubrication.

The hydraulic lifters are fed through longitudinal galleries that intersect the lifter bores. These galleries are fed from the main gallery through grooves cast in the cast-iron camshaft thrust plate. Due to the fact that no lifter pump-up problems have been encountered in this engine, no attempt has been made to meter appreciably the flow of oil to the lifter galleries.

The hollow rocker-arm shafts in each bank receive their oil supply from the lifter galleries, through short drilled holes in the cylinder-head and rocker-arm support bolts. These bolts are set in oversize drilled holes and the oil is carried around the bolt to the rocker shafts. This movement of oil is at the rear of the engine to provide continuous flow through the lifter galleries and thus to prevent the formation of sludge traps.

An interesting approach to the matter of valve-stem lubrication during cold starts can be seen in Fig. 27. You will note that the rocker arms have a milled flat which intersects the oil feed hole. We have found that this flat tends to break the surface tension of the cold oil oozing from the feed hole, and causes it to flow freely to the valve end of the arm. Tests have shown that the time required to get oil to the valve stem during cold starts, has been cut over 75% from our previous design. This milled flat also causes hot oil to

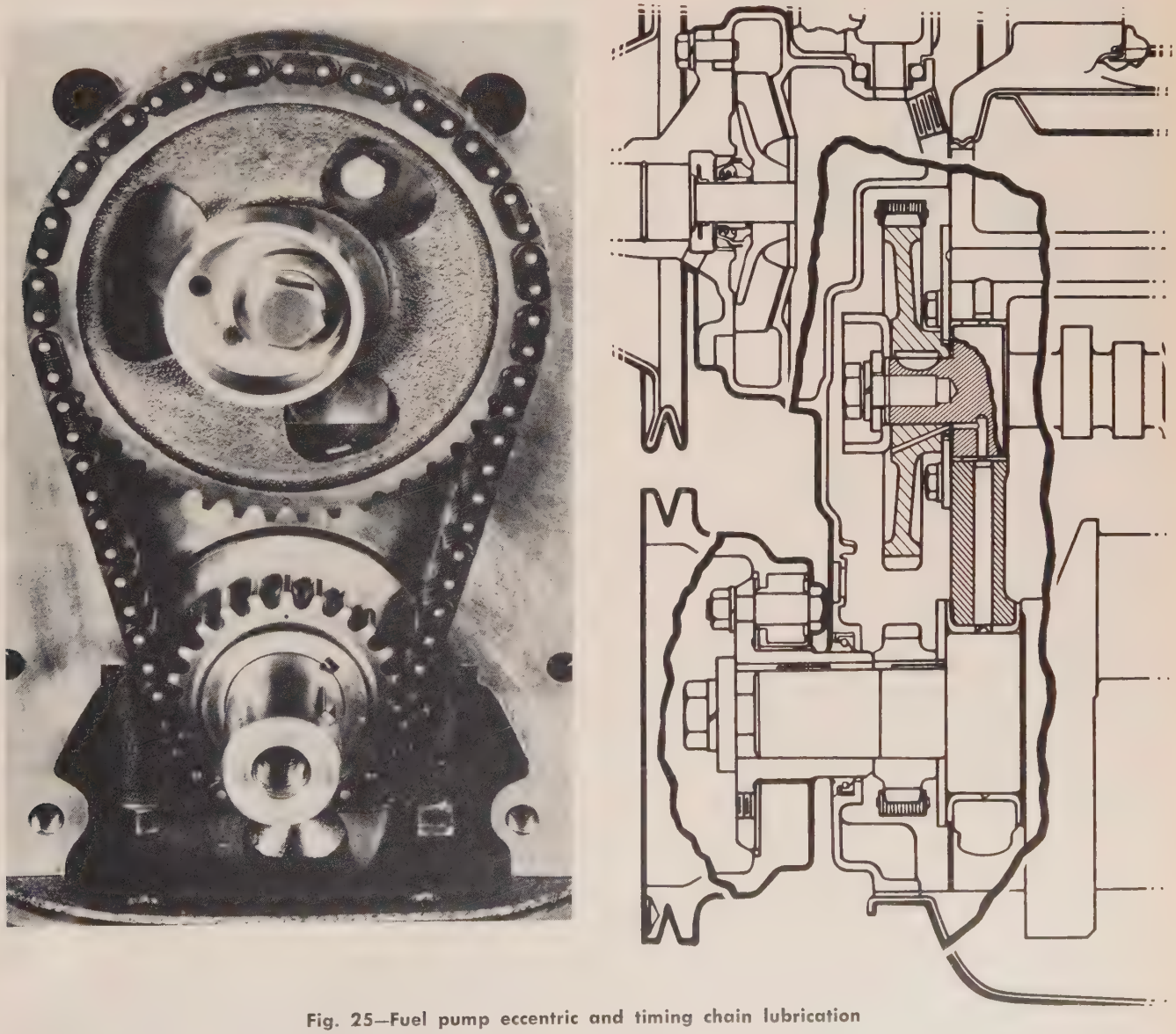


Fig. 25—Fuel pump eccentric and timing chain lubrication

Fig. 26—Cylinder-wall lubrication

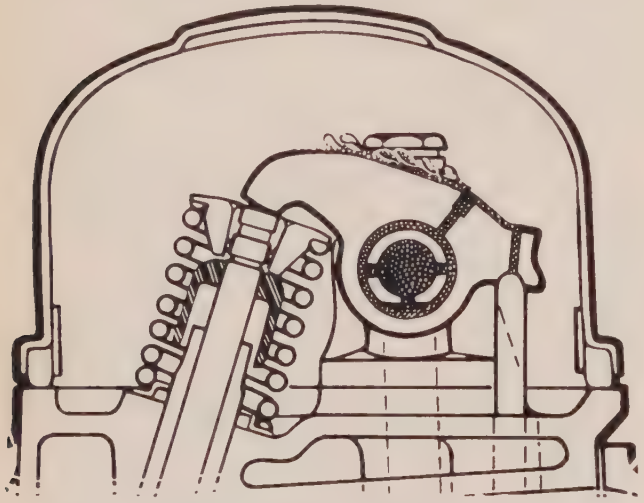


Fig. 27—Rocker-arm lubrication

spill over the sides of the arm and not to run down and overlubricate the valve stems. Oil-resistant rubber deflectors are fitted to the valve stems to prevent excessive oil being fed to the valve guides. These deflectors move with the valve stem, and allow oil mist to lubricate the stem and guide when the valve is in the closed position.

Distributor lubrication is supplied through a drilled hole in the distributor body, which registers with the right-hand lifter gallery. Oil is fed through the distributor body into the distributor bushing, and carried to the gears by means of internal grooving in the bushing.

Piston and Rings

Following our previous practice, the new V-8 uses aluminum-alloy, steel-insert, Autothermic pistons. These pistons have slippered skirts, are tin-plated, and have the piston-pin boss holes bearingized after plating. The latter assures an accurate bearing area for the pin, and helps prevent scuffing.

As can be seen in Fig. 28, these pistons are of the double-slot design, and have three vertical ribs joining the head and pin bosses. These ribs support the bosses and tend to reduce deflections during the periods of high pin loading. The slipper-type skirt is shaped to match closely with the contoured counterweights of the crankshaft when the piston is at bottom dead-center. This serves further to reduce the required height of the cylinder block.

The piston carries a head thickness of 0.25 in. to allow for its use with compression ratios well above the current 8.01. A 0.0625-in. offset of the piston pin toward the major thrust face gives exceptional freedom from "piston slap." Piston finished weight is 511 g and the balancing lugs allow for 31 g of weight control.

As also shown in Fig. 28, three rings are used. The compression rings are $\frac{5}{64}$-in. wide alloy iron, with the top ring having a 0.004-0.007-in. thick, lapped chrome plating. The oil control ring is of the three-piece design made up of two rails and a spring steel spacer. The rails carry buffed 0.0025-in. minimum chrome plating on their reacting faces.

Valve Train

The present design is a continued development of quiet, durable, and trouble-free valve trains, and is made up of compact, light, and well-proved components.

The camshaft is an alloy iron with five bearings, and is chain driven. All lobes carry a 15 maximum micro-in. finish, and the cams are ground with a 0.001-0.002-taper per in. The latter, together with spherical lifter faces and offset centerlines, insures positive lifter rotation.

To obtain further quiet operation and peak performance, hydraulic lifters are used. Lifter bodies are hardenable iron, with a face hardness of 54 RC min, and are lubrited for improved break-in. Push-rod sockets are case-hardened steel, and a lifter internal operating range of 0.100 in. is available.

The diameter of the lifter is 0.904 in., which gives an effective diameter of 0.900 in. when the centerline offset, spherical face, and tapered cam lobes are considered. This value is well above the 0.800-in. diameter required to keep the edge of the tappet face from digging into the cam lobe. The 0.800-in. diameter was computed from the cam-lift velocity, and as is apparent, there is ample space to utilize cam contours with more severe lift velocities. Critical pump-up speed for these tappets has been found to be well over 5000 rpm engine speed.

The solid push-rods are ¼-in. diameter, and are made of SAE 1060 cold-drawn steel with the spherical ends hardened to RC 50 min. The overall length is 8.87 in., which assures a very rigid valve-train component.

Rocker arms are cast pearlitic, malleable iron, and are extremely short to reduce valve-train deflection

further. The arms are lubrited after finishing to help reduce scuffing of the rocker shaft. Rocker-arm geometry has been set up to give a minimum amount of slippage between the arm and the valve-stem tip during the periods of high loading. This high loading period occurs at the point of greatest lift acceleration, which on our cam occurs just prior to 20% of the lift. Fig. 29 shows the movement of the contact points along the rocker arm and the valve, with the vertical distance between these curves representing slippage.

The intake valves, which are made of Silchrome No. 1 steel, have 1.787-in. diameter heads and 30-deg seat angles. The exhaust valves are of the two-piece construction with the head and upper stem made of SAE 2112N steel. Exhaust-valve head diameter is 1.406 in., and both valves operate with a moderate 0.375-in. lift.

Because of the compactness and lightness of the valve train, spring forces can be kept relatively low. This, in turn, reduces lifter face stress, and eases lifter and camshaft wear problems.

Valve Timing and Electrical System

Due to the relatively large valves used with the present displacement, valve timing can be kept to conservative values. A top center valve overlap of only 23 deg, as shown in Fig. 30, has proved to be ample for high-speed breathing, and resulting "high end" performance. This small overlap gives the engine excellent idling characteristics and "low end" output.

The electrical components of this engine have been designed to take advantage of a 12-v system. Spark-plug leads have been kept short and carried well above exhaust manifold heat to make the system as efficient as possible. The distributor has been mounted on the right side of the engine to obtain an upward thrust, and consequent elimination of the thrust washer.

Fig. 31 shows the considerable amount of ignition reserve available in this engine, with both new and used spark plugs. The high spark advance setting and moderate compression ratio are both big factors in

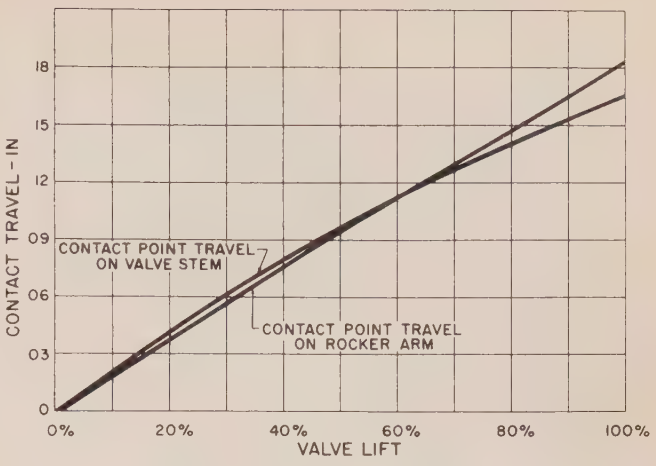


Fig. 29—Rocker-arm geometry

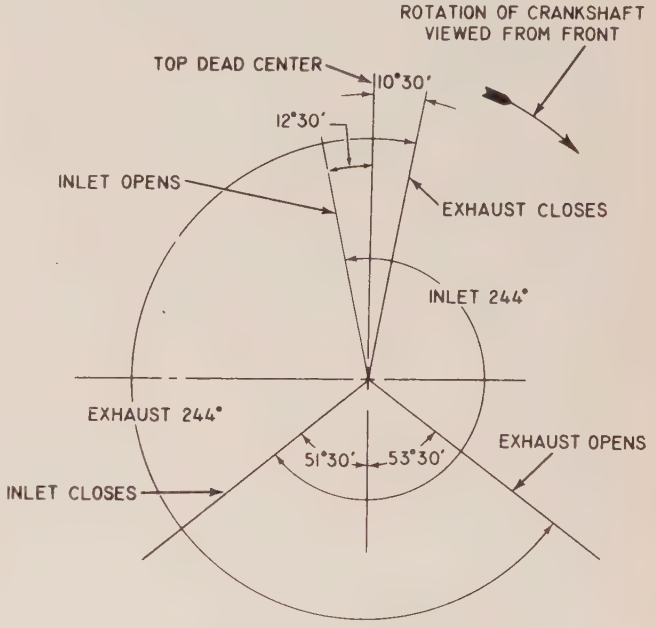


Fig. 30—Valve timing diagram

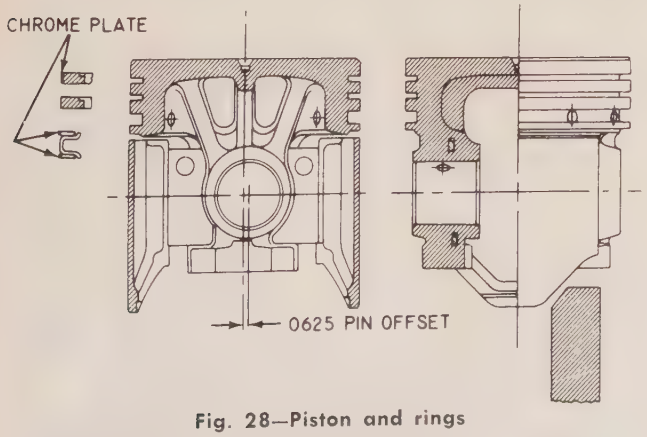


Fig. 28—Piston and rings

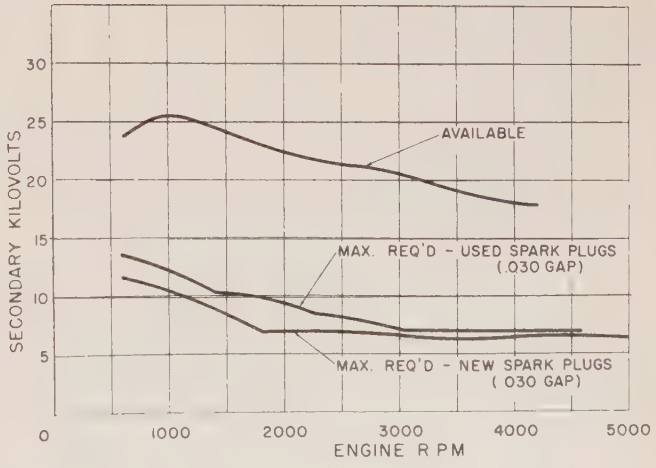


Fig. 31—Ignition reserve

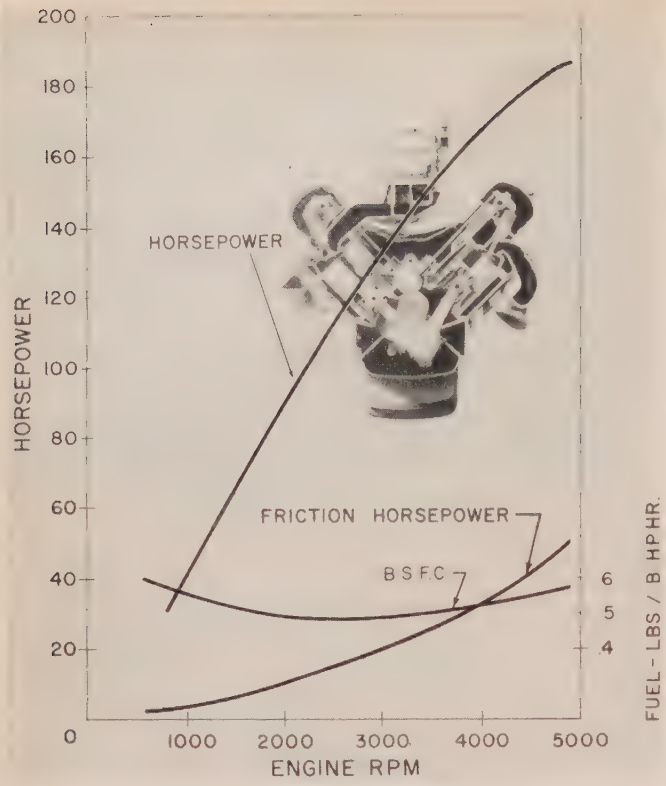


Fig. 32—Full-throttle performance

obtaining this reserve. It is our experience that this relationship of voltage available to voltage required definitely improves cold starting, and helps reduce the ill effects of spark-plug-gap increases and deposits.

Performance

Fig. 32 shows the full-throttle gross performance curves of the new American Motors' engine. These curves are based on dynamometer results obtained with best-power fuel and spark advance, an 8.01 compression ratio, and corrected to standard SAE conditions.

As can be noted in Fig. 32, the peak horsepower of 190 is obtained at an engine speed of 4900 rpm. The friction horsepower curve shows the results of the relatively short stroke design and weight reduction in moving parts, in that it reaches a value of only 34 hp at 4000 rpm.

The specific fuel consumption curve contains the results of attention to the economy factor. It shows a low rate over the entire speed range, and a minimum value of 0.485 lb of fuel per bhp-hr. The fuel fishhooks plotted in Fig. 33 give the carburetor flow requirements of part-throttle operation at 2400 rpm.

The torque curve shown in Fig. 34 depicts one of the most significant features of this engine. Torque is actually one of the more important indications of an engine's ability to give outstanding output, and it also gives the best picture of an engine's real performance in a car. It can be seen that this powerplant has a relatively flat torque curve, and actually shows a differential from peak of only 4% in the engine speed

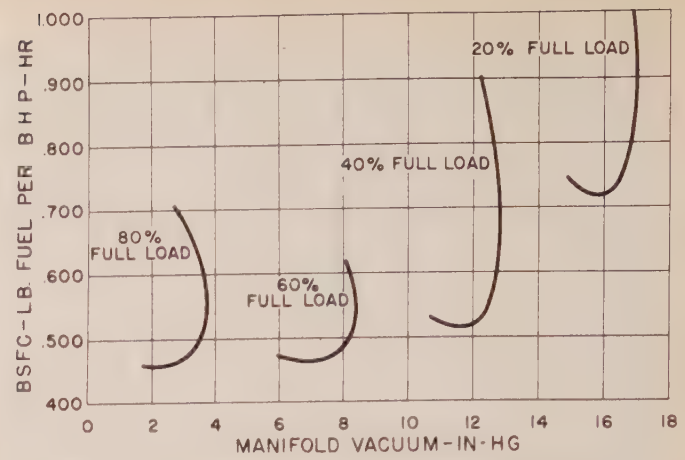


Fig. 33—Part-throttle fuel requirements at 2400 rpm

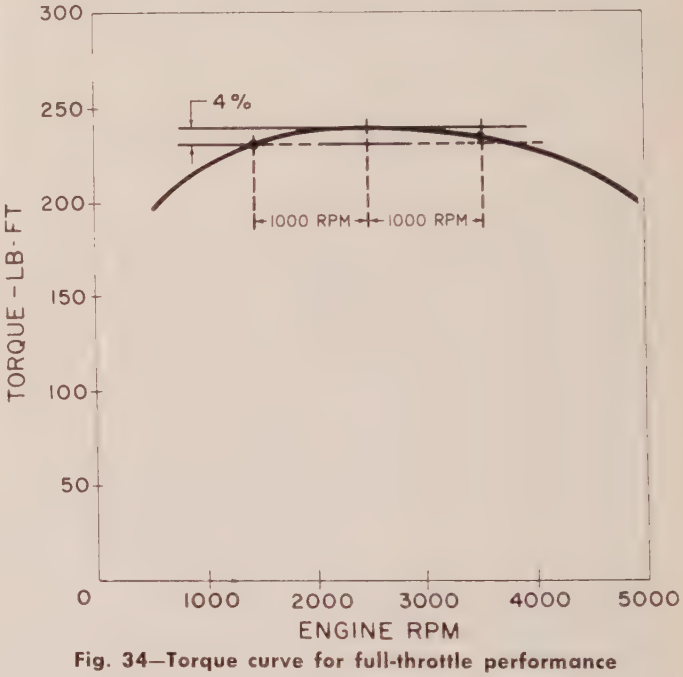


Fig. 34—Torque curve for full-throttle performance

range of 1500–3500 rpm. The maximum torque of 240 lb-ft is obtained at 2500 rpm.

Conclusion

As has been stated earlier in this paper, we had some very definite objectives in mind when we initiated our engine program. Some of these objectives were peculiar to our plans at American Motors, while others were more common with aims found throughout the automotive industry.

In short, our objective was an engine with maximum flexibility with regard to future displacement requirements without sacrificing any of the performance features of the powerplant. Economy of operation and manufacturing processes, along with excellent weight and durability characteristics, were all

part of our aims. We think that we have succeeded in reaching these objectives, and have produced an engine that is second to none. It is an engine of which we are justly proud.

Acknowledgment

The entire program was under the direction of Meade F. Moore, vice-president, Automotive Research and Engineering, and through his efforts the project was carried on cooperatively by our Kenosha and Detroit Engineering Departments. Obviously, such a division of both design and development required the utmost in teamwork by F. F. Kishline, chief engineer, and his assistants, E. L. Monson and J. S. Voigt in Kenosha, and by R. H. Isbrandt, chief design engineer, and W. S. Berry, chief mechanical engineer, in Detroit.

DISCUSSION

Discusses Valve-Gear Problems Of the New V-8 Engine

—Vincent Ayres
Eaton Mfg. Co.

THE new American Motors V-8 engine represents a noteworthy achievement in producing a design to meet satisfactorily a set of specifications.

Manufacture of both L-head and overhead-valve engines for many years acquainted their engineers with the operating differences between the two types of valve gear. Therefore, it is not surprising that in this paper they have discussed the recognized need for valve-train lightness and rigidity which is so necessary for good high-speed performance.

The use of hydraulic valve lifters, and the knowledge that the valve train will operate satisfactorily without pump-up or other disagreeable effects at speeds in excess of 5000 rpm, is a tribute to their design ability. Good high-speed motion can be obtained in an L-head valve gear with high-lift cams having sharp changes in acceleration, due to the inherent rigidity of the parts. We have photographed velocity and acceleration oscilloscope diagrams at high speed that appear to be almost theoretical. However, an overhead valve train may have five to ten times the amount of deflection, and this lowered valve-gear frequency may permit severe vibrations to occur in the operating speed range. The amplitude of these vibrations is greater with a cam which abruptly moves the valve train than when the contour is designed to produce smooth motion. Gradual changes in acceleration or rate of loading are most desirable for a cam in overhead-valve systems. Hydraulic lifters cannot be used in a valve train that will not behave at high speeds. The cam contours used in the American Motors V-8 engine were developed to produce gradual changes of acceleration in order to minimize the high-speed valve-train vibration. The vibration which does occur is of low amplitude, which is within the tolerance of the hydraulic lifter.

The material selected for both the camshaft and tappet, known to the industry as hardenable cast iron, will prove to be a worthwhile choice. This is the combination in most widespread use, and has the ability to resist wear, scuffing, and spalling under varied types of service with the variety of lubricating oils currently on the market.

The Parco-Lubrite tappet face coating and the controlled cam surface finish meet with recognized standards for satisfactory break-in.

The problem of adequate valve-stem lubrication without excessive oil consumption or valve-stem scuffing was solved by American Motors engineers on engines formerly produced. This knowledge assisted them in arriving at a satisfactory combination for their new V-8 engine. It is my understanding that they will continue to use valve stems with a "characterized" surface which is produced by controlled grinding, which results in a surface finish of approximately 40 microin. This provides needed oil-carrying capacity under conditions of borderline lubrication that would result in scuffing or valve squawk if both valve stem and guide were smooth. The intake valve material of Silcrome No. 1 and the 2112N material for the exhaust-valve head have adequate strength and corrosion resistance properties to meet passenger-car service requirements.

Designing Bearings For the New Engine

—Edwin Crankshaw
Cleveland Graphite Bronze Co.

SINCE our association with this engine development is from the bearing standpoint I would like to apply my comments to those things which affect bearing performance.

Early in the development stages of this engine Mr. Adamson requested that we assist him in investigating the bearing design aspects. An early look at the bearing requirements of this engine allowed the flexibility of choosing bearing sizes to not only suffice for this particular engine but also to permit engine power growth facing a major bearing dimensional change.

This close collaboration resulted in a thorough evaluation of bearing requirements, much of which was based on calculations as to the bearing loads, developed bearing oil film thicknesses, bearing operating temperatures, and oil flow requirements.

The set of bearing sizes and design which was finally agreed upon were woven into the design of this engine, and it is quite apparent that American Motors has an engine that is sufficiently flexible to be readily adapted to future displacement requirements, higher compression ratios, and any of the other forward-reaching design advancements of the automotive industry today. It is also apparent that the crankshaft forging is designed to give 100% balance for much heavier reciprocating and rotating weights than are now carried, and that the water and oil pumps have a greater capacity than presently required. We have also noted with equal interest that the main bearing caps and bolts, as well as the connecting rods, have likewise been designed for greater load-carrying capacity and more successful bearing performance.

In Fig. A you can see that the bearing loads for the entire speed range are well within the capacity of the microbabbitt bearing to perform satisfactorily. As bearing loads increase in the future, it will be quite possible to utilize other bearing materials, namely, the intermediate duty type. And then, if loads should climb still higher, they can be easily accommodated on the trimetal-type bearing which, of course, is the highest duty bearing on the market today.

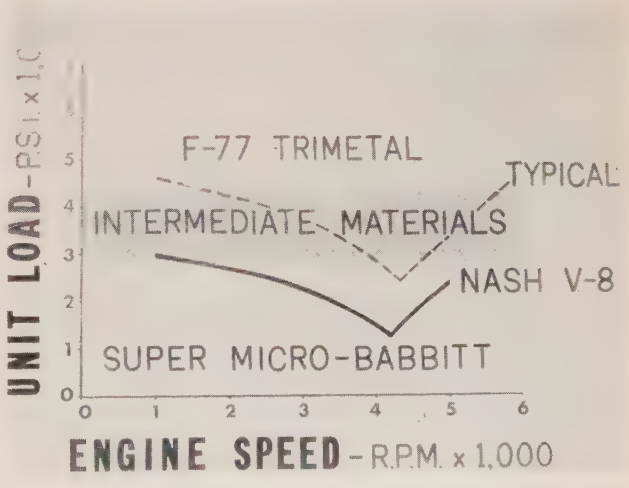
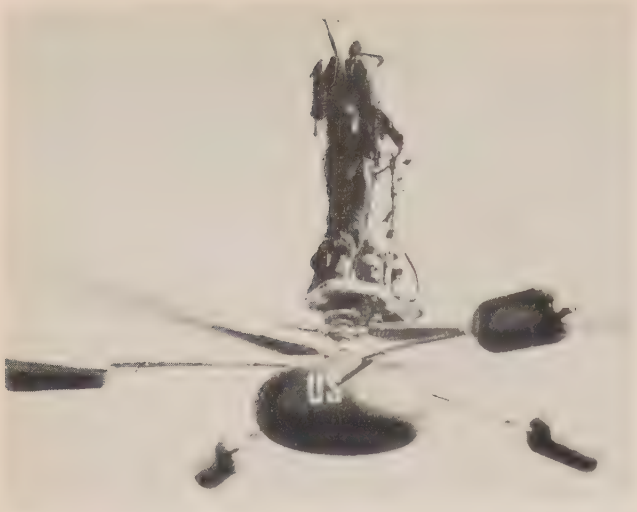


Fig. A—Maximum rod bearing loads



Courtesy, U. S. Army

Fig. 1—DeLackner "Flying Platform" aerocycle

Convertiplanes

Table 1—Vertical Lift Sources and Applicable Aircraft Configurations

Vertical Lift Source	Aircraft Configuration
Air displacement	Balloons and dirigibles
Rotary wing	Helicopters and autogyros (in certain cases) Unloaded rotor-type convertiplanes Tilting rotor convertiplanes
Propeller	Tail-sitter VTOL aircraft Tilting-wing VTOL aircraft Slipstream deflection VTOL aircraft
Ducted propeller	Tilting engine VTOL aircraft Deflected exhaust VTOL aircraft
Turbojet	Tail-sitter VTOL aircraft Tilting engine VTOL aircraft Deflected exhaust VTOL aircraft
Rockets	Missiles—rocket ships

REVIEW is given of some representative current configurations and propulsion arrangements for VTOL aircraft. Support is urged for a comprehensive development program on VTOL aircraft to cover the spectrum of aircraft flight speeds and promising propulsion arrangements.

Special effort and support is needed on ducted-propeller and ducted-fan types, the author says, as these show considerable promise for commercial and military transport use. Development of this type is dependent upon the availability of suitable powerplants.

Efforts to reduce the landing and take-off distance requirements of conventional fixed-wing airplanes should continue, he feels, but it should not be accepted as an interim step toward or a substitute for true VTOL aircraft development.

VTOL when used to identify or describe an aircraft type refers to its capability for making vertical take-offs and landings. Helicopters are currently the most highly developed VTOL aircraft configuration and the only one now in general use. Another VTOL aircraft configuration is the convertiplane. The term convertiplane is usually assumed to mean an aircraft which changes or converts, and hence obtains the lift needed for flight during flight. The most commonly known convertiplane is one that takes off as a helicopter, converts to a propeller-driven, fixed-wing airplane for translational flight, and reconverts to a helicopter for landing. Other types of VTOL aircraft such as those using turbojet, ducted-propeller, or ducted-fan propulsion either divert thrust downward or rotate the propulsion package to achieve direct lift for take-off and landing and thrust for translational flight.

Since the helicopter is currently the VTOL aircraft type in most general use, and in many cases is the basis and background from which other VTOL aircraft concepts develop, it seems proper to start with helicopters.

We have been active in helicopter research, design, and development since 1940. We have designed and built 12 different models of helicopters and currently are producing eight different models for delivery to the United States Armed Services, for foreign governments, and for domestic and foreign commercial sales. Although helicopter concepts and experiments extend as far back in history as Leonardo da Vinci, successful helicopters are not very old. The first really successful United States helicopter was the Sikorsky VS-300 experimental machine designed, built, and demonstrated by Dr. Igor Sikorsky in September, 1939. We followed closely with an experimental heli-

and Other VTOL Aircraft

Robert J. Woods, Bell Aircraft Corp.

This paper was presented at SAE National Aeronautic Meeting, New York, April 11, 1956.

copter flown in December, 1942, which developed into the first helicopter to obtain a CAA Certificate, NC-1 H in 1946. The interest and enthusiasm for "whirlybirds" has spread rapidly since 1940. In the last 10 years there have been 45 helicopter producers in the United States who have produced at least 114 different models. Helicopters have been built in a wide range of sizes and for a wide range of uses. New concepts for their use continue to develop.

Useful and versatile as the helicopter is, it is also recognized that it has limitations. Modern helicopters have high vertical lift efficiency and there is a potential for as much as 50% improvement in this area in the next few years. But they also have a high-speed limitation which is difficult to exceed because it is inherent in the type. It is this high-speed limitation of helicopters that generates the thinking and desires that lead to other VTOL aircraft concepts. There has been interest for some time in the possibilities of combining the vertical-lift and hovering capabilities of the helicopter with the high-speed potential of the fixed-wing airplane which is limited only by available thrust.

Considerable study and research have been applied to convertiplane concepts by many agencies. We have been interested in the convertiplane since early in its helicopter development program. The initial studies first led to a tilting-rotor convertiplane configuration, and such an aircraft, the XV-3, has been designed and built and is now undergoing flight tests. In 1950, our studies were initiated on ducted-fan and turbojet configurations for a VTOL aircraft where the jet-propulsion thrust was controlled and directed at a varying angle between the vertical and horizontal. In 1953, a Bell-supported experimental VTOL air-test vehicle with tilting turbojet engines was built and flown. This aircraft has now completed its prescribed flight test program. Another experimental turbojet VTOL aircraft is now under construction employing controlled exhaust deflection for direct lift and propulsion. It is expected that this aircraft will fly this year.

Our studies have continued on a VTOL aircraft type using tilting-ducted propellers and indicate that this may be one of the most useful VTOL configurations because of its apparent suitability for medium-range, medium-speed commercial and military transport use.

The four VTOL configurations mentioned above generally represent our efforts in this area so far. Numerous other configurations and propulsion arrangements have been studied, tested, and, in some cases, developed both in the United States and abroad. No attempt will be made to identify or describe them all here, but a series of representative configurations and propulsion arrangements will be shown in an effort to display the general character of the VTOL types.

Table 1 lists VTOL aircraft vertical-lift sources and applicable aircraft configurations. While air displacement and rocket vertical-lift arrangements are in current use, the aircraft configurations using them go beyond the scope of this paper, and the following data and discussion will be limited to VTOL types using the remaining four vertical-lift sources.

Some representative VTOL aircraft configurations, starting with a variety of sizes of helicopters covering the range from about the smallest to the largest yet produced in the United States are shown in the following illustrations.

The helicopter shown in Fig. 1 is the deLackner "Flying Platform" aerocycle, one of the smallest and simplest developed so far. It is a stand-on platform

Fig. 2—Latest, licensed model of Bell helicopters, the 47H-1

Fig. 3—Sikorsky S-56 helicopter

Fig. 4—Piasecki YH-16 helicopter

Courtesy, USAF

Fig. 5—Hughes XH-17 "Flying Crane" helicopter

type of helicopter. The stability and control is provided by the pilot's body maneuvers. The design is based on the research and experimental test work of P. R. Hill and C. H. Zimmerman at the Wallop's Island Station of the NACA Langley Laboratory. It can operate from land or water and was first flown in January, 1955. Additional units are understood to be under construction for the U.S. Army. The powerplant is a converted 44-bhp outboard motor. The rotor diameter is 15 ft, and twin 2-blade coaxial rotors are used. It can carry a payload of 300 lb. The gross weight is reported to be 500 lb; the range about 150 miles; and the ceiling approximately 5000 ft.

Fig. 2 shows the Bell model 47H-1 helicopter. It is a 3-place machine using a Franklin 200-bhp piston engine. A single 2-blade rotor 35 ft in diameter is used. The gross weight is 2350 lb. The maximum speed is 100 mph, the cruising speed 90 mph. The service ceiling is approximately 13,000 ft. The range, with 35 gal of fuel, is 207 miles; the endurance 3.3 hr.

Fig. 3 shows the Sikorsky S-56, 26-place helicopter. It uses two Pratt & Whitney R-2800 engines having a total of 3800 bhp. It has one 5-blade rotor, 68 ft in diameter, and is the only 5-blade rotor helicopter built. The gross weight is 28,500 lb. The cruising speed is 120 mph, and the range with normal fuel load is 200 miles. The S-56 is the largest single-rotor helicopter so far produced in the United States. It has not yet been placed in commercial service, but similar aircraft are under construction for the United States Armed Services.

The Piasecki model YH-16 helicopter is shown in Fig. 4. It is the largest passenger helicopter built to date. It was originally powered with two Pratt & Whitney R-2180 piston engines of a total of 3300 bhp. These engines are being replaced with Allison T-56 gas turbines. Another model of this helicopter, the YH-16A was powered with Allison T-38 gas turbines. The commercial version has 43 seats for three crew members and 40 passengers. The YH-16 was built and first flown in 1953. It employs two 3-blade tandem rotors, 82 ft in diameter. The gross weight is approximately 32,000 lb. The high speed at sea level is reported to be 150 mph. Service ceiling is over 18,000 ft. The overall length with rotors turning is 134 ft, and the overall height is 31 ft.

These helicopters are configurations where the rotor is driven by mechanical transmissions and driveshafts from piston or turbine engines. Another helicopter rotor-drive arrangement uses tangentially aligned jets at the rotor tips. In this case the jet propulsion applied at the rotor tips supplies the power needed to turn the rotor at operating speeds.

Fig. 5 shows a rotor-tip propulsion helicopter, the Hughes XH-17 "Flying Crane" developed for the USAF. The empty weight is 28,500 lb; the gross weight for take-off is 45,000 lb. The 2-blade rotor diameter is 130 ft. The height to the top of the rotor mast is 30 ft. This helicopter is intended to carry its

payload in a detachable pack between the stilt-like landing gear legs. Various loads have been carried, including a truck semitrailer for demonstration and test purposes. The Hughes helicopter generates compressed air in two modified Allison J-35 turbojet engines through ducts to the rotor-blade tips where fuel is mixed with the airflow in four burners near the tip of each blade to drive the rotor.

Other experimental tip-propelled helicopters have been built and flown using rocket motors, ram-jets, pulse-jet motors, and compressed air jets. The rotor-tip propulsion arrangement has been thought to permit the development of chaper and simpler helicopters through elimination of the mechanical transmission and driveshaft mechanism. Some disadvantages are that the tip-propulsion power units have higher fuel consumption rates than piston or turbine engines with mechanical drives which tend to critically restrict the helicopter flight range and endurance. Generally they also have objectionably high exhaust noise levels. No tip-propulsion helicopters have been developed beyond the experimental stage so far.

Still another interesting form of helicopters is shown in Fig. 6. This is the Hiller "Flying Platform." It was developed for the Navy Department, reportedly for the Marines. Flight stability and directional control is obtained by means of the pilot's body maneuvers. The design is also based on the research and experimental test work of P. R. Hill and C. H. Zimmerman of the NACA Langley Laboratory. The vertical lift is supplied by two coaxial 3-blade rotors 6 ft in diameter mounted in a short tunnel or duct beneath the pilot's feet. The duct improves the static thrust efficiency and makes the relatively small diameter, compact rotor arrangement practical. This is an example of the ducted-propeller or ducted-fan propulsion arrangement, but since it is fixed in the vertical thrust position, the aircraft is generally considered a helicopter configuration.

The next two aircraft are experimental rotor-type convertiplanes developed by the USAF for the U. S. Army. The first shown in Fig. 7 is a tilting rotor type, the Bell XV-3 4-place convertiplane. The two 3-blade rotors are shaft driven from a Pratt & Whitney R-985 450-bhp piston engine located in the fuselage. The rotor diameter is 25 ft. The rotors tilt through a 90-deg angle from the horizontal position for helicopter function to a vertical position for propeller function. The rotor rotational speed in the propeller position is reduced considerably below the rotational speed in helicopter position in the interest of improved function in both positions. A gear ratio change in the drive mechanism is provided. The XV-3 has converted in ground tests, but has not yet done so in flight. It has flown as a helicopter, and conversion flight tests are under way.

An unloaded rotor-type convertiplane is shown in Fig. 8. This is the McDonnell XV-1 4-place convertiplane. It incorporates both a helicopter rotor and a pusher propeller installation. It has a Continental R-975 piston engine of 550 bhp, driving both the 3-blade helicopter rotor and the pusher propeller alternately or in combination. During transitional flight the helicopter rotor blade incidence angles are

Fig. 6—Hiller "Flying Platform"

Fig. 7—Tilting rotor convertiplane, the Bell XV-3

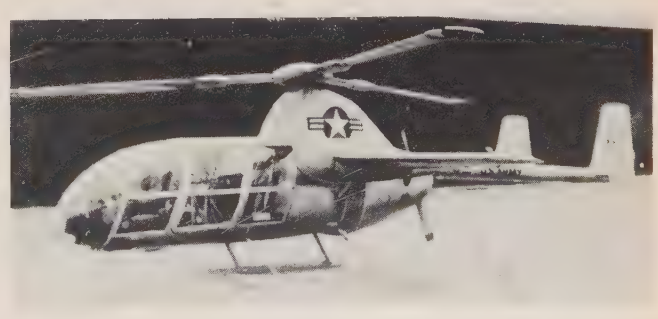
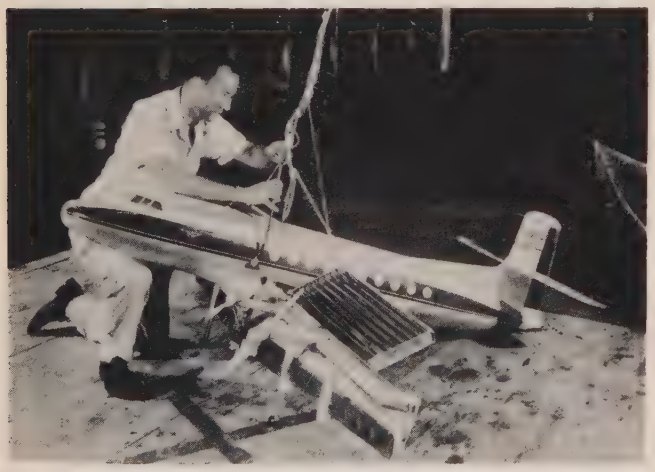


Fig. 8—Unloaded rotor convertiplane, the McDonnell XV-1

Fig. 9—Convair and Lockheed Navy tail sitters

Courtesy, NACA

Fig. 10—NACA test model of tilting-wing VTOL transport



Courtesy, NACA

Fig. 11—NACA deflected slipstream VTOL transport test model

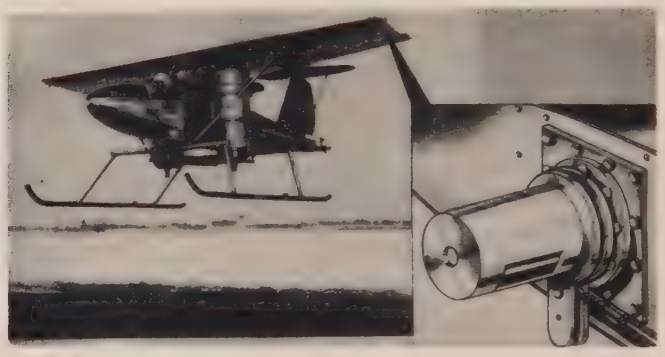


Fig. 12—Bell VTOL air-test vehicle, designed and built to test tilting jet-engine configuration and to serve as a test for jet-reaction hovering and slow-speed flight control system. Inset shows reaction control

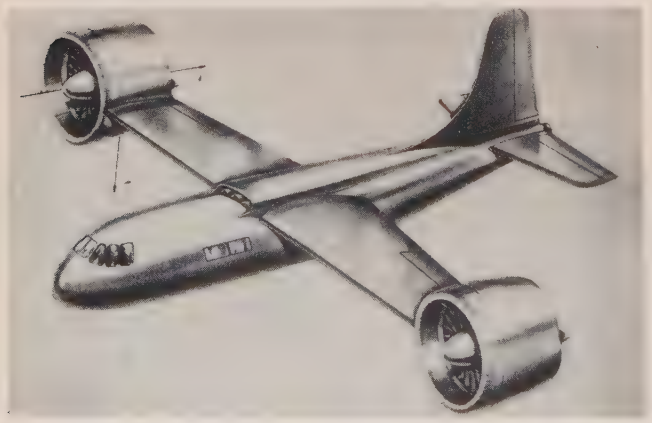


Fig. 13—Bell VTOL ducted-propeller transport

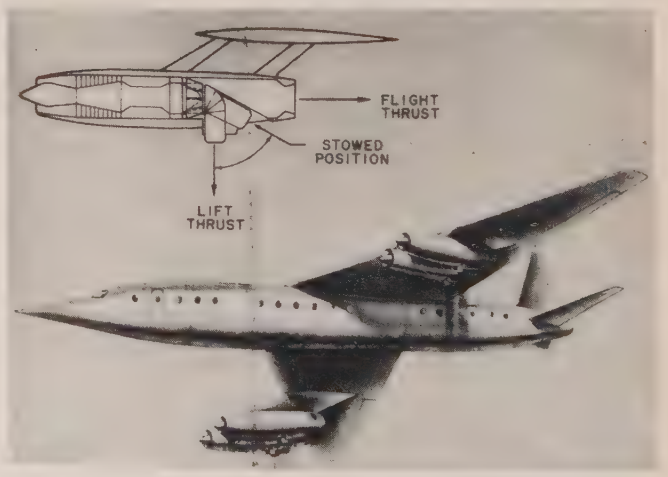


Fig. 14—Design study of turbojet-powered high-speed transport

reduced as lift is supplied by the wings until the rotor blades are feathered and the lift required for flight is supplied largely by the wings. The XV-1 has converted in the air from helicopter to airplane flight in May, 1955. The maximum speed is reported to be 180 mph.

Another VTOL configuration is the tail-sitter type. In this case the aircraft is basically a normal fixed-wing airplane with the propulsion thrust aligned along the longitudinal flight axis but with sufficient available thrust and control devices to permit controlled vertical take-off and landing from a tail-sitting position. The two tail-sitter VTOL aircraft shown in Fig. 9 are experimental propeller-driven fighters developed for the USN by Convair and Lockheed. The dual opposite-rotation propellers driven by a gas turbine engine provide the thrust for vertical take-off and hovering as well as transitional flight. Hovering control is affected by using the propeller slipstream and the aircraft tail in combination with pendulum function. Both aircraft have been flown. The Convair aircraft has successfully taken off and landed in the tail-sitter attitude. The Lockheed aircraft is reported to have made conversions at altitude but not to have made any tail-sitter take-offs or landings.

An experimental turbojet-powered tail-sitter VTOL aircraft is also under development for the USAF by Ryan. This project was originally supported by the Navy Department and a captive test vehicle was built and flown several years ago. The Air Force has since supported the construction of an experimental aircraft which is now undergoing flight test. Security restrictions prevent showing the Ryan jet VTOL aircraft or giving any data on it, except to say that it is a delta-wing configuration and is powered with a British Rolls-Royce Avon engine.

The tail-sitter configuration takes advantage of the possibility of using essentially standard engines and powerplant installations in a fixed environment relative to the airframe. The advantages of this are exchanged for an unconventional ground attitude. The unconventional ground attitude may be acceptable for some types of aircraft, but it has been slow to gain enthusiastic support particularly for the backing-down to a tail-sitting landing operation. It does not appear attractive for multiplace or cargo types. We believe that the horizontal ground attitude is the correct configuration for VTOL aircraft of the types under consideration in this paper.

Considerable work has been done by the NACA Langley Laboratory during the past 10 years on propeller-driven VTOL aircraft suitable for transport use in two configurations.

The first is shown on Fig. 10, where the entire wing including the engines and propellers is tilted or rotated to an upward-pointed position to produce direct lift for VTOL function. The system rotates to the horizontal position for normal forward flight. The scientists of the NACA Langley Aeronautical Laboratory have been studying the problem of stability and control with this one-tenth scale model which is operated in free flight in the laboratory under remote control.

A second NACA model is shown on Fig. 11 where the wing-leading edge, engines, and propellers remain in a conventional fixed-wing aircraft position and the wing rear-section disjoints and deflects downward to form a series of large vanes, slots, and flaps which effectively deflect the propeller slipstream downward through a 90-deg angle to provide lift for low speed or hovering flight. This model is also used for the study of stability and control in hovering, vertical take-off and landing, and transition to forward flight. Both models shown have demonstrated successful VTOL function in laboratory tests.

While the NACA models show arrangements that would appear difficult to accomplish structurally in a full-scale aircraft, the theory and principles demonstrated are thought possible of practical application. No full-scale aircraft of either of these two types is known to be under construction.

Another approach to the VTOL configuration is the rotating jet-engine or deflected jet-thrust arrangement where the propulsion thrust is directed vertically downward for VTOL function and directed aft in a normal manner for transitional flight. Representative of this configuration is the VTOL air-test vehicle shown on Fig. 12. This aircraft was designed and built to test the tilting jet-engine configuration and to serve as a test bed for a jet-reaction type hovering and slow-speed flight control system. It has a gross weight of 1970 lb. The wing span is 21 ft, and the length is approximately 20 ft. It has two Fairchild J-44 turbojet engines for direct lift and propulsion and one Continental Palouste turbine air compressor for the jet-reaction control system. In the interest of quick and inexpensive construction, a Schweizer glider fuselage and a Cessna 170 airplane wing were purchased and modified for this aircraft. This VTOL air-test vehicle has now completed its prescribed test program. Satisfactory VTOL operation was obtained in all areas investigated. The tests of the jet-reaction control system in hovering and slow-speed flight verified a theoretical analysis that direct manual control was entirely practical and satisfactory.

The aircraft shown in Fig. 13 is a VTOL design study employing two ducted-propeller installations mounted on the wing tips so as to tilt from horizontal to 10 deg ahead of vertical. Power is supplied to the ducted-propeller installations by a mechanical transmission from turbine engines located in the fuselage. The design shown has a payload of 4000 lb. The calculated performance shows a maximum speed capability of 430 mph. A sample flight plan would be a range of 800 miles at 350 mph at 30,000-ft altitude with fuel for 16-min hovering time and VTOL capabilities. Using a 400-ft ground run and adding 3500 lb of overload fuel would increase the range, with 16-min hovering time, to 1100 miles.

Fig. 14 shows another Bell design study of a turbo-

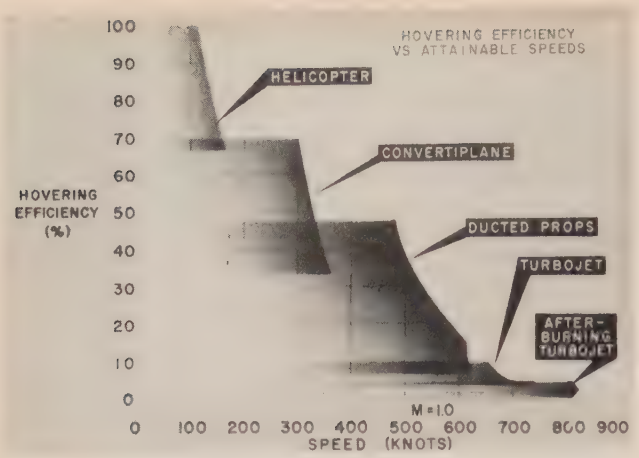


Fig. 15—Estimated comparative spectrum of VTOL aircraft high-speed performance in terms of hovering efficiency

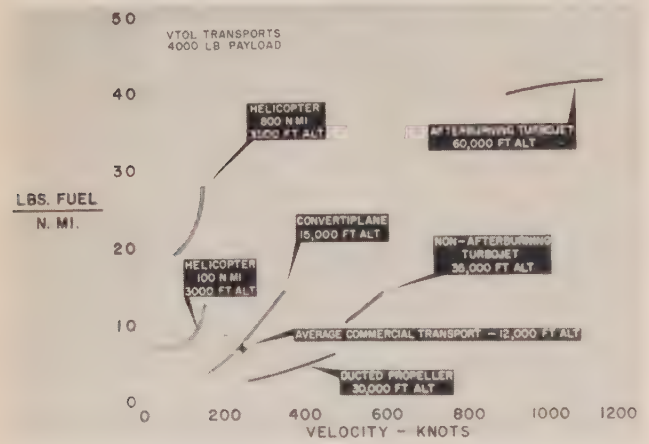


Fig. 16—Generalized comparative data based on design studies of variation of fuel consumption with speed

jet-powered high-speed transport type. The configuration shown is suitable for supersonic flight speeds up to Mach = 2.0 with afterburners installed. This design also has a payload of 4000 lb. To provide the best possible conditions for combined VTOL and supersonic flight speed function, the engines are fixed in position relative to the airframe and a dual-exhaust jet arrangement is used. An internal valve directs the exhaust from the turbine wheel into a pivoted auxiliary jet exhaust for VTOL function. The auxiliary jet exhaust is vertical and functions as an afterburner at take-off. After take-off it is tilted aft to provide transitional speed adequate for aerodynamic flight. The valve is then operated to direct the turbine exhaust to a conventional afterburner and tail pipe, and the auxiliary jet exhaust is stowed. The operation is reversed for landing. In landing, the normal turbojet-engine thrust is adequate to provide direct lift for hovering because of the reduction in fuel load, and the afterburner operation is not used, providing better hovering thrust control and considerably improved fuel economy. The calculated performance for this design study shows a range of 800 miles at Mach = 2.0 with VTOL function and a terminal hovering time allowance of 6 min. If a 1500-ft ground run at take-off is used and 30,000 lb of overload fuel carried, the range is 1300 miles at Mach = 2.0 with vertical landing function and 6-min terminal-hovering-time allowance.

These aircraft have shown a rather wide divergence in configuration and arrangement, and this comes from efforts of the designers to use the various types of available lift- and thrust-producing equipment and to provide for various maximum flight speeds and other operational characteristics. While not all known configurations and arrangements have been illustrated, it is thought that a representative selection has been given. The size and weight of the aircraft has, of course, a controlling influence on its configuration not only directly as size and weight factors influence the airframe design, but also indirectly as these factors affect the availability of suitable powerplants and related equipment.

Depending on the design approach, powerplant problem in the VTOL types can be complicated due to usual specialized nature of the installations and extensive auxiliary devices and mechanisms required. Helicopters have suffered from this trouble. Engine manufacturers accustomed to selling an approved type tested engine as a more or less complete package are usually not interested in building a wide variety of transmissions, attachments, and auxiliaries the VTOL aircraft usually needs, so the aircraft constructor builds his own. Often the aircraft constructor has to learn to design and build suitable power transmission equipment from scratch, and a great deal of the development troubles are often associated with this equipment. This has been the problem in helicopter development, and it is anticipated that it will be a similar problem but of somewhat less magnitude in all VTOL aircraft development.

Perhaps the next thing to look at, having noted some examples of efforts to design and develop VTOL aircraft, is how some of the functional characteristics vary as the cross-breeding takes place between the helicopter and the conventional fixed-wing airplane to produce the VTOL type. The sad story is that you cannot expect to get something for nothing, in spite of the enthusiastic sales appeals of some of the more ardent VTOL disciples. In the light of present know-how, the best VTOL compromise is to give up something you do not consider important or critical for a particular mission in exchange for something you do want or need.

An example is shown in Fig. 15. This chart is based on generalized data prepared in connection with our studies of VTOL convertiplane types. It is intended to show the estimated comparative spectrum of VTOL aircraft high-speed performance in terms of hovering efficiency. In this case, hovering efficiency has been designated as a parameter roughly equivalent to thrust divided by equivalent shaft bhp. Limit maximum speeds have been set at the apparent prac-

tical high-speed limits of a series of design studies made on the VTOL types indicated.

The best conventional helicopter performance available today has been established as 100% hovering efficiency with possible variation down to 66%. At maximum hovering efficiency values, the flight speed appears limited to approximately 110 knots, while if the helicopter is compromised for speed at 66% hovering efficiency the high-speed limit could be approximately 160 knots.

The tilting-rotor type convertiplane, with compromised helicopter design parameters in the interest of improved propulsive efficiency when the rotors are used as propellers appears to have a maximum hovering efficiency potential of approximately 70% which could drop as low as 34% and a practical maximum flight speed limitation of about 300 to 350 knots for the hovering efficiency range indicated.

If higher translational flight speed and a compromise more toward the conventional fixed-wing airplane is desired, current studies indicate that a good tilting type ducted-propeller arrangement could offer comparative maximum hovering efficiencies of 48% and level flight speed limits of about 480 knots. If flight speed limits are pushed up to approximately 610 knots, the comparative hovering efficiency would drop to as low as 10%.

If VTOL function is desired using current nonafterburning turbojet engines, this type of installation would offer flight speeds up to Mach = 0.95 or as limited by available engine thrust, and proper design should permit comparative hovering efficiencies of 10% maximum and could drop as low as 4%.

For the real speed merchant who wants VTOL capabilities and supersonic flight speeds up to Mach = 2.0+ and under present circumstances must use turbojet engines with afterburners, an expected comparative hovering efficiency would be 2 to 4% of that of the best current helicopter.

The relatively low hovering efficiency of the supersonic afterburning turbojet may not mean too much at take-off, as the supersonic VTOL aircraft would not be expected to tarry very long in the high-rent hovering condition as compared to a helicopter, when using the afterburners as would probably be the case for take-off. If the fuel used during flight reduced the aircraft weight sufficiently to permit hovering without using the afterburners the hovering efficiency for landing would be increased to the 8 to 10% range indicated for nonafterburning turbojet operation. This chart does not in itself tell the whole story, but it is thought to be representative of the relative hovering efficiency picture.

Since in any VTOL aircraft type comparative translational flight speed function is also of interest, generalized comparative data based on design studies is shown on Fig. 16.

Fuel consumption per mile flown in level flight is ploted against flight speed for a speed range from 75 to 100% of the maximum limit flight speeds shown in Fig. 15 for various VTOL aircraft configurations and propulsion arrangements. It should be pointed out that these are generalized curves and that in each case it is assumed the aircraft design varies with speed over the ranges indicated.

Fig. 17 shows the general characteristics of the aircraft gross weight variation as the several configuration studies are designed for increased flight speeds while maintaining a constant payload and range. In this, the rapid weight growth of both the helicopter and rotor convertiplane tend to indicate high-speed limitations.

Fig. 18 displays the generalized characteristics of another performance parameter. In this figure the calculated variation of the design range at constant payload is shown as influenced by flight speed variation from the basic design speed.

To give some indication of the general effects of the data given on the overall aircraft, Table 2 compares several types of VTOL aircraft. A conventional fixed-wing transport representative of six current United States commercial transports such as the DC-4 is also tabulated. In each case the aircraft payload is 4000 lb and the range in all cases except one is

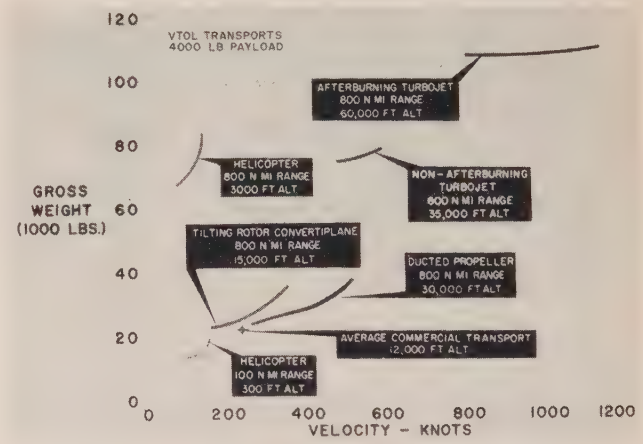


Fig. 17—Variation of gross weight with speed

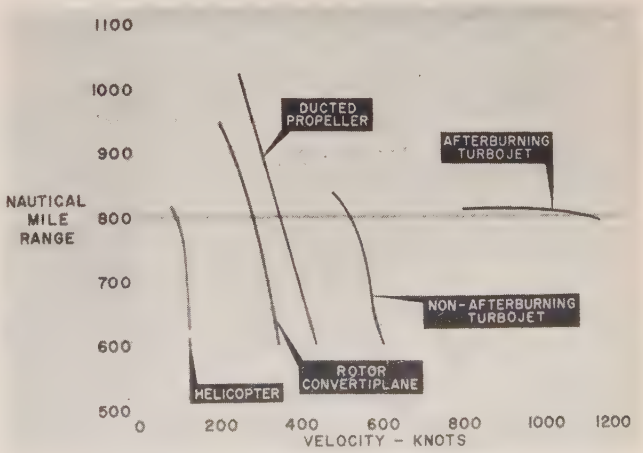


Fig. 18—Variation of range at constant payload with speed

Table 2—Comparison of VTOL Designs

	Helicopters		Rotor Convertiplanes	Ducted Propellers	Turbojet	Afterburning Turbojet	Standard Commercial Transport
Range, nautical miles	100	800	800	800	800	800	800
Cruise Velocity, knots	100	100	250	350	520	1150	240
Cruise Altitude, ft	3000	3000	15,000	30,000	35,000	60,000	12,000
Terminal Hovering Time, min	18	18	16	12	7	6	
Fixed Equipment, lb	1900	2870	2870	2870	2870	2870	2870
Propulsion, lb	4490	15,030	6840	5700	15,000	23,000	3300
Structure, lb	2533	9600	5180	8770	16,190	27,890	4870
Empty Weight, lb	8920	27,500	14,890	17,340	34,060	53,960	11,040
Payload, lb	4000	4000	4000	4000	4000	4000	4000
Crew, number	1	2	2	2	2	2	2
weight	200	400	400	400	440	440	400
Fuel, lb	1050	17,200	5700	4760	17,500	53,800	6680
Take-off Gross Weight, lb	14,170	49,100	24,990	26,500	56,000	112,000	22,120

800 miles. Because an 800-mile range is improbable for a helicopter, a 100-mile range helicopter is also included.

While no particular brief is offered for the quantitative numbers of the data shown in Figs. 16, 17, and 18 in that the source of these data are a series of design studies covering some rather divergent concepts of use and application, it is believed they properly indicate the comparative picture. It should, however, be apparent that, with such a wide potential range of variations of parameters and design conditions to select from, the problem in making a selection is to be sure that the combination chosen has the compromises and answers desired. There seems to be no substitute for having available adequate, reliable, full-scale flight experience over a wide enough range of operations, configurations, and propulsion arrangements.

Conclusion

It may not be that all the configuration variations of VTOL aircraft covered will be needed as fully developed types. Some "weeding out" and further rationization probably will be required. However, the initial development of a sufficient spread of VTOL aircraft configurations, sizes, and propulsion arrangements is necessary to provide the perspective needed for such selection and rationization.

From the standpoint of practical immediate use, the helicopter is the only VTOL type sufficiently developed. Undoubtedly some of the utility area now staked out for helicopters will pass on to other VTOL types in the future. However, it seems obvious that for jobs with emphasis on direct lift and hovering efficiency, the helicopter will be hard to displace.

Partial incorporation of the deflected slipstream VTOL configuration has produced proposals for what are tentatively called STOL aircraft (short distance take-off and landing aircraft). It is being claimed by some that STOL aircraft can provide all that is needed for a variety of military and commercial flight operations. The STOL types are being compared to VTOL types and the argument offered that the STOL types are less complex, generally lighter, and that the state of the art is more advanced. Those who have had practical experience with VTOL aircraft believe that a direct comparison in state of the art cannot be realistically made since the VTOL capability is a definite concept while the STOL capability is indefinite since it is a matter of the degree of compromise between true VTOL capabilities and conventional fixed-wing airplane function. There is also a growing pile of evidence that is beginning to cast doubts on some of the alleged favorable STOL take-off weight versus payload comparisons and on what would at first appear to be the STOL advantage over VTOL types, in complexity.

With over one-half of the earth's land surface unsuitable for anything but vertical take-offs and landings and much of the other half marginal without extensive preparations, including consideration of the often critical landing approach and take-off clearance problems, there seems no argument against the basic need for VTOL aircraft development. There is a current tendency in some areas to retard the VTOL development work in favor of support of STOL applications, the argument being that the STOL type should be an interim development to the ultimately needed VTOL types. This is thought not to be a realistic concept and can be a serious road block to the needed basic VTOL type development. It is recognized that there is need for an aircraft that has basic VTOL capabilities which can also make conventional airplane runway take-offs and landings. In most VTOL types this can be done and the design is usually such that use of a short ground run permits take-off overload and power-off glide or autorotation landings at relatively low speeds.

It is thought the continued improvement of conventional fixed-wing airplanes by application of STOL concepts should continue as it normally would do, but that the STOL concept should not be accepted as an interim step toward or a substitute for true VTOL aircraft development.

References

1. "Comparative Performance of Helicopters, Convertiplanes, and Slow-Flying Airplanes," by P. E. Hovgard. Pub. by Kellett Aircraft Corp.

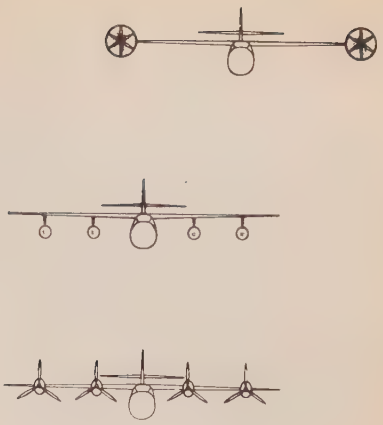
2. "Examination of Some Problems Involved in Design of Propeller-Driven Vertical Take-Off Transport Airplanes," by M. O. McKinney, R. E. Kuhn, and J. B. Hammack. Pub. by NACA.

3. "Application of Ducted Propeller to VTOL Transport Airplane," by J. A. O'Malley, Jr. Pub. by Bell Aircraft Corp.

DISCUSSION . . .

of this paper as well as of the Zimmerman paper, which follows—is to be found on pp. 171–174.

Some General Considerations Concerning VTOL Aircraft



Charles H. Zimmerman, NACA Langley Aeronautical Laboratory

This paper was presented at SAE National Aeronautic Meeting, New York, April 11, 1956.

THE problem of designing an aircraft capable of vertical take-off and landing, VTOL for short, and also capable of forward flight has been an intriguing one for many years. The first workable solution was the powered balloon, which took two forms, the nonrigid blimp and the rigid dirigible. Operational problems and speed limitations have restricted the practical use of such aircraft. The second workable solution was the helicopter. Many engineers attempted to build helicopters before Igor Sikorsky produced a successful one about 1940. Recent developments in powerplants have made it feasible to build other types of VTOL aircraft.

This paper will not present detailed performance studies or weight analyses with a view to arriving at the optimum configuration of VTOL aircraft for any mission. The NACA does not design aircraft and is not in a position to make such analyses. However, in order that research effort may be properly allocated, it is necessary for the NACA to attempt to foresee the trends of future design and the types of aircraft that will become practicable in the light of foreseeable developments. The advent of the turboprop powerplant with only one-half the weight per horsepower of the reciprocating engine (Fig. 1) made it apparent that it would become feasible to install sufficient power in aircraft to make VTOL a possibility with many types of aircraft. The NACA in 1948 embarked upon a program of study designed to reveal and, if possible, to solve the stability and control problems associated with various VTOL configurations. By 1951 there had been sufficient progress to make it apparent that airplanes of conventional configuration, but equipped with integrated powerplant-wing combinations and designed to permit use of the propeller thrust for vertical lift to support the airplane at zero forward speed, represented a possible VTOL type having very attractive features. A program of research was instituted to

FUNDAMENTAL considerations which govern the power and fuel requirements, maximum speed, safety, and practical utility of types of VTOL aircraft are enumerated and discussed. The most interesting types of VTOL aircraft such as the helicopter are discussed and evaluated in regard to their probable applicability to the various fields of usefulness in view of these fundamental considerations.

The question whether vertical take-offs and landings are really necessary is considered. It is recognized that there is an advantage in the power-controlled approaches of VTOL aircraft from small airports under all weather conditions.

It is realized that there is research still needed to make the VTOL a more satisfactory aircraft.

study the stability and control characteristics and problems of such airplanes. A parallel research program was instituted to determine the performance of propeller-wing configurations in the then no man's land between hovering flight with the weight entirely supported by the propeller thrust and forward flight as an airplane with the weight supported by the wings. This research has been reported by the NACA [1-10] and used as a basis for papers.[11,12]

This paper will not attempt to summarize the NACA research but will consist rather of a discussion of general considerations concerning VTOL aircraft, stemming from the background of experience gained by the author in connection with NACA research in this field and from experience with development of a VTOL aircraft proposed in 1934.[13,14]

It would be impracticable to attempt to describe all the many types of VTOL aircraft which have been proposed, or even all of those which offer reasonable promise of being capable of flight. Instead, the fundamental factors which govern the power and fuel requirements, performance, safety, and practical utility of VTOL aircraft will be discussed. The various fields of usefulness, both civilian and military, of VTOL aircraft will be enumerated. The most interesting types in view of the implications of the fundamental factors will be discussed in relation to their fields of usefulness, and the areas in which further research and development are most urgently needed will be indicated.

Fundamental Factors Affecting Performance and Practicability

In order to understand the limitations and select the fields of usefulness of the many possible configurations capable of both vertical and forward flight, it is necessary to consider and study certain fundamental factors. The greatest deterrent at the present time to more widespread use of vertical flight aircraft is cost, either real or feared. The really important factor is the ratio of the cost to the capacity for doing useful work. All that can be done here is to touch upon a few pertinent points and suggest possible fruitful lines of inquiry.

Power and Fuel Required—A primary factor in the design of an aircraft capable of vertical flight is the power required in hovering. This can be most conveniently thought of as horsepower per pound of thrust. The horsepower per pound of thrust can be converted to pounds of fuel per pound of thrust per hour if the calorific value of the fuel and the efficiency of its conversion into power is known. When considering the original cost of the aircraft, maintenance, and similar cost items it is convenient to deal with horsepower per pound of thrust; when considering fuel costs, endurance, problems of logistics relative to fuel supply, and the like, it is preferable to think of pounds of fuel per pound of thrust per hour.

The power required for hovering varies from zero for a balloon to a very large value for a rocket-supported vehicle. The upward thrust necessary to support a heavier-than-air vehicle is equal to the product of the mass of air accelerated downward per unit time and the velocity [15] imparted to it. It can be shown very readily that the power required per pound of thrust varies linearly with the velocity imparted to the mass of air accelerated downward. The necessary power, that is, the kinetic energy per unit time, is equal to one-half the mass of air per unit time multiplied by the square of the velocity imparted to it.

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1 NACA TN 3198 (1954), "Dynamic Stability and Control Characteristics of Cascade-Wing Vertically Rising Airplane Model in Take-Offs, Landings, and Hovering Flight," by M. O. McKinney, Jr., L. P. Tosti, and E. E. Davenport.

2 NACA TN 3304 (1954), "Investigation of Aerodynamic Characteristics of Model Wing-Propeller Combination and of Wing and Propeller Separately at Angles of Attack up to 90 Deg," by J. W. Draper and R. E. Kuhn.

3 NACA TN 3307 (1954), "Investigation of Wing-Propeller Configuration Employing Large-Chord Plain Flaps and Large-Diameter Propellers for Low-Speed Flight and Vertical Take-Off," by R. E. Kuhn and J. W. Draper.

4 NACA TN 3364 (1955), "Investigation of Effectiveness of Large-Chord Slotted Flaps in Deflecting Propeller Slipstreams Downward for Vetrical Take-Off and Low-Speed Flight," by R. E. Kuhn and J. W. Draper.

5 NACA TN 3440 (1955), "Hovering Flight Tests of 4-Engine Transport Vertical Take-Off Airplane Model Utilizing Large Flap and Extensible Vanes for Redirecting Propeller Slipstream," by L. P. Tosti and E. E. Davenport.

6 NACA TN 3630 (1956), "Hovering Flight Tests of Model of Transport Vertical-Take-Off Airplane with Tilting Wing and Propellers," by P. M. Lovell, Jr. and L. P. Parlett.

7 NACA TN 3692 (1956), "Investigation at Zero Forward Speed of Leading-Edge Slat as Longitudinal Control Device for Vertically Rising Airplanes that Utilize Redirected Slipstream Principle," by R. E. Kuhn.

8 NACA TN 3693 (1956), "Preliminary Investigation of Effectiveness of Sliding Flap in Deflecting Propeller Slipstream Downward for Vertical Take-Off," by R. E. Kuhn and K. P. Spreemann.

9 NACA TN 3629 (1956), "Investigation of Effects of Ground Proximity and Propeller Position on Effectiveness of Wing with Large-Chord Slotted Flaps in Redirecting Propeller Slipstream Downward for Vertical Take-Off," by R. E. Kuhn.

10 NACA Proposed TN, "Transition Flight Tests of Model of Transport Vertical Take-Off Airplane with Tilting Wing and Propellers," by P. M. Lovell, Jr.

11 "Some Stability Problems of Vertically Rising Aircraft," by C. H. Zimmerman. Presented at the West Coast Forum of the American Helicopter Society, Los Angeles, Calif., September, 1955.

12 "Examination of Some of the Problems Involved in Design of Propeller-Driven Vertical Take-Off Transport Airplanes." Presented at IAS Annual Meeting, New York, January, 1956.

13 NACA RM L9C29 (1949), "Preliminary Full-Scale Investigation of a 1/3-Scale Model of Convertible-Type Airplane," by R. H. Lange, B. W. Cocke, Jr., and A. J. Proterra.

14 NACA TN 2014 (1950) which supersedes NACA RM L9C24, "Static Longitudinal Stability and Control of Convertible-Type Airplane as Affected by Articulated- and Rigid-Propeller Operation," by R. H. Lange and H. C. McLemore.

15 This assumes a uniform jet velocity. The considerations discussed in this paper apply satisfactorily with normal amounts of nonuniformity in velocity. Products of combustion may also be an important part of the mass accelerated, but they are considered as part of the air mass for simplicity in this discussion. For the aircraft considered in this discussion the mass of the products of combustion can be disregarded.

Hence there is a linear increase in horsepower per pound of thrust with jet velocity. Fig. 2 shows the horsepower per pound of thrust plotted against the jet velocity for the low jet velocities produced by helicopter rotors and airplane propellers. Note that the dividing line between rotors and propellers is drawn rather arbitrarily. In a general way, it can be stated that 0.1 horsepower is required per pound of thrust for a helicopter rotor and 0.2 horsepower is required per pound of thrust for a propeller. (This applies only to propellers designed with static thrust capability as a primary consideration.) Fig. 3 shows the pounds of fuel per pound of thrust per hour required for jet velocities up to 3000 fps for various values of overall efficiency of conversion of fuel heat content into kinetic energy in the jet, assuming the heat content of the fuel to be 20,000 Btu per lb. Representative areas covering roughly the ranges of efficiency and jet velocity for various static thrust generators are also shown. It is immediately apparent that a low-velocity jet, such as that produced by a helicopter rotor or an airplane propeller, is highly desirable for an aircraft performing a mission requiring much hovering. It is interesting to note that although the overall efficiency of a turbojet engine is high, of the order of 30%, the fuel consumption is also high because of the high jet velocity. It seems almost certain any machine required to hover for any appreciable part of its mission will not use the direct thrust of a turbojet engine to do so.

It will be appreciated that the jet velocity is proportional to the jet diameter for a given thrust and a given density of the air in the jet. The jet diameter loading is therefore an indication of the power required per pound of weight or the fuel consumption per pound of weight per hour for a hovering aircraft.

These facts can readily be expressed mathematically. Assuming a uniform velocity in the thrusting jet, the mass of air per unit time is given by:

$$m_j = \rho_j A_j V_j \qquad (1)$$

where:

m_j = Mass of air per unit time, slugs per sec
ρ_j = Density of the air in the jet, slugs per cu ft
A_j = Cross-sectional area of the jet, sq ft
V_j = Air velocity in the jet, fps

The thrust is given by:

$$\begin{aligned} T &= m_j V_j \\ &= \rho_j A_j V_j^2 \\ &= 2 q_j A_j \end{aligned} \qquad (2)$$

where:

T = Thrust, lb
q_j = Dynamic pressure in the jet, lb per sq ft

The kinetic energy in the useful portion of the velocity in the jet, that is, that part of the jet velocity parallel to the jet axis, is:

$$\mathrm{KE}_j = \tfrac{1}{2} m_j V_j^2$$

The brake horsepower to produce the jet is:

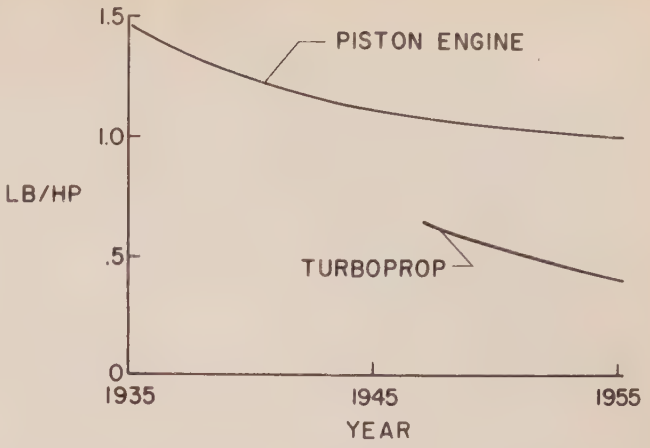


Fig. 1—Trends of minimum specific weight of airplane powerplants

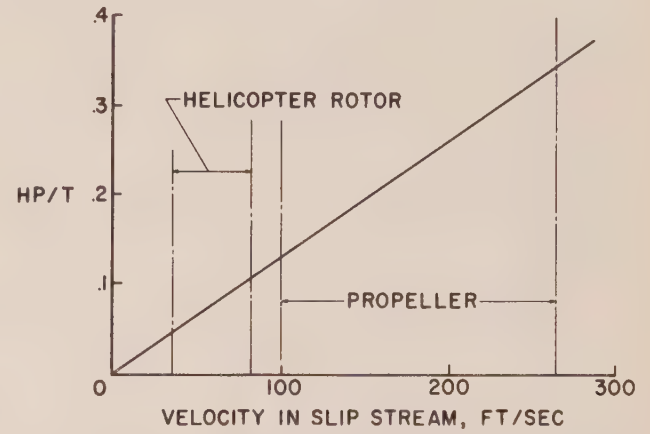


Fig. 2—Relation between horsepower required for hovering and slipstream velocity

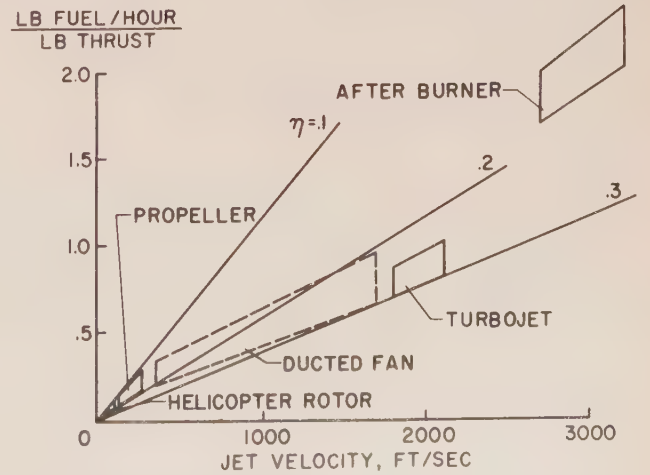


Fig. 3—Relation between specific fuel consumption in hovering and jet velocity

$$\text{hp} = \frac{\text{KE}_j}{\eta_1 550} = \frac{m_j V_j^2}{1100\eta_1}$$

where:

η_1 = Efficiency of conversion of brake horsepower to useful jet kinetic energy

The horsepower per pound of thrust is then:

$$\frac{\text{hp}}{T} = \frac{V_j}{1100\eta_1} \qquad (3)$$

The fuel consumed per hour per pound of thrust is:

$$\frac{W_F/\text{hr}}{T} = \frac{\text{hp} \times \text{sfc}}{T}$$

where:

W_F = Fuel weight, lb

sfc = Specific fuel consumption, lb fuel per hp per hr

If the heat content of the fuel is assumed to be 20,000 Btu per lb:

$$\text{sfc} = \frac{33{,}000 \times 60}{20{,}000 \times 778\eta_2}$$

where:

η_2 = Overall thermal-mechanical efficiency of the powerplant

Therefore:

$$\frac{W_F/\text{hr}}{T} = \frac{V_j}{1100\eta_1} \times \frac{33{,}000 \times 60}{20{,}000 \times 778\eta_2} = \frac{0.000116 V_j}{\eta}$$

where:

η = Overall efficiency of conversion of heat content of fuel into useful jet velocity

The power required per pound of thrust can also be expressed in terms of the air density in the jet and the jet-area loading as follows:

$$\frac{\text{hp}}{T} = \frac{V_j}{1100\eta_1}$$

or from (2):

$$\frac{\text{hp}}{T} = \frac{\sqrt{T/A_j}}{1100\eta_1\sqrt{\rho_j}} \qquad (4)$$

In the case of a propeller or rotor the jet area is, by the momentum theory, one-half the disc area (Fig. 4). Making this substitution:

$$\frac{\text{hp}}{T} = \frac{\sqrt{2T/A_r}}{1100\eta_3\sqrt{\rho_j}} \qquad (5)$$

where:

A_r = Rotor (or propeller) disc area, sq ft

η_3 = Efficiency of conversion of hp applied to the rotor into kinetic energy in the jet

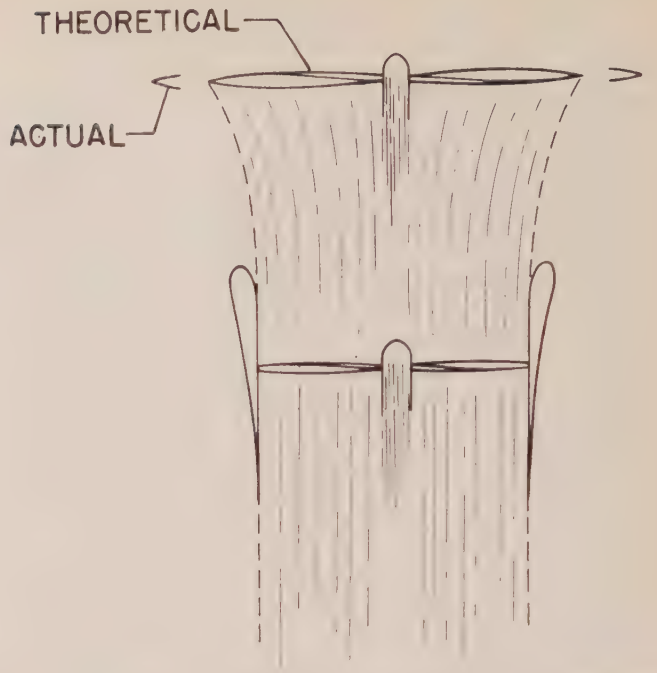


Fig. 4—Comparison of diameters of shrouded and unshrouded rotors required for equal thrust for a given power

When a properly designed shroud is placed around a propeller or rotor the jet area is the same as the shroud exit area. For a simple shroud without expansion in its exit the jet area is the same as the rotor area, as shown in Fig. 4.

It is incorrect to use equation (5) for shrouded propellers. Considerable confusion has arisen because comparisons have been made between unshrouded and shrouded propellers using equation (5) and considering η_3 as a criterion of excellence. This is as misleading as to use overall thermal-mechanical efficiency as a criterion of horsepower or fuel per hour per pound of thrust. Experience indicates that unshrouded, single-rotation rotors or propellers can develop thrust/horsepower ratios corresponding to values of η_3 of the order 0.70 to 0.80 based on the given definition of η_3. Dual-rotation propellers can achieve values of η_3 as high as 0.85. For a dual-rotation shrouded propeller[16] η_1 can approach 1.0. Utilizing these two efficiencies in their proper places shows that to achieve the same thrust per horsepower the ratio of the rotor, or propeller, disc area to the exit area of a propeller shroud is given by:

$$\frac{778\eta_3\sqrt{\rho_j}}{\sqrt{T/A_r}} = \frac{1100\eta_1\sqrt{\rho_j}}{\sqrt{T/A_j}}$$

or:

$$A_r = 2\left(\frac{\eta_1}{\eta_3}\right)^2 A_j$$

[16] NACA RM L7H25 (1948), "Static Tests of Shrouded and Unshrouded Propeller," by R. J. Platt, Jr.

If:

$$\eta_3 = 0.85$$
$$\eta_1 = 0.95$$
$$A_r = 2.5A_j$$

or:

$$D_r = 1.58D_j$$

On this basis it is seen that the diameter of an unshrouded, single-rotation propeller must be of the order of 1.6 times that of the exit area of a propeller shroud in order to support a given weight with the same horsepower. Which of the two is preferable involves such considerations as weight, high-speed performance, stability and control, overall dimensions, and a rational choice can only be made by taking these factors into consideration.

The power required in forward flight is the sum of the power required to overcome induced drag and that required to overcome form and parasite drag. The horsepower per pound of weight is given by:

$$\frac{\text{hp}}{W} = \frac{\dfrac{W}{\pi q S \dfrac{b^2}{S}} V + \dfrac{qSC_{D_o}V}{W}}{\eta_p \times 550}$$

$$= \frac{\dfrac{W/S}{\dfrac{\pi}{2}\rho V A R} + \dfrac{\frac{1}{2}\rho V^3 C_{D_o}}{W/S}}{\eta_p \times 550}$$

where:

hp = Horsepower required
W = Flight weight, lb
q = Dynamic pressure, lb per sq ft
S = Reference lifting area, sq ft
b = Lifting area span, ft
V = Forward velocity, fps
ρ = Mass density of air, slugs per cu ft
AR = Aspect ratio of lifting surface
C_{D_o} = Form and parasite drag coefficient based on reference lifting area
η_p = Propulsive efficiency

It is immediately apparent that at low speeds the horsepower per pound of weight (and hence the fuel consumption per pound of weight per hour for a given overall efficiency) is primarily dependent on the wing loading and the aspect ratio (assuming sea-level air density). At high speeds the primary factors are drag coefficient and wing loading. It is also immediately apparent that a high aspect ratio and a low drag coefficient are always desirable. Whether the lifting area loading should be high or low depends upon the speed at which economy is desirable.

Fig. 5 shows typical variations of horsepower required versus forward flight speed for a helicopter and for a VTOL airplane having the same gross weight and substantially the same span loadings. The relatively low hovering power required by the helicopter is, of course, the result of its low jet-area loading. The substantially equal span loadings are reflected in the substantially equal power requirements at minimum power. The high parasite drag (including rotor drag) of the helicopter is indicated by the rapid rise in power with speed above the speed for minimum power. From a safety standpoint, the rapid decrease in power required with speed and the low power required at the speed for minimum power are very desirable characteristics of the VTOL airplane. Note that in the event of loss of one-half the power when hovering, representing loss of one engine of a twin-engine aircraft, the helicopter must lose enough altitude to accelerate to 60 knots. On the other hand, the high-aspect-ratio aircraft must only lose enough altitude to accelerate to 30 knots. Furthermore, level flight can be maintained with less than one-half the power still available for the high-aspect-ratio aircraft, whereas the remaining engine of the helicopter must operate at nearly full power in order to continue flight.

An important point which should be considered in a VTOL aircraft to be used for moving loads over terrain obstacles under conditions where speed is not of intrinsic importance is illustrated in Fig. 5. Which of these two aircraft will be most suitable from a fuel consumption standpoint will not only depend on their respective ratios of useful load to gross weight but also upon the ratio of time spent in hovering to that spent in forward flight. For the two curves shown the two aircraft will consume equal amounts of fuel for a mission requiring transportation of the load 7 nautical miles with a hovering time of 1 min, assuming the two aircraft have the same ratio of payload to useful load and each cruises at its speed for maximum range.

A factor of great importance in the design of VTOL aircraft is the rapid increase in power required for hovering and slow-speed flight with increase in weight. As can be seen from equation (4), the power required increases as the three-halves power of the weight when hovering. From another point of view, this fact has the practical significance that the power

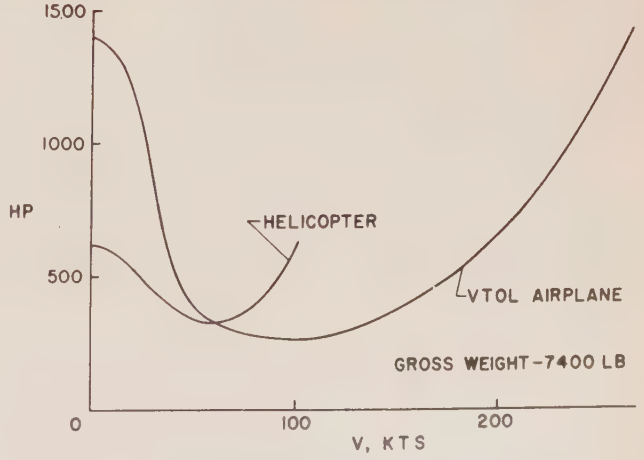


Fig. 5—Typical variations of power required with speed, VTOL aircraft

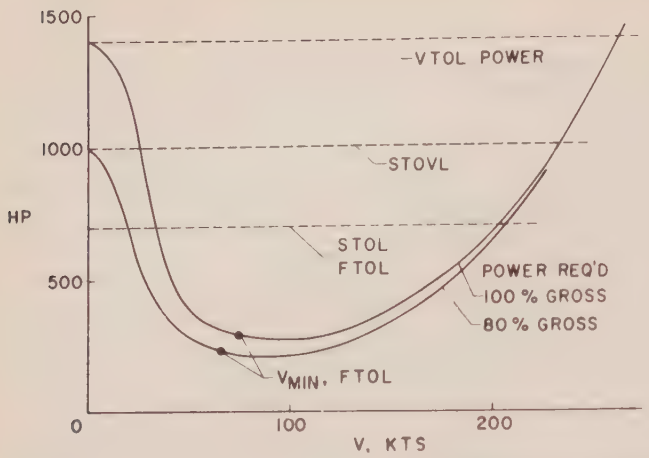


Fig. 6—Effect of power on performance capabilities of airplane

Table 1—Civilian Applications

1. Medium-range transports.
2. Short-range transports.
3. Business transports.
4. Personal vehicles.
5. Miscellaneous:
   a. Rescue.
   b. Police.
   c. Transmission line patrol, and so forth.

required for hovering decreases rapidly with decrease in weight. For example, Fig. 6 shows two power-required curves for a possible VTOL airplane, the upper curve for the aircraft fully loaded, the lower curve for the aircraft after having expended a fuel load of 20% of the gross weight. The power required for a minimum flight speed of 26 knots at sea level when fully loaded is sufficient for hovering after the cruising fuel load has been expended.

Balance, Stability, Control—Not only is it a fundamental fact that the thrust exerted in the vertical direction is equal to the mass per unit time multiplied by the velocity vertically downward imparted to it, but also it is a fundamental fact that the vertical thrust acts along the center of the jet and the center of gravity must be above the center of the jet for balance in hovering flight. As a result, provision must be made for effectively translating or inclining the axis of the jet relative to the aircraft to permit balance over the range of center of gravity positions possible with various conditions of loading. This can be done by translating the jet-thrust unit, or by rotating it if it is sufficiently far above or below the center of gravity. It also can be done by changing the direction of the jet by airfoil surfaces used as deflection vanes, by differential thrust control on two or more thrust units, or by variation of the thrust of auxiliary thrust control units. Provision for an adequate center of gravity travel can be very troublesome in some types of vertical flight aircraft.

The fundamental stability and control requirement for a vertical flight aircraft is that it shall not be so unstable as to be difficult to fly by a normally skillful pilot with the controls provided. Actually, it is desirable that the aircraft be stable but, so far as the author knows, no one has as yet come forward with a practical configuration which is completely stable in hovering flight without use of artificial stabilizing devices. Complete stability is not necessary, however. All helicopters are unstable to a certain degree. The penalties paid for instability are requirement of more pilot training than would otherwise be necessary and inability to fly satisfactorily under blind flying conditions. The detail requirements for stability are outside of the scope of this paper. A general discussion of the problem has already been presented elsewhere.[11]

Accommodations for Crew, Passengers, Cargo—When considering the possibilities of a new and different kind of aircraft, it is easy to forget such factors as comfort, freedom from noise and vibration, range of vision, ease of loading and unloading. Nevertheless, these factors are important, and in the long run no aircraft that does not meet requirements in these respects to a reasonable degree can be expected to be widely used. In many cases there has been a temptation to accept a large attitude change as a means of making possible the utilization of the same thrust means for both static thrust and propulsive thrust. This may be acceptable for certain specialized aircraft but strongly detracts from the general utility of such a configuration.

Ratio Useful to Gross Weight—The importance of achieving a reasonable ratio of useful load to gross weight is so obvious as to need no discussion. This bears so directly and powerfully on the cost and the utility of an aircraft type as to be of overriding importance in many cases. A relatively low value of this ratio tends to be characteristic of vertical flight aircraft and tends to put them at a definite disadvantage which must be counteracted by important advantages in other respects if such aircraft are to come into general use. As an illustration of the importance of this ratio consider the helicopter and VTOL airplane of Fig. 5. When it was assumed they carried equal payloads the break-even range, above which the VTOL airplane would be the more economical from the standpoint of fuel, was 7 nautical miles. If it be assumed that the ratios of useful load to gross weight are 0.35 and 0.25 for the helicopter and the VTOL airplane, respectively (these are believed to be representative figures), the break-even range increases to 23 nautical miles.

Safety and Reliability—A reasonable degree of safety is indispensable. It may be possible to view with equanimity the fact that failure of a powerplant or an electronic device will inevitably result in a fatal crash while an aircraft is in the preliminary stages, but this fact assumes ever-increasing importance as construction progresses. When the time comes for flying the aircraft with human occupants, it becomes of such vast importance that not only do inspection and testing procedures to insure proper functioning neces-

sarily greatly increase the cost and time required but also, and generally more important, the actual operational use of the aircraft is likely to be severely curtailed. Of course, the degree of safety which is required varies with the type of aircraft. Efforts to increase the flight safety of a fighter aircraft might, for example, impair its performance or maneuverability and make it much less capable of coping with enemy fighters and hence much more dangerous than before when its primary mission is considered. On the other hand, a commercial transport aircraft must have a high degree of safety.

Since safety is of such great fundamental importance, it is of interest to consider those characteristics which contribute to safety. First and foremost, it is absolutely essential that the aircraft be capable of continuing flight and of landing on a field of reasonable size after failure of any unit whose continued operation depends on the burning of fuel, the successful operation of hydraulic or pneumatic devices, the successful operation of electric or electronic devices, or the integrity of materials forced to operate near their limit of strength under conditions such that imperfect lubrication, combustion, or other environmental conditions may cause failure. Such devices must be duplicated, or an alternate system provided when their use cannot be avoided.

The statistical advantage of two adequate powerplants over a single powerplant of equal reliability is tremendous. However, a very important point to remember is that this advantage may disappear if continued flight imposes too great a burden on the remaining powerplant. Full realization of the theoretical statistical advantage is possible only when the remaining powerplant is not required to operate beyond its normal cruising power. The type of power-required curve indicated for the VTOL airplant of Fig. 5 is desirable from a safety standpoint.

Also of primary importance is the requirement that failure of any unit of the failable type shall not result in a temporary loss of attitude control or descent rate to such an extent as to cause a serious crash under any flight condition required by the mission of the aircraft or likely to be encountered by the aircraft.

The above discussion assumes that failable units are essential and that even they are reasonably reliable. Obviously, from a safety standpoint it is better to use units of the nonfailable type whenever possible. Also, even the great statistical advantage of duplicate units is meaningless if they are highly failable.

The fundamental consideration of safety has been discussed in some detail because VTOL aircraft are likely to be very unsafe unless special care is taken to insure safety. If such care is taken, VTOL airplanes capable of level flight with a relatively low percentage of their installed power can be as safe or safer than their conventional counterparts.

Complexity—Avoidance of complexity is not absolutely necessary but is fundamentally desirable. Complexity generally increases first cost, maintenance, and the likelihood of failure. The influence of this factor upon the practicability and acceptability of a VTOL aircraft is much greater than may be inferred on the basis of their brief treatment.

Operational Problems—In consideration of vertical flight aircraft from the standpoint of arriving at a decision as to where we shall put our research and development effort, we cannot afford to overlook the effects upon the actual utility of the aircraft of operational problems attendant upon its use. Such considerations as noise, the requirement of special equipment or a special crew for ground handling, jet blast effects, vulnerability to icing conditions, vulnerability to wind when on the ground may make the difference between successful or unsuccessful performance of the intended mission or competition with other types.

Fields of Usefulness

In considering the fields of usefulness for aircraft capable of vertical flight, it is convenient to divide them into two groups, civilian and military. There will be, of course, a certain degree of overlapping since the usages are substantially identical in certain cases and in other cases substantially identical vehicles may be put to rather widely different uses in civilian and military applications. Some applications which will be considered are questionable, but it is believed they should not be ignored in a paper of this type.

Civilian—Possible civilian applications are shown in Table 1. Medium-range transports are transports carrying 20 to 100 passengers over ranges from 50 to 1000 miles. They might also be called feeder-line transports. It is frequently stated that STOL (short, or slow, take-off and landing) airplanes are adequate for their purpose. The reason why medium-range transports are included in this discussion of vertical flight aircraft will be discussed later.

Short-range transports are smaller craft of 8 to 20 passengers, suitable for operation over ranges of 5 to 50 miles. These machines are intended for use between airport and city center or between outlying districts of a metropolitan area and the city center. The need for vertical flight is more apparent than for the medium-range transports.

Business transports are considered as four- to eight-passenger vehicles capable of operating from business establishments having no airport facilities. The need for vertical flight capability is obvious.

Personal vehicles are envisioned as short-range with capacities for one to four persons to be used for personal transportation from place to place to overcome the limitations increasingly being imposed by highway congestion. This is the most questionable of the categories listed, but deserves mention.

The miscellaneous category covers some of the most important and vital uses of vertical flight aircraft. The first four categories cover use of the aircraft as a means of transportation, the fifth covers uses in which the ability to hover over a spot is in itself use-

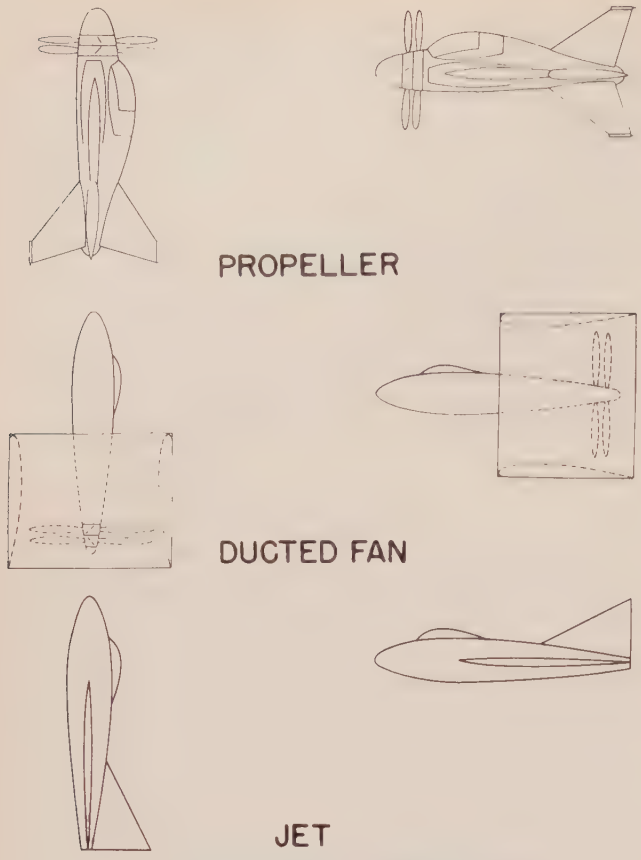


Fig. 7—Typical tail-sitter configurations

Table 2—Military Applications

1. Assault transports.
2. Load lifters.
3. Combat vehicles:
   a. Airborne tanks.
   b. Airborne cavalry.
4. Fighters.
5. Bombers.
6. Supply transports.
7. Utility transports.
8. Rescue vehicles.
9. Ambulances.
10. Reconnaissance and liaison vehicles.
11. Miscellaneous:
    a. Police.
    b. Patrol.

ful. The value of the helicopter in rescue work, for example, has been forcefully and amply demonstrated.

Military—Possible military uses for vertical flight aircraft are listed in Table 2. The value of assault transports capable of vertical landings and take-offs regardless of terrain and indeed capable of taking advantage of bad terrain when desirable seems very great to a layman.

One very important military application for VTOL aircraft is its use as a load lifter to move heavy items over terrain obstacles. The value of such a vehicle is obvious. The movement of tanks and trucks across rivers, the bringing of heavy items ashore from ships without use of small boats, and the placement of artillery in otherwise inaccessible locations are among the applications for which load lifters should be very valuable.

The use of airborne tanks and airborne cavalry has been proposed by representatives of the U. S. Army. Their value has not been proved but seems to this layman to be possibly very great. They at least deserve consideration. One obvious difficulty is the logistic problem of supplying fuel for such operations.

Capability of taking off and landing without forward velocity and hence without runways would seem to add greatly to flexibility in advantageous positioning of fighters and reduction of the vulnerability of fighter bases. The presence in the combat area of fighters capable of maintaining air superiority is almost certainly an essential factor if success is to be achieved in any combat operation.

Provision of vertical take-off and landing ability for bombers has been proposed as a means of avoiding large and hence vulnerable installations.

Granting any or all of categories 1 through 5, it is immediately apparent that vertical flight cargo-supply transports and utility personnel transports become indispensable. The supply transport might possibly be the same machine as the assault transport. The utility transport might be the same or very similar to the business man's transport of Table 1.

The value of vertical flight aircraft for categories 8 through 11 needs no elaboration.

Most Interesting Types and Uses

The portion of this paper which has gone before is largely introductory, an attempt to set the stage for consideration of practicable types of vertical flight aircraft. Not all possible types will be considered. To attempt to consider each of the many types which have been proposed would extend this paper beyond reasonable limits. Rather, the discussion will be limited to those types which appear to offer the greatest promise of successful development or of continued practical use.

Helicopter—We have seen (Fig. 3) that on the basis of fuel required per pound of weight when hovering, the helicopter with its large diameter, lightly loaded rotor is at a distinct advantage, requiring of the order of one-half the fuel per pound of thrust of a propeller-supported aircraft or as little as one-fortieth that of one supported by turbojet thrust. For this reason alone it appears probable that the helicopter is here to stay in connection with those missions in which hovering time is a significant percentage of the total flight time or is the primary function of the aircraft.

Its relatively low fuel consumption when hovering, or its relatively low installed horsepower is, however, not the only advantage of the helicopter as a hovering vehicle. The relatively low jet velocity is advantageous in such operations as rescue operations or load lifting. There is obviously a limit to the jet velocity which can be tolerated in a rescue vehicle which hovers above

the person being rescued. A high jet velocity is a distinct disadvantage for load lifting operations in which it is desirable or necessary for personnel to work in or near the thrusting jet during loading and unloading operations. The relatively low jet velocity and low power requirement is accompanied by relative ease in achieving a low noise level, a very important consideration in operations in heavily populated areas.

It is, of course, possible that a low jet velocity can be achieved with devices other than the articulated rotor. In fact, a simpler and more rugged device than the present rotor is highly desirable. It is believed likely, however, that the improvement will come through improvements to the rotor rather than from development of some entirely different device. As will be pointed out later, it is likely that other types of vertical flight aircraft will be used for those operations where hovering is not the primary mission. The primary reasons for this are indicated by Fig. 5 and were discussed in connection with it. This will shift the emphasis in helicopter development away from attempts to improve the forward speed performance and permit concentration on improvement of the helicopter as a hovering vehicle.

With these foregoing thoughts in mind, it appears certain that helicopters will continue to be used for both civilian and military rescue operations, police and other patrol operations which require remaining over a given spot for appreciable periods of time, and load lifting operations where movements for short distances over terrain obstacles are involved. If existing deficiencies in stability and simplicity can be overcome, helicopters may come to play an important role as vehicles for individual transportation over short distances. The NACA opened up one interesting possibility when tests confirmed predictions that a person can stand on a platform supported by a thrust jet or a small helicopter rotor simply by taking advantage of his instinctive reactions normally called into play in standing.[17,18] Whether this approach is the proper one only time will tell. It is certainly true that a one-man helicopter need not be expensive to own and operate from the standpoint of power and fuel required and would greatly facilitate movement in regions where highways are inadequate or have become choked with automobile traffic.

So little is known about the actual requirements which may develop in connection with the possibility of using air vehicles for combat purposes that one hesitates to comment on the suitability of a particular type of machine. Because of its inherent load-lifting ability when hovering, the helicopter certainly cannot be ignored in considering the possibility of armed and armored vehicles which are to be capable of slugging it out with light ground installations and landing on practically any kind of terrain. The use of simple one-man helicopters for air cavalry and messenger services appears particularly attractive again on the premise of achievement of a degree of stability such as to reduce drastically skill and training time required for their operation and achievement of simplicity and ruggedness to a practical degree.

Tail Sitters—A discussion of VTOL aircraft is not complete without some consideration of possible tail-sitter configurations, some of which are illustrated in Fig. 7. These were first proposed primarily because they appeared to offer the twin possibilities of minimum structural weight and minimum drag in high-speed forward flight. Where these considerations are of overriding importance, as the case of an interceptor fighter, the tail sitter is still an attractive type.

The tail-sitter configurations suffer from serious disadvantages, however, which probably render them unacceptable for use except in certain special applications. An obvious difficulty is that of the large change in attitude from hovering to forward flight, necessitating rotatable seats for the pilot and passengers. This objection might be overcome by using a prone installation for pilot and passenger for one- or two-place aircraft, but is hardly likely to be acceptable for craft used for general transportation purposes.

The tail-sitter configurations are seriously at a disadvantage as cargo transports because of obvious difficulties in connection with loading and unloading the cargo and providing tie-downs suitable for the large change in attitude. In general, servicing tail sitters is a problem because of the difficulty of reaching items normally requiring service.

Most proposed tail sitters suffer from the very serious disadvantage of being incapable of making use of ground run for overload take-offs and for landings in cases where insufficient power is available for hovering. For certain applications overload take-offs might be accomplished by use of jet assistance or

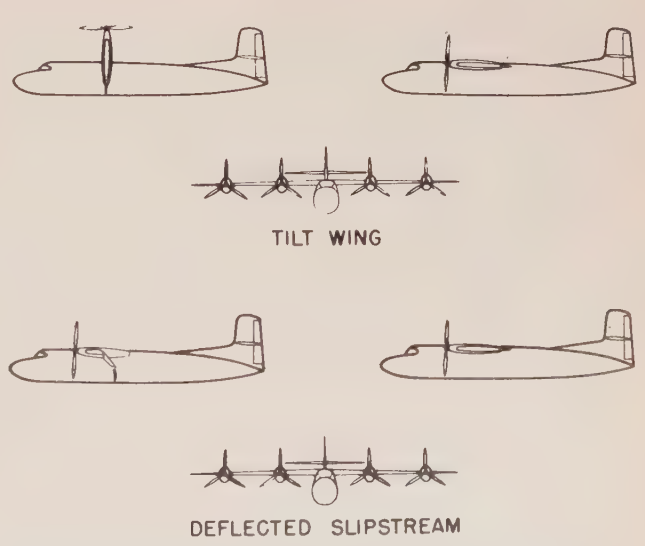


Fig. 8—VTOL airplane configurations using propellers for sustentation when hovering and for propulsion in forward flight

17 NACA RM L52D10 (1953), "Preliminary Experimental Investigation of Flight of Person Supported by Jet Thrust Device Attached to His Feet," by C. H. Zimmerman, P. R. Hill, and T. L. Kennedy.

18 NACA RM L54B12a (1954), "Flight Tests of Man Standing on Platform Supported by Teetering Rotor," by P. R. Hill and T. L. Kennedy.

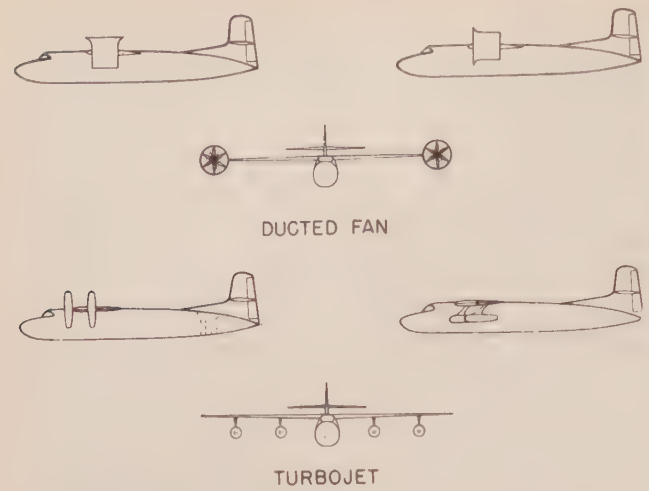


Fig. 9—VTOL airplane configurations using jets for sustentation when hovering and for propulsion in forward flight

Fig. 10—Free-flight test model of tilt-wing VTOL airplane

similar means. In some military applications the inability to land safely following partial power failure may be acceptable, but is not likely to be acceptable in any sort of nonmilitary operations or in military operations where substantially equivalent performance can be achieved with other types.

VTOL Airplanes—The most interesting and promising category of VTOL aircraft in terms of general use for cargo and passenger transportation is the VTOL airplane, which is capable of taking off and landing with ground run as a conventional airplane but can be landed with zero forward speed if necessary or desirable when it has a light fuel load. Figs. 8 and 9 show several possible configurations. It is believed that design, construction, and operational experience with full-scale aircraft of the various suggested types will be necessary before a final choice can be made. Figs. 10 and 11 are photographs of dynamic-scale models of two configurations which have been flown at the NACA Langley Aeronautical Laboratory. These models have already been described.[5,6]

It is the purpose here to discuss these aircraft in terms of their probable use in view of their fundamental capabilities. As among themselves a certain one, or ones, may turn out to be superior to the others, but the major characteristics which fit them for general use as transports and cargo carriers are common to all. These characteristics will be discussed in the following paragraphs.

Of very great importance is the fact that in normal flight operations they are conventional airplanes of proved characteristics and capabilities. In flight above power-off stalling speed the stability, control, and operational problems are the same as for conventional airplanes that have proven their general suitability in many millions of miles of service.

The use of high aspect-ratio wings to support the weight in forward flight at all except the lowest speeds is not only advantageous from the standpoint of range and economy at cruising speeds, but is also of great advantage in making possible endurance and economy at low speeds. Also of very great advantage from a safety standpoint is the ability to fly on a very small percentage of the installed power, assuming proper interconnection or disposition of the propulsion units. (See Fig. 5.)

The configurations illustrated will normally be operated from small airports such as the one indicated in Fig. 12 which will permit operation with useful loads considerably greater than would be possible if they were forced to take off vertically. They will take off in exactly the same manner as conventional airplanes, but will require relatively short ground runs and be able to climb relatively steeply as compared to conventional airplanes. Landings will be made from a power-controlled approach in which it will be possible to control the glide-path angle and the speed independently of each other. This will be done by variation of the power and by variation of the thrust direction, the wing incidence, or the wing-flap setting. The landing technique will be to approach along a flight path making only a small angle with the surface gradually slowing up so that when the airport boundary is reached the speed will have dropped to 30 or 40 knots. From this point under normal conditions of weather and visibility the glide will be steepened, if necessary, and the airplane brought in and flared to a normal airplane-type landing without further speed reduction. However, when weather or airport conditions make it advisable, the speed can be further reduced and even brought to zero before touchdown. Only in rare cases will landings be made by stopping with appreciable altitude and descending vertically. Such an approach will be undesirable from the standpoint of fuel and maintenance costs and from the standpoint of safety, but the ability to make such an approach if necessary will add greatly to the pilot's peace of mind and willingness to operate from small airports and to the overall safety and practicability of

such operation. The power-controlled approach technique will thus make it feasible to operate from small airports under all weather conditions and will eliminate the need for the delays and inconveniences of stacking and use of alternate airports under adverse weather conditions.

Obviously, such aircraft can be used for operation from close-in airports and will be suited for short range and medium range transports. In short range applications, they can be used under conditions that require vertical take-offs and landings. Whether or not they will be superior to or inferior to helicopters in extremely short range and load-lifting operations will depend on such factors as relative ratios of useful to gross weight, relative maintenance costs, relative amount of time spent in hovering as compared to time spent in cruising, and relative ability to operate in all weather conditions. It is probable, however, that developments in stabilization and control devices will make it possible to operate either the helicopter or the vertical flight airplane in all weather conditions, and that this consideration will not be an important factor in the choice between the two.

It is probable that vertical flight airplanes will be used widely as business transports because of their ability to operate between close-in airports and from small plots adjacent to factories and business offices.

The suitability of vertical flight airplanes for such military applications as assault transports, supply transports, command and utility transports, reconnaissance and liaison vehicles in which ability to use small unprepared fields is of very great importance is obvious. There is a great deal of discussion and controversy over the question of whether VTOL or STOL (short, or slow, take-off and landing) aircraft are required for this purpose. It is believed that this controversy will be resolved with a clear-cut victory for the proponents of each type. It is certainly true that a VTOL aircraft will not be satisfactory unless it has good STOL capabilities. On the other hand, it will probably be found that to be truly operable under the desired conditions an STOL aircraft which can fly vertically with light loads for landing and take-off when necessary or desirable will be sufficiently superior in actual operational use to STOL aircraft not having the VTOL ability that the former type will come into general use. The controversy will then be resolved by the adoption of STOL aircraft which can VTOL under emergency conditions, a clear victory for the STOL proponents. However, from the other point of view they will be VTOL aircraft with extremely good STOL capabilities, a clear victory for VTOL proponents. A suitable designation for this airplane might be STOVL (slow take-off, vertical landing) airplane.

A rosy picture is painted of the use of vertical flight airplanes for the future. If these airplanes are so attractive, why are they not being built for the enumerated purposes to the exclusion of all others? One reason is inertia. It takes time for the possibilities of new types to become sufficiently apparent and generally realized to permit their actual construction and trial. A second is the lack of full-scale flight experience with such airplanes. A very limited amount of full-scale flight experience has been obtained with one experimental VTOL airplane which utilized jet engines for hovering. This flight experience did not include study of landings over obstacles under adverse conditions.

A third very important consideration is that the need for vertical landing capability as compared to slow landing capability through use of power in order to operate under all weather conditions cannot be considered to be fully established. From the example shown in Fig. 6 it is apparent that an STOL airplane having half the installed horsepower necessary for VTOL operation can maintain level flight at a low speed. In the case shown a minimum speed of 35 knots is indicated. Such an airplane is theoretically capable of operating from small airports. Obviously an STOL airplane will be considerably less expensive than the comparable VTOL airplane. The VTOL airplane used for STOVL operation will fall somewhere

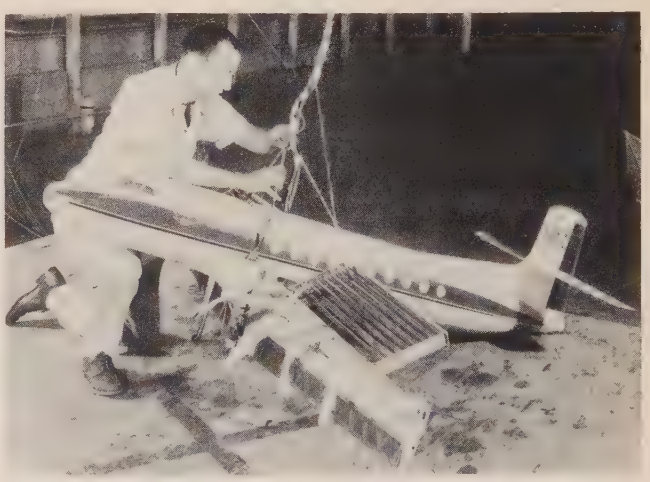


Fig. 11—Free-flight test model of deflected-slipstream VTOL airplane

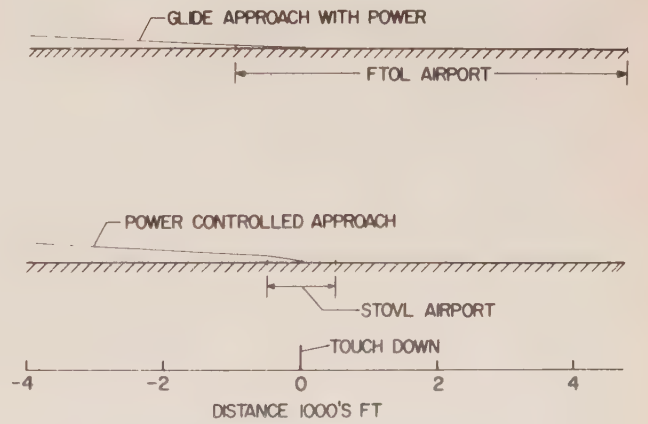


Fig. 12—Comparison of airport requirements for conventional (FTOL) and STOVL airplanes

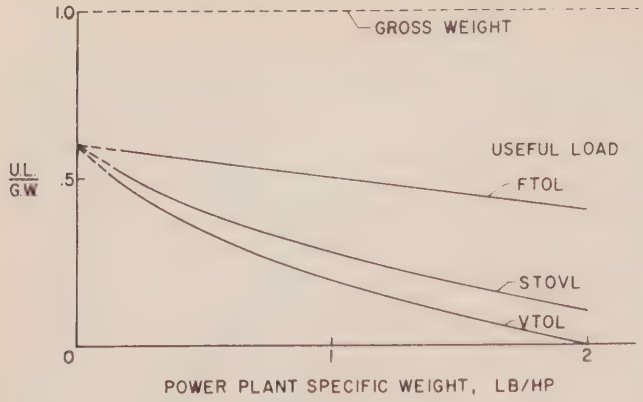


Fig. 13—Variation of ratio of useful load to gross weight with powerplant specific weight

between the two. The crux of the matter lies in whether the STOL airplane can actually use the low speeds of which it is theoretically capable under actual all-weather operating conditions. There is some reason to believe that full use of the low-speed capabilities theoretically possible is not feasible and that actual operation must be limited to speeds well above the theoretical minimum *unless the power, stability, and controllability, necessary to VTOL are available.* This reasoning is based partly on the fact that experimental STOL airplanes have been built from time to time during the past 30 years, but pilots have apparently been reluctant to make use of their slow-landing capabilities as a regular operational procedure. Whether the next crop of STOL airplanes will be similarly handled remains to be seen. Better powerplants and more positive control at slow speeds through use of control devices developed for VTOL airplanes may alter the STOL picture. It is certainly true that there is reluctance to go all out toward development of VTOL airplanes until it is definitely established whether VTOL capability is essential to satisfactory all-weather operation from small airports.

It is undoubtedly true that some penalty will necessarily be paid in payload in order to achieve VTOL capability. It is also true that the airplane will cost more to build than the conventional FTOL (fast take-off and landing) airplane. These facts result from the necessity of using some of the payload capacity for additional powerplant. How much reduction in payload will actually be involved is the subject of considerable study at the present time. A survey of available engineering estimates indicates that a true VTOL airplane will have about half the load-carrying ability of its conventional counterpart assuming use of turboprop powerplants currently becoming available. Obviously, a VTOL airplane used only in STOVL operations need not suffer to such an extent, since it can take off with a larger useful load than its counterpart required to take off vertically, the increase being roughly equivalent to the weight of the fuel to be used on the mission, as discussed earlier.

Fundamentally, the question of whether to use VTOL, STOL, or conventional (FTOL) fast take-off and landing airplanes for transport operations is going to have to be resolved on the basis of overall economics and utility in which the cost of more expensive airplanes will have to be weighed against the cost and vulnerability of large airports and the cost of passenger inconvenience and dissatisfaction with stacking delays and use of alternate airports in bad weather. It is believed that the time has been reached when designers, operators, and municipalities must consider whether additional funds should be expended on large airports or whether the airplanes should be built to operate from small ones. Figs. 1 and 13 illustrate certain trends which bear on this subject. Fig. 1 shows the weight per horsepower trend for piston engines and turboprop engines during the past 10 years. Notice that the trend is still downward for turboprop engines.

The effect of change in engine specific weight upon the ratio of useful load to gross weight is shown in a general way in Fig. 13. It is obvious that if the powerplant weight dropped to zero from 1 lb per hp there would only be an increase of 20% in the useful load. If it dropped to zero from 0.5 lb per hp there would be an increase of only 10% in useful load.

With powerplants weighing more than 2 lb per hp, the useful load possible with VTOL is indicated to be zero. However, with powerplants weighing of the order of ¾ lb per hp, the useful load is of the order of half that for a conventional airplane, or 25% of the gross weight. If the powerplant weight could be reduced to zero, then the ratio of the useful load to gross weight could approach 60%, or the same value as for the FTOL airplane with zero powerplant weight.

The point of this discussion is that as powerplant weight goes down the penalty for VTOL also goes down. It is believed that at some point of the curve the further benefit of utilizing lower powerplant weights to provide greater ratios of useful load to gross weight will be less remunerative than using them to provide VTOL capabilities. This is true to an even greater extent when STOVL operation is considered.

The curves of Fig. 13 obviously are oversimplifications. No attempt will be made to defend specific values other than to say they are believed to be approximately true. It is also certainly true that the powerplant weight will never reach zero. However, despite the obvious deficiencies and shortcomings of Fig. 13, it is believed that the trends are truly indicated. It is further believed that designers and operators will do well to explore this subject to determine where the cross-over point lies beyond which it is desirable to use VTOL airplanes for transport purposes.

Conclusion

The relatively low fuel consumption and low jet velocities of helicopters using large, lightly loaded rotors will insure the continued use of this type of air-

craft in applications where hovering is an important part of the flight mission. Development is needed, however, along lines leading to simplicity, ruggedness, and freedom from vibration. Much also needs to be done in the way of making helicopters easier and simpler to fly to reduce necessary training time and to permit their all-weather use. A simple, easily flown helicopter may prove to be the answer to the problem of individual flight either for civilian purposes or for military uses such as air cavalry.

Because of superior overall performance possibilities when forward flight is the primary function of the aircraft and hovering is used only for very short time intervals, or not at all except under emergency conditions, the VTOL airplane appears likely to come into general use for a wide range of applications in which short or moderate range missions are involved and the ability to operate into and out of small fields in all kinds of weather is important. It has been proved that VTOL airplanes can hover and can satisfactorily make the transition from hovering flight to forward flight as an airplane. Weight and performance studies indicate their practical possibilities. However, only a relatively small amount of full-scale flight experience exists with such aircraft. The choice between the various types shown in Figs. 8 and 9 and other possible ones not shown may well lie in the exact shape of the power-required curve at speeds between hovering and minimum flight speed as a wing-supported airplane not only from the standpoint of performance but also from the standpoints of stability, controllability, and safety, and its overall effect upon the willingness of pilots to operate the aircraft from small airports under all weather conditions. The most urgent need at present is full-scale flight experience with such airplanes, particularly experience in landings in small fields under conditions representative of all-weather operation.

DISCUSSION

Stresses Importance of Structural Efficiency in VTOL Design

—Z. M. Ciolkosz
Piasecki Aircraft Corp.

MR. ZIMMERMAN's interesting paper stresses as a factor of great importance the rapid increase in power required for hovering and slow-speed flight with increase in weight. Fig. 6 in his paper shows also how rapidly power to hover decreases with weight reduction.

I would like to elaborate on this important aspect of VTOL aircraft and show the implications associated with minimum WE (weight empty) on VTOL design, especially on shaft-turbine-driven aircraft.

Since in our present-day VTOL designs, power to hover is predominant and not the power required for high-speed conditions, endeavor must be made to utilize the power for hovering in the most economical way. Although the most economical way to secure high lifting efficiency will be to use a propeller of the helicopter rotor size, the desire for high forward speed and physical limitations restrict the selection of such a propeller.

Once the propelling system has been determined for given VTOL design based on either the shrouded or unshrouded propeller, highest efficiency in utilization of developed thrust is then of the greatest concern.

Many papers have discussed different methods of the application of propeller thrust for vertical flight. These methods use direct propeller thrust (tail sitters, rotating wings, or rotating propellers with fixed wings).

Other methods use deflection of the propeller slipstream. Reduced efficiency due to the deflection is considered as a penalty, and the adverse effect on the ability to develop successful VTOL aircraft has been frequently mentioned.

The range of efficiency for different methods of thrust application varies for different design methods. To obtain true figures of the efficiency of design, all factors such as propeller profile drag, reduced efficiency due to tip losses, frictional losses in the slipstream, loss of energy due to the rotation of the slipstream, and pitching moment compensation must be considered. In other cases, account must be made of the gain in thrust due to the composite methods of traction.

Having arrived at certain solutions which appear to be the best, the question which still remains open is, "Is the structural solution for VTOL the best, and its weight penalty a minimum, and how does the weight penalty associated with VTOL design affect the power requirements?" If the weight penalty is relatively high, *even very good aerodynamic efficiency will be overshadowed quickly by predominant weight effect. Consequently, the best VTOL solution is that solution where good aerodynamic efficiency is combined with high structural efficiency.*

On the other hand, the structural efficiency is closely tied to the selected solution, and once the solution is accepted for some reasons other than structural efficiency, not much can be done in this respect.

The starting point to evaluate the structural efficiency will be a comparison with existing FTOL aircraft. The wealth of data available will permit us to evaluate the weight of such an aircraft within close limits.

Fig. A shows WE and GW (gross weight) for a few shaft-turbine-driven commercial aircraft. Those selected were aircraft with approximately 10-lb per hp loading. The WE/GW ratio varies from 0.55 to 0.60, but these figures are rather high since military transport aircraft are already flying in this country where this ratio is less than 0.50. The commercial aircraft data were presented because they are not of a confidential nature.

The value of WE/GW = 0.55 and power loading 10 lb per hp

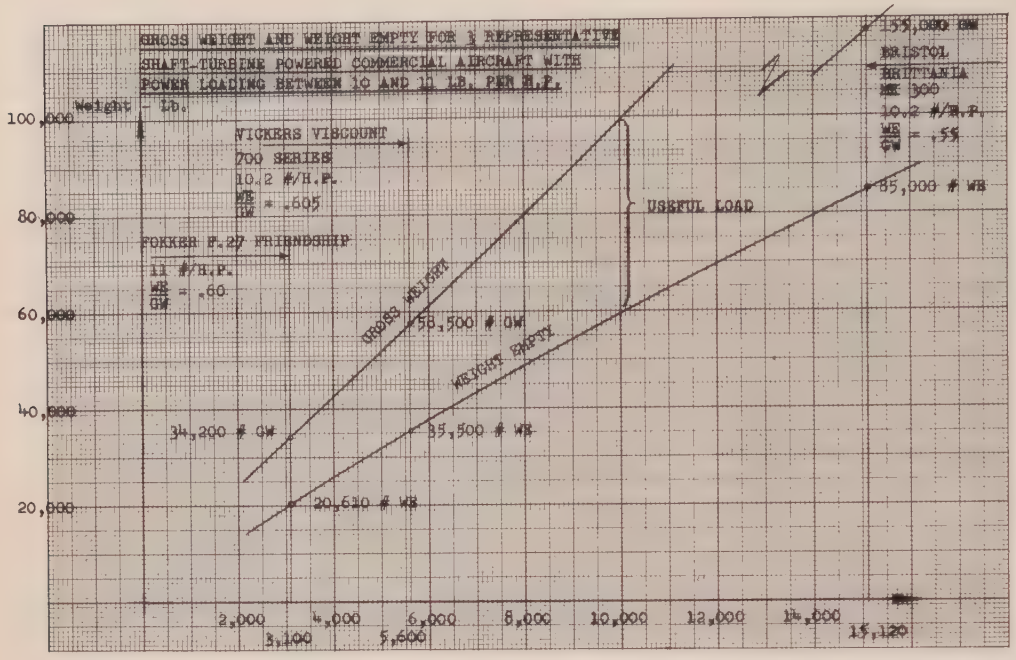


Fig. A—Gross weight and weight empty for three representative shaft-turbine-powered commercial aircraft with power loading between 10 and 11 lb per hp

were selected as the basis to determine the average GW and WE of shaft-turbine-powered FTOL aircraft. This is shown in Fig. B.

The conversion of FTOL transport aircraft into VTOL aircraft requires quite a few changes.

First of all, to simplify the comparison, a VTOL design was selected in which the power loading in hovering was half of the FTOL power loading; namely, 5 lb per hp. The upper horizontal scale shows installed power in VTOL design.

The weight penalty for the conversion into VTOL aircraft was then calculated. It consists of:

1. Additional power to decrease power loading from 10 to 5 lb per hp.

2. Additional engine sections and accessories.

3. Additional cooling system for zero and low forward speed for all engines installed.

4. Additional propeller weight for increased power.

5. Interconnecting transmission and gear boxes essential for the safety of operation.

6. Additional structure "beef-up" to provide for increased speed.

7. Structure weight for VTOL features such as rotating wing, large flaps, tilting engines, and actuating mechanisms.

8. Additional control system for zero and low-speed conditions.

The additional weight for VTOL aircraft with conventional wing design (either rotating or provided with large flaps) and multiplicity of propellers amounts to approximately 18.5% of GW.

Since my first assumption was that, for FTOL, WE = 55% of GW, therefore, for VTOL design, WE will go up from 55% of GW to 73.5% of GW and the UL (useful load) will be 26.5% of GW.

The new line was drawn on the graph represented on Fig. B showing the weight penalty for transformation of conventional aircraft into VTOL, and the space between the ΔWE line and the WE (for FTOL) line represents UL for VTOL aircraft.

This study represents the classical approach to VTOL weight problem and up to this point the figures are in very good agreement with Mr. Zimmerman's paper.

If some other methods of design are selected which may reduce the weight penalty for conversion into vertically performing aircraft, the saving in power will be striking. Possible methods to reduce the weight penalty are:

1. Selection of the design where the same elements have different assigned functions for different regimes of flight.

2. Elimination of the rotation of heavy structure items, and application instead of light items easy to control and easy to change their relative position in flight.

3. Reduction of the number of propellers by application of composite method for thrust development.

4. Elimination of interconnecting shafting without endangering the safety features.

5. Reduction of structural penalty for VTOL design by avoiding concentrated high loads and high bending moments, and providing high design rigidity.

6. Design of the aircraft control system with the minimum new elements to eliminate duplication of control system for VTOL flights.

Such a design, if carried to the extreme, can limit the weight penalty for VTOL design only to the weight of additional power system plus some small penalty for zero and low forward speed control system.

For the sake of this discussion and graphical demonstration, a structural solution was selected where ΔWE was limited to 12% of GW instead of 18½% (say for GW of 100,000 lb).

If the design requirements ask for a *definite UL*, the GW of the VTOL aircraft may be decreased (for the same UL) to 80,000 lb, and the associated installed power from 20,000 to 16,000 hp, but this is only part of the power saving.

If physical limitations were dictating the size of the propellers

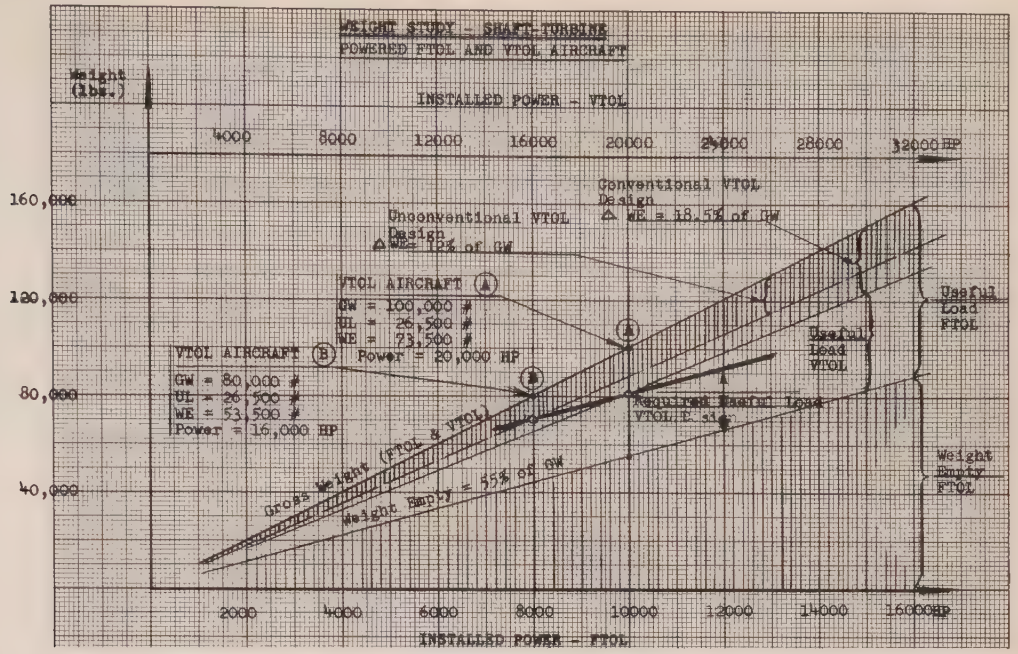


Fig. B—weight study of shaft-turbine-powered FTOL and VTOL aircraft

in our first selection, and this size does not need to be changed, the thrust loading may be decreased by 20%, with further reduction in power required to hover. (Ideal power is considered only.) This is shown on Fig. C.

Summarizing, by selecting a design for the VTOL aircraft which enables us to reduce weight penalty for VTOL features from approximately 18½% to 12%, (for the same UL) the GW was reduced to 80% of the initial weight.

The installed power was reduced twice: first, because GW was reduced by 20%, and second, because a decreased propeller disc loading reduced power required per lb of thrust. The second reduction amounts to an additional 11% saving. In simple figures, for the investigated VTOL aircraft, *reduction by 6½% of the GW reduces the power installed (for the same UL) by 28.5%.*

This example shows clearly that although aerodynamic efficiency is very important and for optimum VTOL design cannot be neglected, selection of the design which offers the best chances of reducing the WE *may have more pronounced repercussions on total efficiency of the design.*

In conclusion, the need should be emphasized for a new design criterion for VTOL evaluation. By combining aerodynamic and structural efficiency into design efficiency, a much better compari-

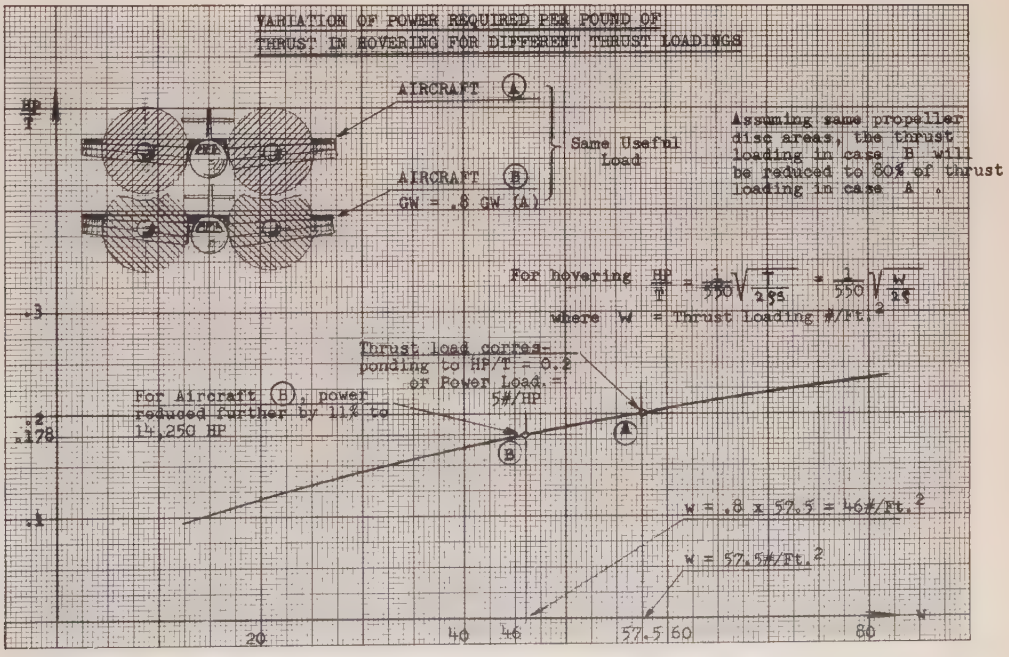


Fig. C—Variation of power required per pound of thrust in hovering for different thrust loadings

son of proposed VTOL aircraft with FTOL aircraft can be obtained.

Improvement of Range Efficiency, Adaptability Necessary for Successful VTOL Aircraft

—J. Beebe
Army Transportation Corps

IN THE quest of recent years to increase maximum flight speeds, the field of low-speed aerodynamics has been badly neglected. The advent of vertical rising aircraft in the form of the helicopter proved the value and the versatility of such aircraft. The configuration which makes the helicopter efficient as a vertical lifting device, makes it inefficient in forward flight.

Many schemes have been devised to reduce the adverse effects of the helicopter rotor in forward flight, in an effort to improve the range efficiency of this machine. However, it remained largely for Mr. Zimmerman and his associates at NACA to prove that the soundest approach was not to build a better helicopter, but to build a conventional wing type of aircraft capable of rising vertically. As pointed out by Mr. Zimmerman, this was not practical until the advent of the gas turbine engine having low installed weight/power ratios.

Except for very special missions, the actual landing and take-off represent only a very small per cent of the total flight time of an aircraft. Obviously then, range efficiency assumes major importance. Such VTOL types as the "tilt wing" and the "deflected slipstream" hold promise of obtaining overall range efficiencies approaching those of conventional aircraft. Under conditions of medium range they promise the possibility of excelling conventional aircraft if full advantage can be taken of their flight characteristics. In fact, except in cases where the helicopter can sling-carry cargo externally, these types may well challenge the helicopter even at short ranges.

Mr. Zimmerman points to the importance of specific weight of powerplants. Of equal importance will be good propulsive efficiency over the entire flight speed range. Admittedly, vertical flight can be achieved by brute power on one hand or increased propeller diameter on the other, but these approaches are economically undesirable. The brute power approach leads to poor specific fuel consumption during cruise (assuming gas turbines, low or medium altitudes, and that no engines are shut down) and increasing diameter results in propeller unloading with increasing speed. In this regard, the ducted fan or propeller appears to offer interesting possibilities. This device has received little attention until recently when its possible application to VTOL types was realized.

Of importance from a commercial as well as a military standpoint, is the characteristic ability of most VTOL types to accept large overloads under conditions where runways are available. For certain types, overloads of 100% or more are not unrealistic.

The Air Force and the Navy have generally utilized the helicopter for very specialized missions, such as rescue. The Army, in its quest for an aerial vehicle which could "live with the troops" and not require prepared landing and take-off areas, has leaned heavily on the helicopter. The disadvantages of this machine have been pointed out and are well recognized. Realizing the distinct advantages of improved VTOL types, the Army has supported such projects as the XV-1 and XV-3 and is currently funding the most extensive program in this area.

One problem associated with VTOL aircraft, generally overlooked, but which must be solved before full utility can be obtained from this type is the problem on landing gear suitable for landing and handling of the aircraft on all types of surfaces. Many types, which might be applicable, have been tried on helicopters, but a completely satisfactory type remains to be proved.

It is quite certain the next few years will bring forth aircraft combining VTOL and high-speed capabilities with far greater utility than any possible today.

ORAL DISCUSSION

Reported by F. Mamrol
Piasecki Aircraft Corp.

D. C. Prince, consulting engineer: Have the free gas turbines been considered in the study of VTOL aircraft?

Mr. Zimmerman: They have been considered by Lewis Laboratory of the NACA. The relative propeller speed requirements of VTOL aircraft are opposite to those of helicopters, since VTOL airplanes require high propeller rotational speeds in hovering and low speed flight. There is also concern about control of the engine at low flight speeds, and hence, the fixed turbine might be better.

A. I. Roman, United States Research & Development, Inc.: Have lift and drag measurements been made at high angles of attack of the flapped wing and propeller combination? Does the propeller slipstream delay the stall?

Mr. Zimmerman: The lift-drag curves usually have a peak after which the lift decreases with increasing drag. The propeller slipstream tends to delay stall. It might be desirable to add leading edge slats for high turning angles.

J. C. Squiers, Continental Aviation and Engineering Corp.: Where does the jet flap fit into the picture of these new type aircraft?

Mr. Zimmerman: The NACA is working on the subject. The jet flap can cause very large reductions in landing speeds. For instance, the DC-8 with the jet flap could have a landing speed of 60 knots. The structural problem of carrying the jet-engine hot gases through the wing would be severe, and because of the large pitching moment, the jet flap might be better suited to short take-off and landing rather than VTOL aircraft.

F. H. Sharp, Convair Division, General Dynamics Corp.: Is there any problem with pilot's reaction time in flying the scale model VTOL aircraft at NACA?

Mr. Zimmerman: The reaction time required decreased as the size of the aircraft decreased, but the NACA had no particular trouble in this respect. The NACA uses four pilots (one each axis plus one power) to fly its models, and these pilots are experienced in flying these models with "flicker" (on-off) controls. Since these pilots have not flown full-scale VTOL aircraft, the scale effects on time cannot be compared.

A. Satin, Office of Naval Research: There is a program designed to obtain information on VTOL aircraft through flying test-beds. Among the types that will be sponsored by the military departments and industry are the Collins Radio/Lippisch Aerodyne, the Fairchild/Hunt Dragonfly (vectored slipstream), and the Breguet STOL deflected slipstream. Both the STOL and VTOL aircraft cannot be separated at this stage because many VTOL aircraft have STOL capabilities.

J. Beebe, Army Transportation Corps: The DeLackner Rotorcycle has a universal landing gear. The success of VTOL aircraft may well depend upon their ability to actually "land" any place on the earth's surface.

Mr. Woods: I believe that it is unrealistic to put effort into STOL programs.

An Approach To Obtaining Road Octane Ratings In A Single-Cylinder Engine

E. M. Barber, H. I. Wilson, and T. H. Randall, Texas Co.

This paper was presented at the SAE Summer Meeting, Atlantic City, June 6, 1956.

THREE different octane numbers commonly are used to describe the antiknock quality of motor fuels. Two of these, the Research [1] and Motor [2] octane numbers, are measured by laboratory tests in the ASTM-CFR knock test engine. The third, the road octane number, commonly is measured in cars operated on the road under test conditions which rate the fuels in accordance with customer-type experience. In general, all three of these octane numbers are different. Even though the road octane number is most descriptive of the antiknock quality of a fuel in service, it is necessary, practically, to base direct manufacturing control of the antiknock quality of motor fuels on a laboratory test.

The magnitude and causes of the differences between laboratory and road octane numbers have been the subject of numerous investigations. The laboratory test methods have undergone several minor revisions. From time to time there has been a shift in emphasis between the Motor and Research methods, and various functions of these two laboratory tests have been correlated with road octane numbers.[3-8] Currently, differences of as much as 10 octane numbers may occur between the laboratory and road octane numbers of a fuel. In the various investigations of octane ratings it has been shown that the octane number of a fuel depends both on the composition of the fuel and on the operating conditions of the engine in which the measurement is made. In this connection it should be noted that the road octane value of a fuel is not a single-valued number; somewhat different road octane values may be obtained for a fuel from each of a group of automobiles. For a typical group of automobiles, however, these differences are usually small compared to differences between road octane values and the ASTM Research- and Motor-method octane values.

General Plan of Investigation

It is generally accepted that knock results from spontaneous ignition of the end gas,[9] and that knock does or does not occur depending on whether the end-gas reactions do or do not reach the point of

1 Standard Method of Test for Knock Characteristics of Motor Fuels by the Research Method, ASTM Designation D908-55.
2 Standard Method of Test for Knock Characteristics of Motor Fuels by the Motor Method, ASTM Designation D357–53.
3 "Relation between Road and Laboratory Knock Ratings," by J. B. Duckworth, G. T. Moore, and C. J. Domke. Presented at ACS Symposium on Relationship of Fuel Properties to Engine Design and Future of High Octane Fuels. Sept. 6–10, 1948.
4 SAE Transactions, Vol. 43, 1938, pp. 416–420: "Correlation of Road and Laboratory Octane Numbers," by J. R. Sabina.
5 Proceedings of the API, Vol. 3, November, 1937, pp. 98–111, 124–125: "1937 Road Knock Tests: Report from Cooperative Fuel Research Committee," by T. A. Boyd.
6 SAE Transactions, Vol. 30, 1935, pp. 165–179, 215–223: "CFR Committee Report on 1934 Detonation Road Tests," by C. B. Veal.
7 SAE Transactions, Vol. 28, 1933, pp. 104, 105–120: "Antiknock Research Coordinates Laboratory and Road Tests," by C. B. Veal, H. W. Best, J. M. Campbell, and W. M. Holaday. See also pp. 187–190 for discussion.
8 *Petroleum Processing*, August, 1951: "Road Antiknock Performance of Motor Gasolines," by M. L. Alspaugh and C. A. Hall.
9 End gas is the last portion of the charge to burn, that is, the last part of the charge to be reached by the flame front.

An approach to obtaining laboratory single-cylinder engine fuel octane ratings that agree with car road octane ratings is described.

The technique is based on attaining, in the single-cylinder engine, a general duplication of the combustion environment which exists in the multicylinder automotive engine. Because it proved suited to this purpose, a special single-cylinder power section mounted on a CFR crankcase was used. The influence of speed on fuel borderline ratings was in substantial agreement with automotive engine results.

Results were obtained on a variety of commercial gasolines and special blends representing extreme variations in hydrocarbon composition. Comparative rating work covered a variety of 1947–1952 and 1955 cars; automotive engine compression ratios ranged from 6.5 to 9.0, and ratios of 10, 11, and 12 were obtained with special engines, heads, and pistons. For fuels of 68 to 105 octane number, single-cylinder engine ratings agreed with car ratings with a maximum spread of three octane units. In contrast to this, the ASTM Research and Motor ratings deviated by as much as 10 octane units.

spontaneous ignition before the end gas is consumed by the normal progress of the flame front. For a given engine and fuel-air mixture, the progress of the end-gas reactions has been related to the temperature, density, and residence time of the end gas.[10] It is a convenience to think about the knock-rating problem in these terms, for the many operating variables that are known to affect the occurrence of knock can then be resolved into their effects on these more fundamental quantities. Similarly, combustion-chamber design factors such as length of burning path, combustion duration, and surface-to-volume ratio can be treated as affecting end-gas residence time and temperature. This temperature-density-time concept has been employed frequently in recent years.[11,12,13] It has been employed in the present investigation indirectly as a way of thinking about the problem and as a measure of the combustion environment. Temperature-density-time parameters, however, do not occur in the measurement or expression of fuel ratings.

Two preceding reports [14,15] presented information on the combustion environment in the standard knock test method and in a substantial group of automotive engines. Some comparisons from these earlier reports are shown in Fig. 1, and it will be noted from this and from other well-known characteristics of their comparative design and operating conditions that the knock test methods rate fuels at quite different conditions than do automotive engines.

In the present investigation, the approach to the problem of improving the correspondence between laboratory and road octane ratings is based on the assumption that differences between laboratory and road octane numbers result from these discrepancies between the combustion environment of the standard knock test methods and of typical automobile engines. Starting with this point of view, the investigation was planned and carried out in three phases, approximately as follows:

1. The basic ASTM knock test unit was used. Operating conditions (cylinder charge density, intake air temperature, mixture strength, and the like) were selected to produce typical automotive values of compression density, temperature, and mixture strength.

2. Certain combustion-chamber features which affect combustion duration, length of burning path, and surface-to-volume ratio were investigated; all of these have important influences on the knock reactions. Adaptability to operation over a range of compression ratios also was a consideration. Typical automotive operating conditions were used.

3. A combustion chamber was selected and operated at typical automotive conditions. Various types of fuels were rated, and ratings were compared with road octane values obtained in a variety of makes and models of automobiles.

This approach involves the possibility of an occasional modification in the laboratory test to reflect the influences of major changes in automotive practice. In the following, it will be noted that laboratory engine-automobile engine octane-number comparisons have been carried out successfully over a period of years during which developments in automotive practice have been occurring at a rapid pace.

ASTM Knock Test Engine

In this phase of the work, an engine setup similar to that for the ASTM Supercharged method [16] was

[10] Pp. 198–212: "Chemical Background for Engine Research," by E. F. Frock, F. D. Rossini, F. C. Whitmore, G. von Elbe, B. Lewis, and O. Beech. Pub. by Interscience Publishers, Inc., New York, 1943.

[11] "La Combustion Détonante dans les Moteurs à l'Explosion," by M. Serruys. Pub. by Pub. Sci. and Tech.

[12] NACA TR 655 (1939), "Knocking Characteristics of Fuels in Relation to Maximum Permissible Performance of Aircraft Engines," by A. M. Rothrock and A. E. Bierman.

[13] NACA ARR E6E13, (1946), "Correlation of the Effects of Compression Ratio and Inlet Air Temperature on Knock Limits of Aviation Fuels in a CFR Engine II," by H. E. Alquist, L. O'Dell, and J. C. Evvard.

[14] Proceedings of the API, Vol. 3, June, 1947, pp. 103–115: "Some Characteristics of Combustion in ASTM-CFR Knock Test Engine," by E. M. Barber, L. E. Endsley, Jr., and T. H. Randall.

[15] SAE *Quarterly* Transactions, Vol. 2, July, 1948, pp. 401–411: "Knock-Limited Performance of Several Automobile Engines," by E. M. Barber.

[16] Tentative Method of Test for Knock Characteristics of Aviation Fuels by the Supercharge Method, ASTM Designation D909–49T.

[17] Charge air temperature, that is, mixture temperature multiplied by r^{n-1}, where r is compression ratio, and n is the exponent of polytropic compression in the cylinder, yields the "compression temperature" at top center of the compression stroke.

[18] Charge air density multiplied by $(r-1)$, where r is compression ratio, yields the "compression density" at top center of the compression stroke.

[19] Knock intensity was determined by a Sperry KM-1 knockmeter in the Borderline testing and by a Sperry Model KM-3 knockmeter in bracketing testing discussed later.

[20] Borderline Test Procedure for Determination of Antiknock Quality of Fuels on the Road, CRC Designation F-8A-943.

used but with the substitution of a variable speed dynamometer and with added means to control fuel/air ratio, exhaust back pressure, and intake air pressure and temperature as functions of speed. Typical automotive conditions of mixture strength and of compression temperature [17] and density [18] were imposed on the single-cylinder engine. Numerical values of these parameters were selected to correspond approximately to the automotive conditions that had been observed in tests on a substantial group of 1939–1942 model automobiles.[15]

The single-cylinder engine was operated at fixed compression ratio and several speeds; spark timing was varied to produce knocking [19] in a fashion which simulated the Borderline Road Octane Test Procedure.[20] Reference fuel frameworks and borderline fuel ratings were developed in the single-cylinder engine and in the above automobiles. Comparative reference fuel frameworks for the single-cylinder engine and for one of the automobiles are illustrated in Fig. 2. It will be noted that there is at least a family resemblance between the two frameworks. Neither the reference fuel frameworks nor the fuel ratings, however, were in as good agreement as seemed desirable and possible.

Combustion-Chamber Investigations

In considering what factors may have been responsible for the above rating results, available engine performance data were reviewed. This review directed attention to several characteristics of the CFR combustion chamber: the relatively long duration of combustion, the length of burning path, and the disc-shaped combustion chamber. The long duration of combustion is suggested by the comparatively high maximum power spark advance shown in Fig. 2. The other two features are illustrated in the schematic diagram of Fig. 3 in comparison to a domed combustion chamber that will be mentioned later. Experience, backed up by the reaction-rate (temperature-density-time) theory of knocking, suggested that these factors might have a substantial effect on fuel ratings. Hence, work was conducted to bring them into better accord with automotive practice.

Desiring to use the variable-compression-ratio ASTM-CFR cylinder, if possible, the first work involved attempts to obtain a more desirable result in this cylinder. These attempts included: (1) the use of

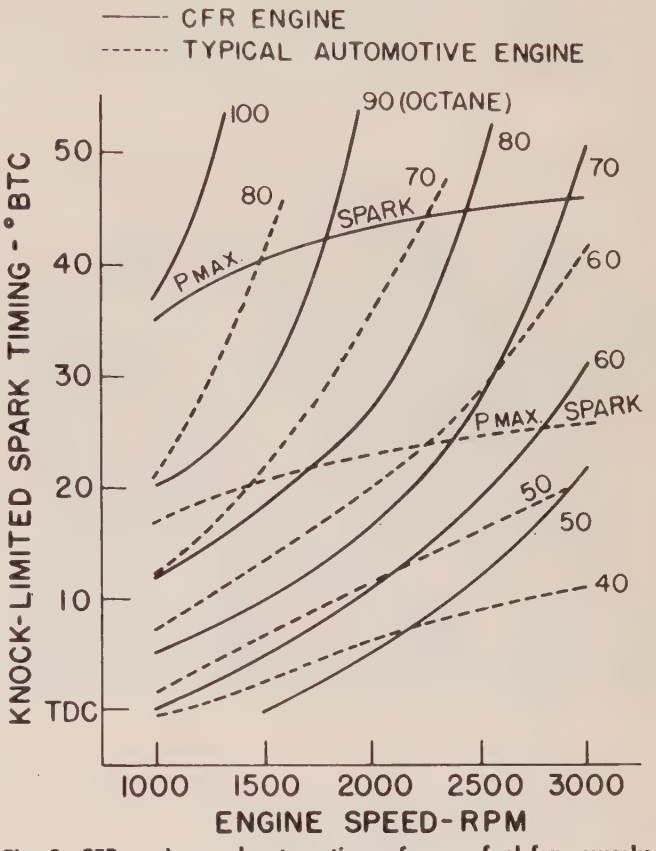


Fig. 2—CFR engine and automotive reference fuel frameworks

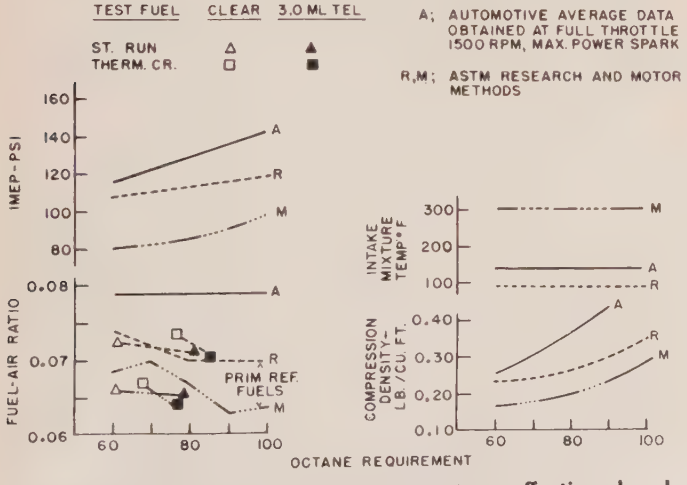


Fig. 1—Variation in important parameters affecting knock

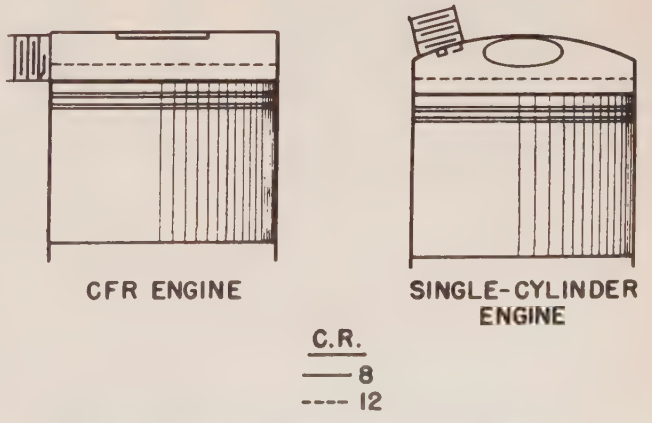


Fig. 3—Comparative combustion-chamber geometry

Fig. 4—Single-cylinder assembly

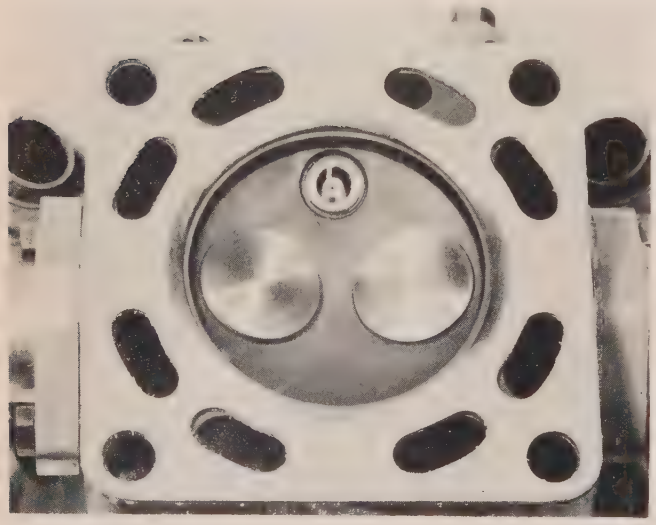


Fig. 5—Single-cylinder head

Table 1—Car Engine Characteristics

Code	Year	Head Type	Compression Ratio	Number Cylinders	Displacement, cu in.	Transmission Type
A-1	1948	OV	6.50	6	216.5	Synchromesh
A-2	1948	OV	6.50	6	216.5	Synchromesh
A-3	1948	OV	6.50	6	216.5	Synchromesh
A-4	1948	OV	6.50	6	216.5	Synchromesh
B	1947	OV	6.30	8	248.0	Synchromesh
C	1947	L	6.75	8	239.0	Synchromesh
D	1947	L	6.60	6	217.8	Synchromesh
E	1947	L	6.50	8	248.9	Synchromesh
F	1949	OV	7.43[b]	8	303.7	Hydramatic
G[a]		OV	8.12[c]	8	287.0	Hydramatic

[a] Special engine G was installed in car F for test purposes.
[b] As measured, nominally 7.25.
[c] As measured, nominally 8.00. A special high compression test engine.

a shrouded inlet valve, (2) dual ignition, and (3) piston shape modifications to induce "squish" type turbulence. These schemes either were ineffective or introduced other abnormalities. For example, dual ignition combined with a shrouded inlet valve resulted in a burning time comparable to automotive engines. At a given octane level, however, the change in knock-limited spark timing per unit change in

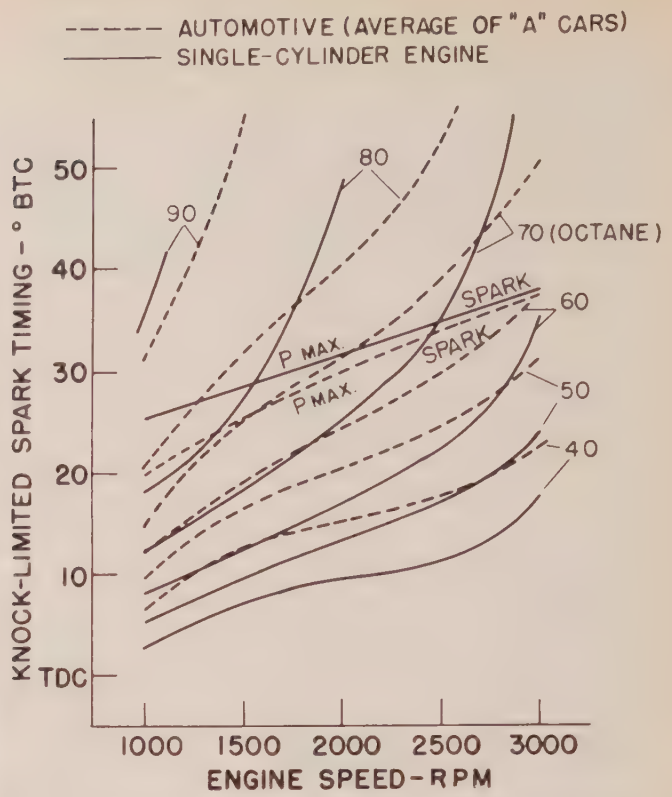


Fig. 6—Single-cylinder engine and automotive reference fuel frameworks

speed became very large at the higher engine speeds.

After these investigations of the CFR cylinder, attention was focused on a special cylinder head and cylinder assembly. This is the slightly domed combustioned chamber illustrated in Fig. 3. It will be noted that the more nearly central location of the spark plug gives a reasonably short burning path and that the doming creates a mild "squish" turbulence. It was not practical to include an automotive type "quench zone" and still provide for continuously variable-compression-ratio adjustment. Actual construction is illustrated by Figs. 4 and 5. The assembly was built for fixed-compression-ratio operation, the desired ratio being selected by shims placed between the cylinder base and the deck of the crankcase.

Domed Combustion-Chamber Operation

Preliminary results with the domed combustion-chamber cylinder head and barrel assembly, used with typical automotive operating conditions, were quite encouraging. This combination has been retained throughout the remainder of the investigation, and it is referred to hereafter simply as "the single-cylinder engine."

Comparative Borderline Road Octane Test Procedure type of reference fuel frameworks and fuel ratings were obtained for the single-cylinder engine and for four 1948 automobiles of the same make and model (Cars A of Table 1). Comparative reference

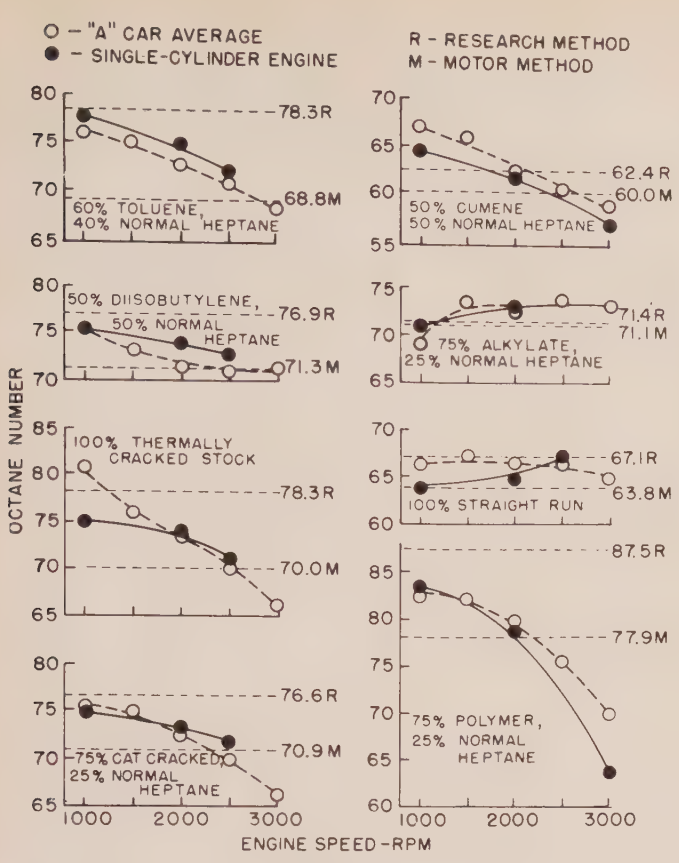


Fig. 7—Single-cylinder engine and A car average borderline ratings

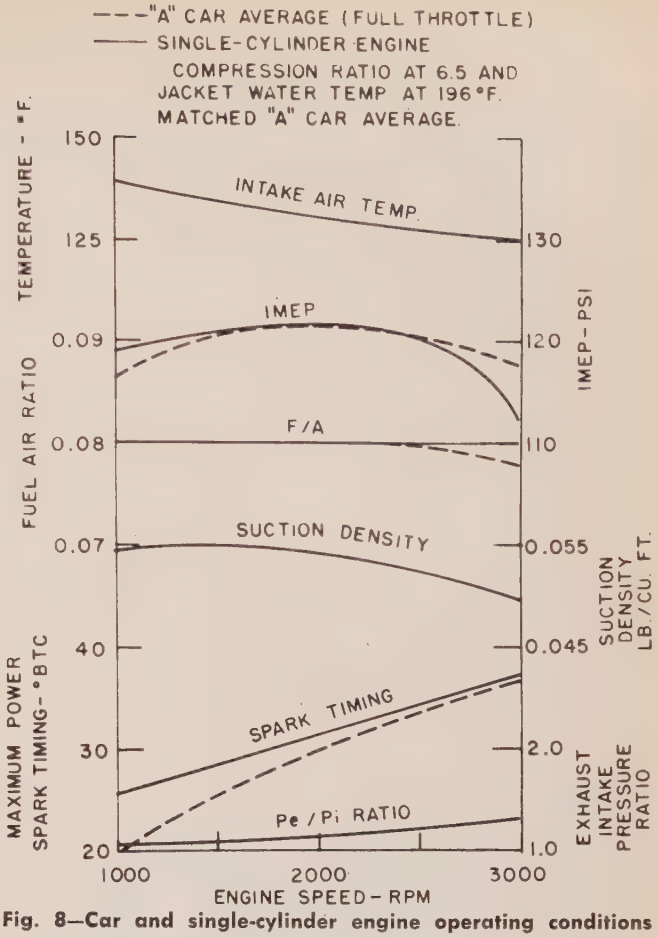


Fig. 8—Car and single-cylinder engine operating conditions

fuel frameworks and maximum power spark advance data are illustrated by Fig. 6, where it will be noted that the results for the single-cylinder engine and for the four-car average are in good agreement.

Borderline ratings of substantial numbers of fuels were obtained. The comparative borderline ratings shown in Fig. 7 are of particular interest since they are for fuels of extreme compositions which normally produce the greatest deviations between road octane and Research- and Motor-method octane numbers. These results were considered as somewhat of a critical test of the single-cylinder engine's ability to duplicate automotive engine antiknock performance. It will be noted that the single-cylinder engine ratings exhibit similar trends and good numerical agreement with the automotive engine ratings.

Fig. 8 illustrates some of the comparative operating conditions [21] of the automobile engines and of the single-cylinder engine during this testing. A single or common curve means exact correspondence, the single-cylinder engine being arbitrarily controlled to duplicate the automotive values. The automobile engine data presented here are the average of the four cars, each of which, individually, was reasonably close to the four-car average.

Table 2—Test Fuel Compositions

Fuel Designation	Composition
A B C D E	Commercial gasolines
F	Straight-run gasoline plus 0.5 ml tel/gal
G	Thermally cracked gasoline
H	Catalytically cracked gasoline 75% Normal heptane 25%
I	Polymer gasoline 65% Normal heptane 35%
K	Diisopropyl 25% Alkylate 50% Normal heptane 25%
L	Cyclohexane 25% Alkylate 50% Normal heptane 25%
M	Toluene 25% Alkylate 50% Normal heptane 25%
N	Cumene 25% Alkylate 50% Normal heptane 25%

Single-Cylinder Engine Bracketing Procedure

The Borderline method was useful to indicate the similarity in trend and level of knock-limited perform-

[21] These operating conditions are shown at a specified compression ratio; hence suction density and intake air temperature are satisfactory specifications in place of compression density and compression temperature.

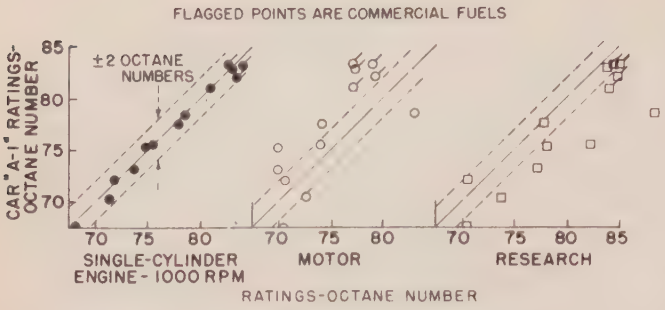


Fig. 9—Comparison of laboratory engine ratings with car A-1 Modified Uniontown-method ratings

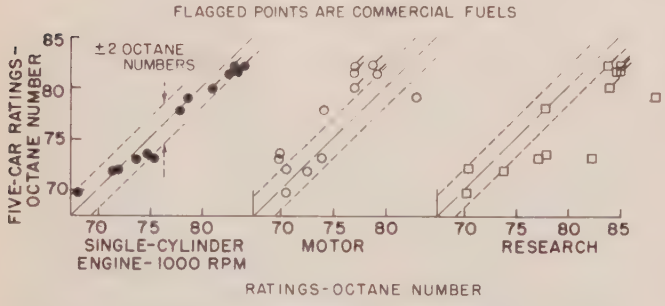


Fig. 10—Comparison of laboratory engine ratings with five-car average Uniontown-method ratings

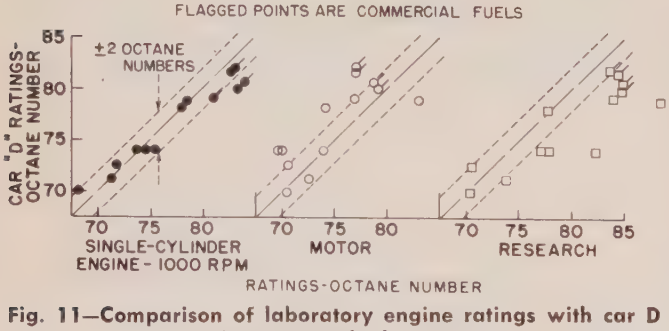


Fig. 11—Comparison of laboratory engine ratings with car D Uniontown-method ratings

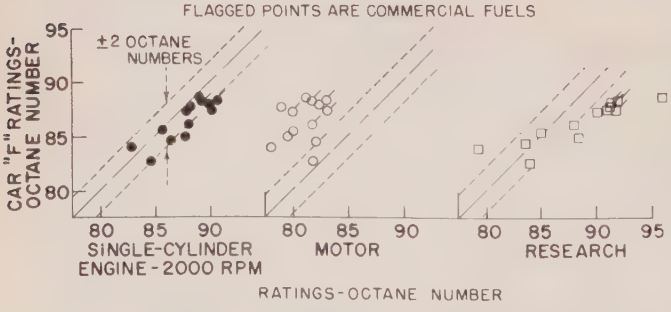


Fig. 12—Comparison of laboratory engine ratings with car F Modified Uniontown-method ratings

ance of the single-cylinder engine and automotive engines. Additional work was conducted using a bracketing procedure [22] to obtain single-cylinder engine ratings for comparison with car road octane ratings. Since the single-cylinder engine did not have continuously variable compression ratio, spark advance was used as the independent variable to induce knocking.

For a preliminary trial of the bracketing procedure, comparative ratings were obtained in the single-cylinder engine and car A-1 of the previous four-car group [23] on the fuels listed in Table 2. The car tests were conducted by the Modified Uniontown method. Knocking in car A-1 occurred predominantly at low speed. A good correlation of Modified Uniontown method road octane values from the car test was obtained with single-cylinder engine bracketing method ratings at 1000 rpm, as illustrated in Fig. 9; deviations from perfect agreement are less than 2 octane numbers. For comparison, Research- and Motor-method octane numbers also are shown plotted against the Modified Uniontown method ratings for car A-1.

To extend the automotive side of the comparison, the fuels of Table 2 were also road octane rated by the Uniontown method for a group of five cars: A-1, B, C, D, and E of Table 1. These cars predominantly showed maximum knock intensity in the lower speed region. The five-car average Uniontown method road octane ratings are compared in Fig. 10 with the 1000-rpm single-cylinder engine bracketing method ratings and also with Research- and Motor-method ratings. It will be noted that the 1000-rpm single-cylinder engine ratings agree considerably better with the five-car average Uniontown method road octane ratings than do the Research- and Motor-method ratings.

The Uniontown method road octane ratings from car D, an L-head engine type, are compared, in Fig. 11, with the 1000-rpm single-cylinder engine bracketing method ratings and with the Research- and Motor-method ratings. The comparisons are approximately the same as for the five-car average.

Extension of Octane Number Range

This point of the investigation was reached about 1949–1950, and it appeared that it would be desirable to extend the investigation above the then commercial level of octane numbers and compression ratios. As a basis for this extension, a program of comparative tests was planned to carry above 100 octane number. Car F and higher compression-ratio engines for car F were to be used for the road octane side of

[22] See Appendix for bracketing technique for fixed-compression-ratio single-cylinder engine ratings.

[23] Car and single-cylinder engine bracketing ratings are the average of four independent determinations for each fuel. The ASTM Research and Motor ratings, being derived from standardized tests, are the average of at least two check determinations.

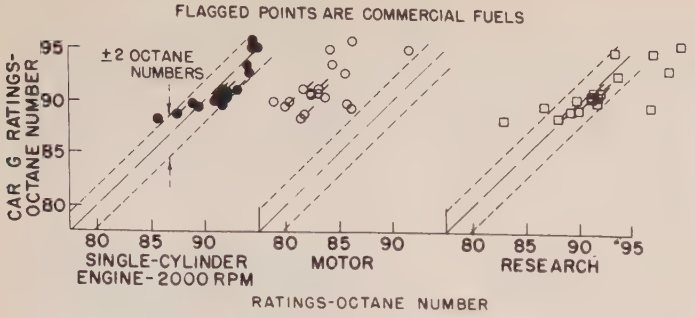


Fig. 13—Comparison of laboratory engine ratings with car G Modified Uniontown-method ratings

the comparison. The single-cylinder engine was to be used, with compression ratio varied by altering the thickness of the shim at the base of the cylinder.

The portion of this program involving car F and engine G (Table 1), at compression ratios of 7.43/1 and 8.12/1, was completed as planned. The portion of the program at still higher compression ratios was not completed, however, because of an unsatisfactory situation that arose in connection with stepwise shimming of the single-cylinder engine to adjust compression ratio. The shimming resulted in some of the fuels being rated at a low compression ratio and overly advanced spark or at a high compression ratio and overly retarded spark. These effects became particularly confusing at the higher compression ratios, and that part of the program was abandoned temporarily.

The technique described under the first unit of work on the domed combustion chamber was used to establish operating conditions of the single-cylinder engine for rating comparison with cars F and G. Similarly, the fuels tested covered a range of types and compositions. Modified Uniontown method road octane ratings were obtained in cars F and G. Both of these cars had automatic transmissions and some degree of the volumetric efficiency and spark advance practices that have tended to shift the occurrence of knocking into the mid-speed range.

Fig. 12 shows a comparison of the road octane ratings obtained on car F with 2000-rpm single-cylinder engine ratings and with Research- and Motor-method ratings. Fig. 13 shows a similar comparison for car G ratings. As before, the single-cylinder engine ratings agree much better with the road octane ratings than do the Motor- or Research-method ratings.

Fixed Compression Ratio Cylinder Results

In continuing the development program, a group of more recent model cars was used. This group consisted of nine 1949–1952 models with synchromesh and four different kinds of automatic transmissions; compression ratios ranged from 6.6 to 8.1. As in the previous work, a variety of commercial and specially

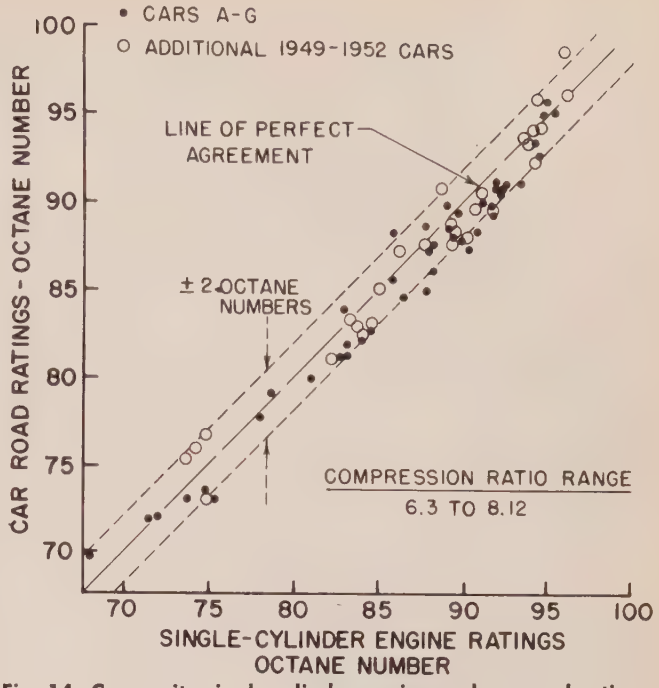


Fig. 14—Composite single-cylinder engine and car road ratings

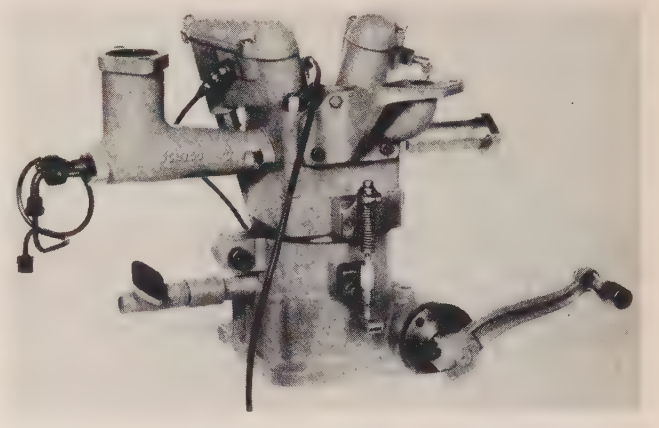


Fig. 15—Variable-compression-ratio cylinder assembly

blended fuels was used. Operating conditions were based on previous performance data. Single-cylinder engine and car road ratings from this work are shown in Fig. 14, together with those from the previous unit of work on the single-cylinder engine. This figure shows that a consistent pattern of good agreement between single-cylinder engine and car road octane ratings prevails for the several units of work covering a variety of cars.

Variable-Compression-Ratio Cylinder

Because of the difficulty previously mentioned, arrangements were made to adapt the domed combustion-chamber cylinder-head assembly for continuous variation of compression ratio, similar to that obtain-

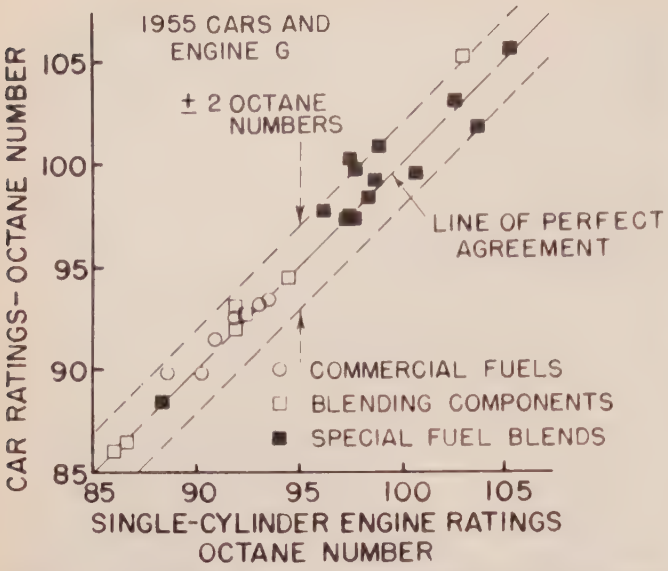


Fig. 16—Variable-compression-ratio single-cylinder engine results

able with the ASTM-CFR variable compression cylinder. The resulting construction is illustrated by Fig. 15.

With the variable-compression-ratio cylinder, spark advance is varied slightly with compression ratio to maintain approximately maximum power setting. Compression ratio is varied with fuel octane number, to maintain a uniform level of knock intensity; this operation is analogous to use of the familiar guide curves in the ASTM Research and Motor methods.

Road Octane Comparisons

With the variable-compression-ratio cylinder available, a program was conducted extending the range to 100+ octane number.[24] Once again, a variety of commercial and special fuel blends was rated. Road octane ratings were obtained using eight 1955 model cars; compression ratios up to 11 were obtained by using available optional cylinder heads and pistons, and engine G of Table 1 was used with 10 and 12 compression ratio heads. Single-cylinder engine operating conditions were selected as indicated by the earlier work.

Continuing the trend established previously, the results of this work in Fig. 16 show good agreement between single-cylinder engine and car road octane ratings.

High-Aromatic Blending Stocks

During the more recent work, there was occasion to rate several neat samples of high-aromatic refinery blending stocks. These stocks exhibit a characteristic dull thudding, erratic type of knock which makes them difficult to rate, and which has also been noted by others. This behavior was not observed with 50% blends of these stocks. It appears that knock-rating of these stocks may pose special problems of instrumentation to detect this type of knock and of investigation to determine more about its occurrence and significance. These matters are beyond the scope of the present paper.

With the trend toward higher octane automotive gasolines, there is a possibility that commercial products may approach the above blending stock characteristics. This suggests that work on these special problems might prove highly desirable.

Summary

Road octane ratings of a given fuel vary by a few octane units depending on the automobile in which they are determined. Hence, one does not expect to find a highly exact correlation between laboratory single-cylinder engine ratings and road octane ratings obtained from groups of automobiles of varying make and model. Comparisons of single-cylinder engine ratings with road octane ratings obtained from various makes of automobiles over a period of seven years, however, show two desirable characteristics:

1. The mean line of the comparisons is a line of numerical agreement.

2. Maximum differences between single-cylinder engine ratings and road octane ratings is about three octane units.

Simultaneous comparisons between ASTM Research- and/or Motor-method ratings with the same road octane ratings show:

1. The mean line of the comparisons is not the line of numerical agreement. Thus, ASTM Research-method ratings tend on the average to be higher than road octane ratings, and ASTM Motor method ratings tend to be lower than road octane ratings.

2. Differences between ASTM Research- and Motor-method ratings and road octane ratings may be as great as 10 octane units.

This improvement in comparative ratings is attributed primarily to the choice of a combustion environment for the single-cylinder engine that is more like the combustion environment of typical automobile engines than is the combustion environment of the ASTM Research and Motor methods. The essential factors of combustion environment to obtain this duplication are:

1. A compression ratio coupled with mixture temperature, density, and strength which produce mixture conditions at knock that compare to those in automobile engines.

2. A length of burning path and combustion dura-

[24] The scale used for octane numbers over 100 is: ON = 128 — 2800/PN, where PN is obtained by conversion from iso + tel.

tion and, perhaps secondarily, a surface-to-volume ratio of the combustion chamber which compare to automobile engine practices.

It is implicit in this approach to the problem that changes may have to be made in the single-cylinder engine to maintain a combustion environment that corresponds to that of automotive engines. Evidently, however, the requirement for change will not be drastic or frequent. Thus, in this investigation, the initial selection of combustion environment was made for automobiles of 1939–1942 vintage, and the only really significant change in conditions was to change the single-cylinder engine operating procedure to emphasize ratings at 2000 rpm as suggested by the performance characteristics of more recent model cars. From this cumulative experience, it appears that fine details of automotive engine operation do not require duplication in the single-cylinder engine; rather only a few essentials that have a major effect on fuel ratings need to be approximated.

It appears that a single-cylinder engine knock test method that agrees with road octane values better than the present ASTM Research and Motor methods would be of considerable utility to the petroleum and automotive industries in their joint efforts to make most advantageous developments of fuels and engines. It is hoped that the experiences presented in this report may contribute to the eventual cooperative development of such improved methods.

APPENDIX

Bracketing Technique for Fixed Compression Ratio Single-Cylinder Engine Ratings

The engine is operated on the test fuel, and spark is adjusted until a knockmeter reading of 50 is obtained. Without further adjustment of any engine variables except to compensate for fuel gravity differences in order to maintain 0.080 fuel/air ratio, the engine is switched from operation on the test fuel to a reference fuel sufficiently higher in octane value than the test fuel to produce a knockmeter reading of about 20–25. Following this the engine is operated on a reference fuel of sufficiently lower octane number than the test fuel to produce a knockmeter reading of about 75–80. The octane rating of the test fuel is then determined by linear interpolation between the octane numbers and knockmeter readings.

DISCUSSION

Discusses Fuel Ratings and Engine Octane Requirements

—J. D. Caplan
General Motors Corp.

WITH regard to the changes made by the authors in order to obtain a more meaningful laboratory rating, let me state that their attempts to make the change by duplicating in the single-cylinder engine the combustion environment that exists in a multi-cylinder engine is certainly sound. We approached the problem in a similar manner some years ago by duplicating in single-cylinder engines the combustion environment in both Chevrolet and Oldsmobile engines. In Fig. A are shown comparisons of modified Borderline ratings of the same fuel in both single-cylinder engines and test cars. These data indicate the degree of duplication of fuel ratings that can be obtained by this approach. I would like to point out that in principal this duplication of ratings appears to be an easy task. However, from our experience we know that it is not, and the authors are to be commended for the excellent results they have obtained.

There are three questions I would like to ask about the Texas Co.'s engine:

1. At what speed was the single-cylinder engine run to duplicate 1955 car fuel ratings?

2. How can you maintain "squish" or hold it constant if the compression ratio is varied?

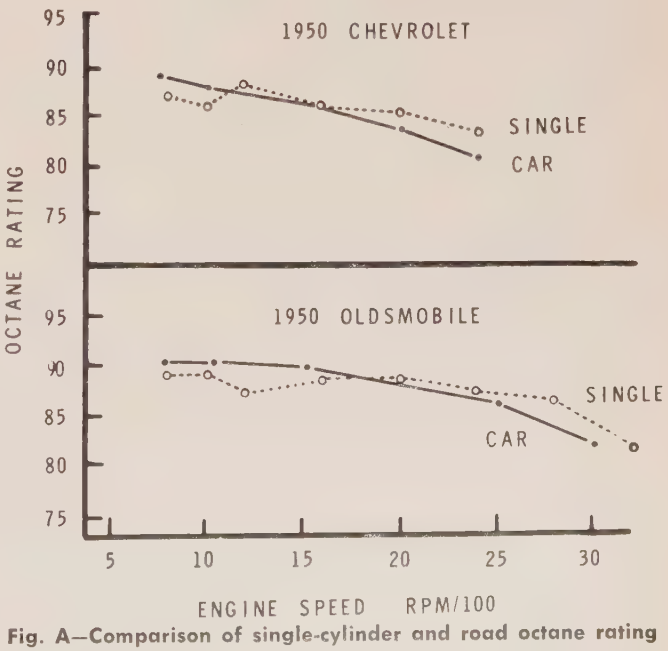


Fig. A—Comparison of single-cylinder and road octane rating

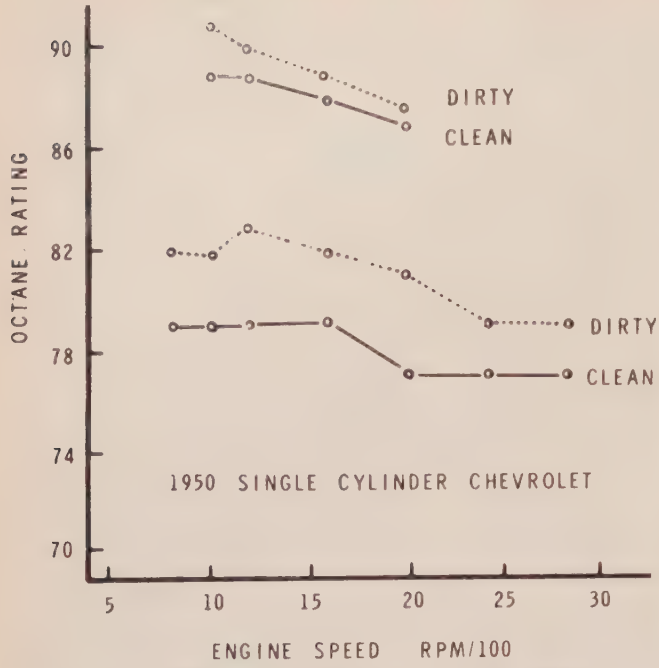


Fig. B—Effect of lead deposits on octane rating

3. Did the valve timing on the single-cylinder engine duplicate commercial practice?

Finally, I would like to point out that the "dull thudding" or "rumble" observed with high-aromatic fuels in the single-cylinder engine is also observed with similar fuels in experimental high-compression-ratio engines.

There is one general problem that should be considered. All our fuel rating work both in the laboratory and on the road is done in either essentially clean engine or in engines with heavy duty deposits due to the severity of our rating techniques. In Fig. B the ratings of two fuels are compared in a clean engine and under conditions where special efforts were made to retain light duty deposits. You will note that with both fuels the ratings were higher with the light duty deposits present. We know that these light duty deposits increase the engine octane requirement; however, we do not know if they increase the ratings of all fuels uniformly. This effect of deposits also makes it difficult to predict from the clean rating of fuels their ability to satisfy engine octane requirements determined with light duty deposits present.

The considerable attention being given to Motor numbers these days is most encouraging. Practical evidence of the need for such attention on the part of both petroleum and automotive industries is shown in Fig. C. In this figure are shown the knock-limited spark advances for three fuels in one of our 1956 engines. Note that two of the fuels have equal Research ratings and two of the fuels have nearly equal Motor ratings. All three of these fuels are commercial products of major oil companies. Note that the manufacturer's distributor spark advance curve is also shown on this figure.

Turning our attention first to the low-speed end, we see indeed that the engine does like Research numbers since the two 96 Research gasolines are knock-free and the 92 Research gasoline will knock. This type of performance was most certainly typical of engines up to five years ago where knock was a problem at low speed. In those days automatic transmissions were not so prevalent, and it was possible to operate the engine at full throttle down to speeds of 800 rpm. Thus the conception of Research octane numbers being what was required naturally arose.

Looking now at the high-speed end, we obtain a different answer. Here we observe that one of the 96 Research gasolines knocks and that the other is knock-free. We also can observe that the 96 Research gasoline with the low Motor rating is the one that knocks. In fact it does not perform as well as the 92 Research gasoline. You will notice in the curve for the 96 R/83 M gasoline that this fuel is satisfactory up to an engine speed of about 2400 rpm. To get up to engine speeds of 4000 rpm in high gear, we would be running at speeds in excess of 100 mph. You might thus look at these curves and say this high-speed end looks most interesting, but I generally never get above 40 to 50 mph so what do I care about Motor octane number? With our 1956 engines this is no longer the case. To take advantage of the increased high-speed torque of our engines, the transmissions are designed so that the engines, even in normal city driving, often get up to speeds in excess of 4000 rpm. As an example, one of our 1956 engine-transmission combinations during an acceleration, with the throttle at the detent position, does not make the 1–2 shift until about 24 mph which corresponds to an engine speed of 3500 rpm. The 2–3 shift does not occur until 40 mph which corresponds to 4000 rpm, and the final 3–4 shift does not occur until 69 mph where the engine is also at 4000 rpm.

This questioning of the engine indicates that it has an appetite for both Research and Motor octane numbers and that we must be concerned with the Motor rating of fuels and Motor octane requirements of engines not only in the future, but today.

Improved Testing Methods Difficult to Develop

—H. E. Hesselberg
Ethyl Corp.

WE agree with the authors that it is possible to operate single-cylinder laboratory engines in such a manner that fuel ratings thus obtained show a higher degree of correlation with those obtained on the road than do those obtained by either the Research or Motor methods considered alone. Our experience would indicate, however, that the development of a generally applicable single laboratory knock-rating technique to predict adequately the performance of a wide range of fuel compositions in the variety of engines under the wide range of operating conditions at which they knock on the road would be quite difficult. It would furthermore involve relatively frequent changes in both engine and operating variables as automotive engine and transmission designs progress.

We would like to second the motion of the Texas Co. investigators regarding the need for improved methods of rating highly aromatic blending stocks. In common with many laboratories we have experienced difficulty in obtaining reproducible ratings of such stocks and seriously question not only the numbers that are obtained but also their meaningfulness in terms of our normal concepts of knock and antiknock quality.

Knocking Conditions Need Further Research

—F. C. Burk
Atlantic Refining Co.

THE authors have presented one way of improving the relationship between laboratory octane numbers and those obtained on the road. Their data clearly show that the present ASTM knock test methods are not as satisfactory for predicting road performance as would be the case if a cylinder and combustion-chamber design similar to the one described was adopted as an industry standard. This should be a desirable objective for the ASTM Research Division I of Committee D-2, which has the responsibility for improving on the knock test methods. In fact, a Section in RD-I is giving serious consideration to such changes at the present time.

Also under consideration by this Section is another approach to the problem, already mentioned by the authors, and that is the development and application of basic concepts of combustion involving time, temperature, and density. Briefly, this concept theorizes that the combustion environment existing in any engine

can be duplicated in any other engine regardless of its size or shape by varying the operating conditions of the control engine to match those of the engine in question. For example, if a multicylinder engine knocks on a given fuel under certain operating conditions, and if the combustion environment which exists in this engine when it is knocking can be duplicated in a laboratory single-cylinder engine, the same fuel will also knock to the same degree in the laboratory engine. This concept appears to be sound technically, and a fair amount of experimental work has already substantiated it. Unfortunately, we do not yet have the tools or instruments to measure exactly the time, temperature, or density conditions in the combustion chamber when knocking occurs. All of the experimental work has been carried out on a trial and error basis along with mathematical calculations for compression density and temperatures. Obviously these calculations are not true end-gas densities or temperatures; hence the degree of error entering this approach is not truly known. The fact remains, however, that the limited data which have been obtained appear to support the concept.

From the above it may be concluded that two approaches show promise in better relating laboratory engine octane determinations with the road. One, by improving on the design of the combustion chamber described by the authors and, two, by a better understanding and application of the time-temperature-density concept. The importance of antiknock quality to both the petroleum and automotive industries indicates that both of these approaches should be given increased emphasis to achieve the gains which may result therefrom.

Two Different Test Procedures Are More Satisfactory Than One

—E. J. McLaughlin
Calif. Research Corp.

THE Texas Co. has done an excellent job in developing a test engine which approximates the combustion-chamber environment of automobile engines more closely than either the F_1 or F_2 engines. In fact, this paper has answered one of the questions which has been under consideration in various cooperative research committees for some time; namely, is it possible to achieve satisfactory combustion-chamber environment with the present CFR cylinder? The very comprehensive investigation conducted by E. M. Barber and his associates indicates that a new cylinder will be required and proposes a design which appears to be well suited for the development of future knock testing methods.

If new laboratory knock test methods are to be developed, we would favor a somewhat different approach than that used by the Texas Co. investigators. It has been our experience that it is generally more satisfactory to have two laboratory test procedures (preferably using the same basic engine) which differ widely in the severity of their operating conditions. If these test procedures are properly worked out, they will provide two ratings which represent the best and poorest performance of the test fuel and also provide a measure of the fuel sensitivity. The use of two test procedures greatly minimizes the amount of test-method modification that is required as new production engines having different degrees of severity become available, because the correlation with the road is not one rating but is rather the appropriate interpolation between the two extreme laboratory values. For these reasons, we would prefer using the very excellent test engine and data provided for the development of two test procedures which will bracket road ratings rather than one test procedure designed to match the road ratings.

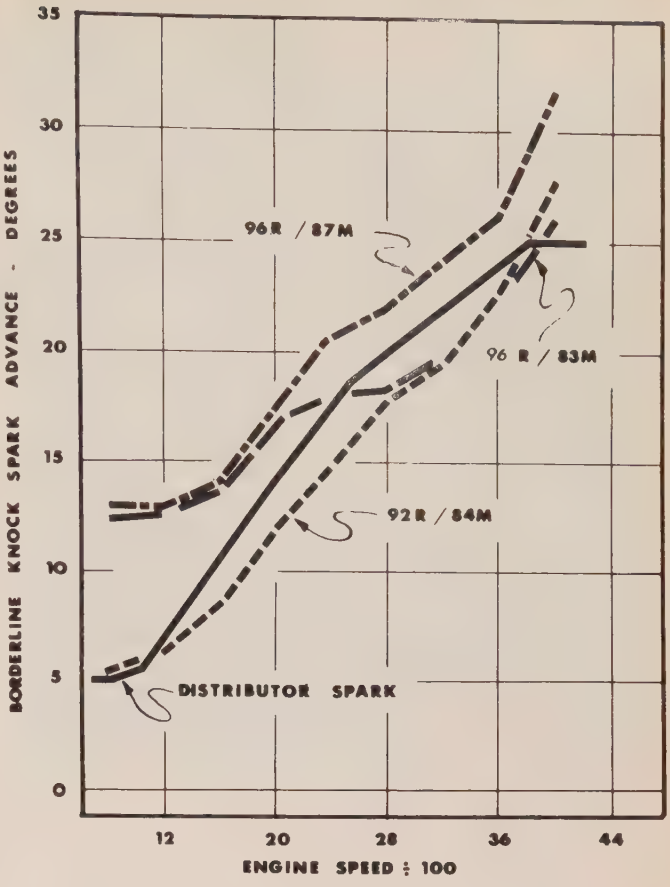
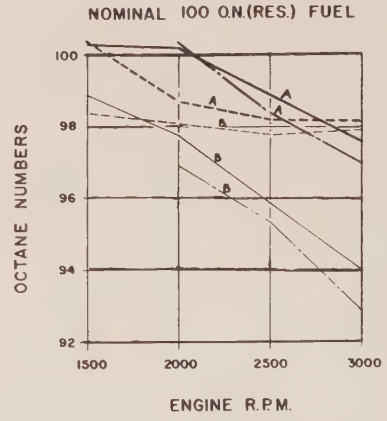


Fig. C—Knock-limited spark advances for three fuels



—— CAR 'A', 10.0:1 CR.
---- CAR 'C', 9.25:1 CR.
—·— CAR 'E', 9.0:1 CR.

A 35% ISOMERATE PLUS CAT. REFORMATE
B MIXTURE OF F.C.C. & CAT. REFORMATE

Fig. D—Road octane values of experimental blends

Need Better Laboratory Method Of Road Octane Determinations

—F. T. Finnigan
Pure Oil Co.

EIGHT to ten years ago it was not uncommon to encounter a 10 octane spread when rating a single fuel in engines of different basic designs. At the 75–85 octane level, this was equivalent to approximately 13 performance numbers. In our modern V-8 engines at the 95–100 octane level, this rating spread has been reduced to approximately five octane numbers. Fig. D illustrates such a situation. Three vehicles, ranging in compression ratio from 9 to 10/1, rate these two fuels quite differently. The blend of fluid catalytically cracked gasoline shows quite a spread in road response among the vehicles as speed increases. In terms of per-

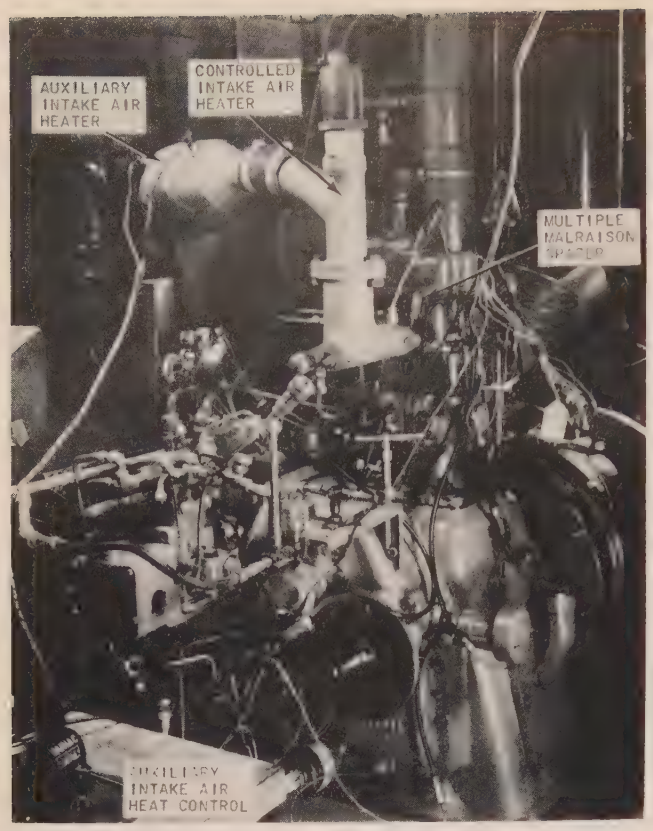


Fig. E—Instrumentation used for single-cylinder fuel rating investigations

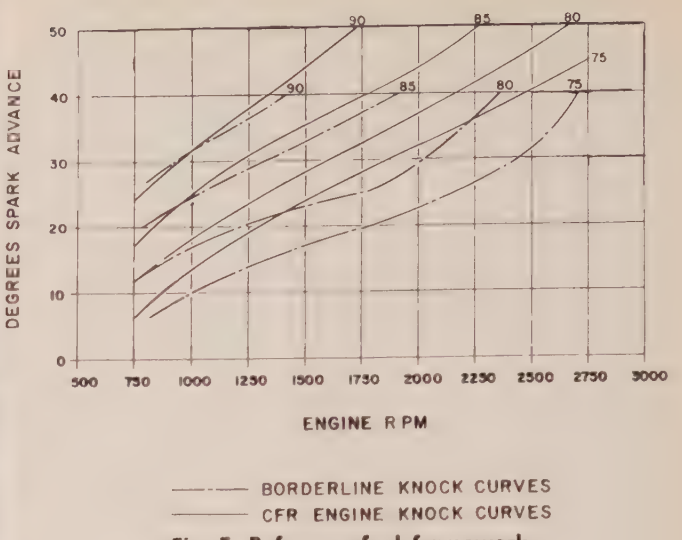


Fig. F—Reference fuel framework

Table A—Comparison of Single-Cylinder Ratings and Average Multicylinder Ratings

Rpm	Single	Multi	Rpm	Single	Multi
	Fuel A			Fuel D	
1000	75.5	75	1000	85.0	85.5
1500	75.5	75	1500	85.5	85.5
2000	75.0	74.5	2000	85.5	86.5
2500	74.0	74.0	2500	84.5	86.0
	Fuel B			Fuel E	
1000	74.0	74.0	1000	83.5	83.5
1500	74.5	74.0	1500	84.5	84.5
2000	74.5	73.0	2000	85.0	84.5
2500	73.0	72.5	2500	——	82.5
	Fuel C			54% DIB	
1000	74.5	74.5	1000	85.0	86.0
1500	74.0	73.0	1500	83.5	84.7
2000	73.5	72.5	2000	82.5	83.8
2500	72.0	71.0	2500	81.0	81.4

formance, this is equivalent to approximately 15 performance numbers and is actually greater than the spread of 10 years ago when considered in those terms. The isomerate-reformate blend, on the other hand, appreciably narrows the spread in road ratings among the test vehicles.

From 1946 to 1950 our research laboratories carried on an extensive single-cylinder fuel rating investigation. Fig. E is a photograph of the installation and gives an idea of the extent of the instrumentation used. Fig. F is an illustration similar to Fig. 2 showing the similarity existing between the single and multicylinder reference fuel frameworks. Table A shows the correlation existing between a series of road ratings in a group of valve-in-head, multicylinder engines and the single-cylinder engine. Similar correlation was possible with L-head engines; however, changes in the single-cylinder method were required in order to obtain it.

With the maximum level of knock in today's vehicles occurring over a range of from 1300 to 3000 engine rpm, ratings from a laboratory engine could not be expected to define individual engine performance. However, as indicated, the number obtained could be a more accurate reflection of the fuel's road performance than the currently used Motor or Research octane ratings.

Author's Closure To Discussion

THE complimentary remarks of the discussors are appreciated. It is indeed gratifying to note that the information cited tends to confirm our results. Everyone apparently agrees that continuing research on this problem is needed.

The single-cylinder engine speed used for correlation with 1955 car ratings, requested by Mr. Caplan, is 2000 rpm. We provide only a moderate "squish" and make no attempt to either control it or hold it constant as compression ratio is varied. The single-cylinder engine valve timing and valve opening are essentially that of the ASTM supercharge engine. The standard camshaft ramps are reground to accommodate hydraulic, rather than the standard mechanical, valve lifters.

As representatives of the petroleum industry, we are naturally concerned with production and evaluation of gasoline for engines with critical octane requirements at and about 4000 rpm. We cannot take issue with the accuracy of Mr. Caplan's statement of the acceleration conditions that can occur in city driving. As private citizens, we are thankful to live in a community where such driving is not normal.

Mr. Burk discusses investigating the problem using two approaches which seem to separate operating conditions from combustion-chamber shape; it is our opinion that these approaches will not be fruitful if pursued independently since both are a part of the combustion environment. Hence, our basic approach was to match the combustion environment (as best we could) and see what ratings resulted.

Mr. McLaughlin must be psychic. Actually, we have obtained ratings both at two different speeds, and at two different intake air temperatures. While work along these lines is continuing, it can be mentioned that the 2-temperature scheme looks very promising.

The interest shown in this type of work is very encouraging and promises an early and successful solution to our present problems.

COMPRESSOR STALL PROBLEMS IN GAS-TURBINE-TYPE AIRCRAFT ENGINES

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EARLY models of aircraft turbojet engines occasionally encountered compressor stall or surge during very rapid acceleration. As engine development progressed, high-pressure-ratio axial-flow compressors were utilized to improve the operational ceiling and the fuel economy of this type of engine. These increases in compressor pressure ratio greatly aggravated the compressor stall problems and necessitated the use of variable-geometry types of fixes to ensure satisfactory engine acceleration.

Recent emphasis on high-altitude flight has shown severe altitude effects on the compressor stall characteristics. These adverse effects of altitude on compressor stall limits are a result of reduced Reynolds number and present a serious limit to the altitude ceiling of a turbojet engine. Increases in flight Mach numbers have placed increased emphasis on the adverse effects of distortions of flow distribution at the compressor inlet.[1] The adverse effect of these inlet-flow distortions on compressor stall limits is also a serious restriction on the operational limits of the aircraft.

At low values of rotative speed, a compressor stall phenomenon defined as rotating stall has been observed. Recent research has shown that the unsteady flow which exists during rotating stall is a prevalent source of destructive blade vibrations in axial-flow compressors.

The objectives of this paper are to discuss the compressor stall phenomenon and to show how compressor stall limits the operation of turbojet engines for aircraft. This discussion will be restricted to axial-flow-type compressors. Stall problems for both single- and 2-spool turbojet engines will be considered. Potential solutions to the compressor stall problems with regard to engine fixes and engine controls have been discussed elsewhere.[2,3] The significance of these problems with respect to the customers has been also discussed.[4]

Compressor Performance

Dimensional analysis has shown that the pressure

1 "Inlet-Duct Engine-Flow Compatibility," by J. S. Alford. Paper presented at Fifth International Aeronautical Conference, June, 1955.
2 "Turbine-Engine Stall from Design Point of View," by J. S. Alford. Paper presented at SAE National Aeronautic Meeting, April 12, 1956.
3 "Control Approaches to Solution of Compressor Stall," by W. A. Gebhardt and R. Bodemuller. Paper presented at SAE National Aeronautic Meeting, April 12, 1956.
4 "A Customer's View on Turbine-Engine Stall," by Major K. L. Jones. Paper presented at SAE National Aeronautic Meeting, April 12, 1956.

STALL of axial-flow-type compressors limits the flexibility of operation of 1- and 2-spool aircraft turbojet engines. Complete compressor stall or surge is a problem with regard to engine acceleration as well as low- and high-Mach-number flight at high altitudes.

Deterioration of compressor performance due to low Reynolds numbers at extreme altitudes aggravates the stall problem. Inlet-flow distortions increase the severity of this problem, particularly for supersonic flight.

Rotating stall at low values of equivalent rotative speed is a serious source of blade vibrational failure. This latter problem is also aggravated by inlet-flow distortions.

ratio and efficiency of a compressor are determined by the following six parameters:

1. Equivalent rotative speed.
2. Equivalent weight flow.
3. Reynolds number.
4. Ratio of specific heats.
5. Prandtl number.
6. Froude number.

Of these parameters, only the first three are significant for compressors of aircraft jet engines. The equivalent speed is a measure of the tangential Mach number of the compressor blades; the equivalent flow is a measure of the axial Mach number of the air passing through the compressor; and the Reynolds number is representative of the ratio of inertia forces to viscous forces in the compressor. Inasmuch as the ratio of specific heats for air is essentially constant for the range of temperatures of interest, the fourth parameter can be neglected. Parameter five, the Prandtl number, is an index of heat-transfer effects which are considered negligible for aircraft compressors. The sixth parameter, the Froude number, is the ratio of inertia forces to gravitational forces and is also negligible.

The derivation of these performance parameters is based on the assumption of uniform flow conditions at the compressor inlet. In aircraft application, however, uniform flow distribution at the compressor inlet is frequently not achieved, and an additional parameter, inlet-flow distribution, must also be considered. Thus, the compressor performance as measured by pressure ratio and efficiency can be determined as functions of equivalent rotative speed, equivalent weight flow, Reynolds number, and inlet-flow distribution.

Conventional Performance Maps—Compressor performance is normally evaluated at inlet pressures approaching those for sea level, and the value of Reynolds number is sufficiently large so that moderate variations of inlet pressure are not significant. Furthermore, uniform inlet-flow distributions are also normally maintained. For these conditions, the compressor pressure ratio and efficiency are only functions of equivalent rotative speed and equivalent weight flow. The equivalent weight-flow parameter is taken as $W\sqrt{\theta}/\delta$

where:

W = Flow rate, lb per sec
θ = Ratio of inlet temperature to standard sea-level temperature, deg Rankine
δ = Ratio of inlet total pressure to standard sea-level pressure

The equivalent rotative speed is $N/\sqrt{\theta}$

where:

N = Compressor rotative speed

The equivalent speed is generally given in per cent of the rated or design value. Within certain limits, fixed values of equivalent speed and equivalent flow represent a fixed geometry of flow and fixed Mach numbers of flow relative to any compressor blade row regardless of inlet pressure and inlet temperature.

A typical compressor map based on these parameters is shown in Fig. 1. In this plot, the total-pressure ratio across the compressor is plotted against equivalent weight flow for constant values of equivalent speed as shown by the solid lines. The compressor efficiency can be plotted as contours of constant efficiency as shown by the dotted lines on Fig. 1, or it can be presented as a separate plot against either equivalent weight flow or total-pressure ratio with equivalent speed as a parameter. Fig. 1 represents the compressor preformance for standard sea-level inlet conditions. For moderate changes in inlet temperature and pressure, the compressor operating

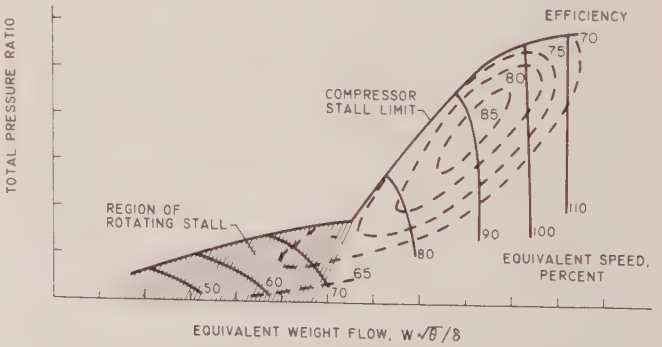

Fig. 1—Conventional compressor performance map

condition is obtained by applying the appropriate values of ϑ and δ.

Two compressor stall phenomena are also shown on the compressor map of Fig. 1. The low speed-region in which rotating stall originates in the front stages is indicated by the shaded area. This shaded area could be referred to as a region of "partial" stall. The compressor stall limit or maximum pressure ratio obtainable at any value of equivalent flow is indicated by the heavy dot-dashed line. This compressor stall limit is usually the minimum usable flow at any given speed. Under certain circumstances, surge may occur at the compressor stall limit line. This line is, therefore, often referred to as the surge line.

The weight-flow variations with speed which are indicated on Fig. 1 are typical of most axial-flow compressors. At high values of equivalent speed, the flow range is small, and the performance characteristic is nearly vertical. As speed is decreased, the flow range at a given speed increases, and at low speeds the performance characteristics become more nearly horizontal. The island of peak efficiency, as indicated by the contour of maximum efficiency, generally occurs at slightly less than design speed and is located relatively close to the compressor stall limit.

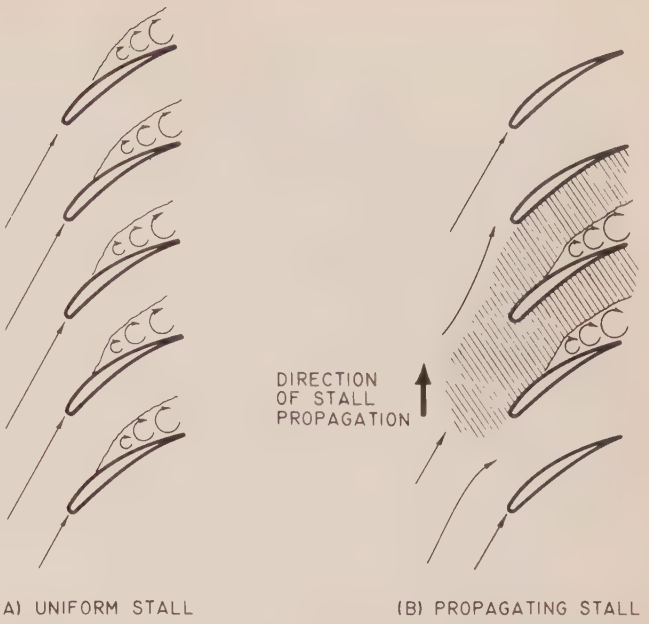

Fig. 2—Cascade stall of a single-blade row for uniform stall and propagating stall

Compressor Stall Phenomena

As has been pointed out on the multistage-compressor performance map of Fig. 1, two principal stall phenomena have been encountered in the compressor: inlet stage or rotating stall and compressor stall. In addition, a phenomenon called surge is often encountered at the compressor stall limit. It is the purpose of the following discussion to define and explain the origin and differences among:

1. Rotating stall.
2. Compressor stall.
3. Surge.

Rotating Stall—The phenomenon of rotating stall has received a considerable amount of research emphasis in recent years [5,6] because it has been found to be a major cause of compressor blade vibrations and engine failures. This section will present a simplified description of the mechanism of rotating stall, the operating range in which it occurs, and its characteristics.

Single-blade rows: In order to understand the stall problems in a multistage axial-flow compressor the stall of a single-blade row should be considered. When the flow through a compressor blade row is reduced, the angle of attack on the blading is increased. If the angle of attack is increased too much, separation will occur on the suction surface of the blading. The occurrence of flow separation is accompanied by a large increase in loss and is referred to as stall of the blading.

At high angle of attack, it might be expected that all blades in a blade row would stall uniformly as indicated in Fig. 2a. This figure depicts an end-view of a blade row as it would appear if unwrapped from the compressor circumference. In general, however, uniform stall of all blades of such a cascade is an unstable condition. Normal flow instabilities will cause a small group of blades to stall before the others in the blade row. Separation occurs on the suction surface of the blades in this group as indicated in Fig. 2b. This flow separation results in a severe flow restriction in these blade passages. Furthermore, this group of blades is no longer capable of supporting the pressure rise imposed by adjacent unstalled blades and produces a reverse flow through the stalled blade passages. As a result of these factors, a partly stagnated region is formed ahead of the stalled blade or blades. This region is indicated by the shaded region of Fig. 2b. The incoming air is deflected around this low flow region as indicated by the two arrows. The angle of attack is increased on the blades immediately above the stalled blades, and the angle of attack on the blades below the stall region is decreased. Thus, the next blade above the stalled region stalls, and the last blade at the bottom of the stall zone unstalls. This process is continuous, and the stall zone propagates upward for the blade orientation of Fig. 2b. For a straight cascade of blades, therefore, a propagating

5 "Compressor Surge and Stall Propagation," by H. W. Emmons, C. E. Pearson, and H. P. Grant. Paper presented at ASME Meeting, 1953.

6 *Journal of Aeronautical Sciences*, Vol. 20, December, 1953, pp. 835–845: "Some Stall and Surge Phenomena in Axial-Flow Compressors," by M. C. Huppert and W. A. Benser.

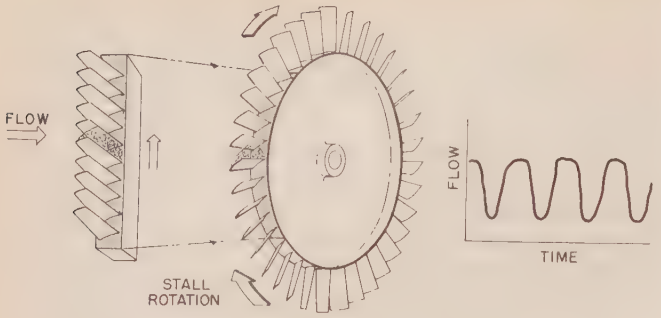

Fig. 3—Rotating-stall propagation in rotor or stator. Plot of flow against time shows flow variations as measured on fixed instrument as stall zones pass

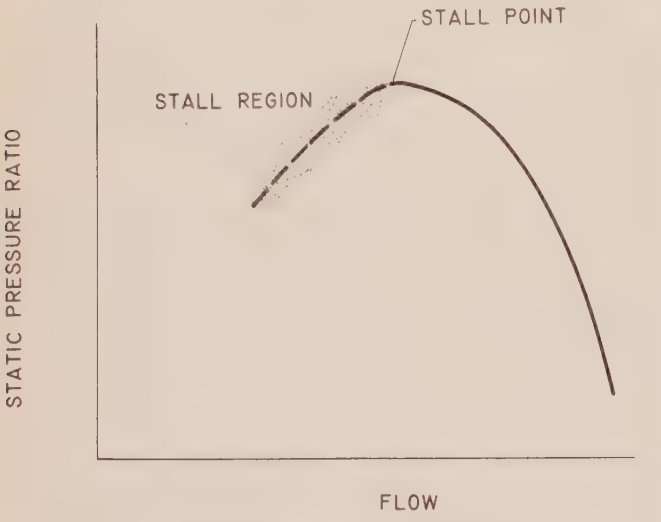

Fig. 4—Performance of a single-stage compressor

type of stall is obtained. Experimental evidence of this propagating stall is given in literature.[5]

When this cascade of blades is wrapped around a compressor rotor, as indicated in Fig. 3, the stall zone propagates around the compressor axis. This phenomenon is defined as rotating stall. The plot of flow against time on the right-hand side of this figure shows the flow variations that would be measured on a fixed instrument as the stall zones pass. The simplified picture of the mechanism of rotating stall applies to either rotating or stationary blade rows so rotating stall can originate in stator blade rows as well as rotor blade rows.

The flow range over which rotating stall usually occurs in the single-stage compressor is indicated in Fig. 4. This figure presents typical single-stage performance in terms of the static-pressure ratio varying with equivalent flow at a given equivalent speed. Reduction in flow results in an increase in angle of attack and an increase in static-pressure ratio which eventually leads to stall of the blade elements. As a result, the static-pressure ratio reaches a peak and then decreases with further reduction in flow. Stage performance becomes unstable at the peak of the curve, and rotating stall is initiated at this peak value. Rotating stall then exists at all flows below that for peak static-pressure ratio.

A wide variety of stall patterns may exist for single-stage compressors. Whereas theories have been developed that approximate the propagational rate of the stall zones for isolated blade rows, no theory is currently available to predict the number of stall zones that will exist. There may be one or more stall zones around the axis of the compressor. As many as eight zones have been observed. In a given compressor, the number of stall zones usually varies with flow. These stall zones may extend over the entire span of the blade or only over a part of the blade span. They are generally equally spaced and of approximately equal size, but variations from this uniformity have been observed. The propagation rate varies appreciably, but is nearly always in the direction of rotor rotation. Variations of stall patterns and their effects on stage performance are discussed elsewhere.[6] For instance, the abrupt or sudden stall of a large portion of the radial span of a blade row usually leads to an abrupt and discontinuous drop in pressure rise at the stall point. In contrast, progressive stall in which the stall zones grow gradually in circumferential and radial extent results in the continuous type of performance shown in Fig. 4. The abrupt type of stall is usually obtained in blade rows with a high hub-tip ratio such as exit stages in the multi-stage unit, whereas progressive stall is usually encountered in the ratio-inlet stages with low hub-tip ratio.

Multistage compressor: A multistage axial-flow compressor consists of numerous blade rows operating in series. The stalling characteristics and the range of stall-free operation of the individual blade rows varies appreciably. Furthermore, the variation of angle of attack for the individual blade rows is also a function of the speed of the compressor and the location of the blade row in the compressor. For example, at high values of rotative speed, the range of angle of attack imposed on the front stages is extremely small as indicated by the nearly vertical constant-speed lines on Fig. 1. As weight flow is decreased or compressor pressure ratio increased at this speed, however, the angle of attack on the rear stages varies from that for choked flow to that for stall. For low-speed operation of the compressor, the angle of attack on the inlet stages approaches the stall value, whereas the angle of attack for the rear stages approaches the choked-flow condition.

These variations in stage operating angles of attack are a result of compressibility, the fixed-area convergence through the compressor, and the fixed blade angles. At design speed, the air density in the rear stages is much greater than that in the front

stages. Therefore, the annular flow area for successive stages is decreased as indicated in Fig. 5, so as to give the desired variation in axial velocity through the compressor. Fig. 5 presents a longitudinal view through a multistage compressor in which the area variation through the compressor was set so that the inlet and exit stages would operate at the design points circled on the stage performance curves at the bottom of the figure. The inlet- and exit-stage performance is presented in terms of an equivalent pressure ratio, which within certain compressibility limits may be considered to be independent of speed, varying the ratio of flow to speed. At low values of rotative speed, the pressure ratio and the density ratio across the compressor are much less than for design rotative speed. As a result, the flow area of the rear stages of the compressor is too small, and these rear stages choke as indicated by the arrow on the exit-stage curve.

The flow restriction imposed by the rear stages at speeds below design speed causes the front stages to approach stall as indicated by the arrow on the inlet-stage curve. At sufficiently low speeds, inlet-stage stall is encountered, and a rotating stall results. This region of operation is indicated by the shaded area on Fig. 1. The reverse trend occurs as the speed is increased above the design speed. At high speed, the pressure rise through the compressor is greater than the design value so that the exit stages move toward stall while the inlet stages choke. Increase in overall compressor design pressure ratio results in greater overall variations of flow area through the compressor and also greater variations of density ratio with speed. As a result, the inlet-stage stall generally extends to higher speeds for high-pressure-ratio compressors than for low-pressure-ratio units. The previous discussion indicates the mechanism by which rotating stall develops at low and intermediate speeds, as a result of inlet-stage stall and also indicates the possibility that rotating stall may be observed at high speeds because of stall of the exit stages.

The rotating-stall phenomenon at low speeds has been observed in practically all multistage axial-flow compressors in which measurements of unsteady flow have been made. The magnitude of the low-speed rotating-stall flow fluctuations generally increases through the first few stages and then decreases rapidly towards the rear of the compressor. However, definite flow fluctuations extend throughout the entire compressor.

A wide variety of rotating-stall patterns have been observed. The stall pattern may consist of from one to eight or more low flow zones around the compressor circumference. The number of zones depends on the operating condition of the compressor as well as on the compressor and engine design. In some compressors, several numbers of stall zones have been observed intermittently at a fixed operating point. These stall zones are generally concentrated at the tip of the compressor blades. However, in some designs the stall zones are concentrated at the hub, and in rare instances stall zones at both the hub and tip have been observed simultaneously. The stall zones are usually equally spaced around the circumference, but exceptions have been observed. The propagation rate of the stall zones varies from about 45 to 55% of rotor speed, and propagation is in the direction of rotor rotation. No theories have been developed for predicting rotating-stall characteristics for multistage compressors.

The existence of rotating stall does not necessarily result in large decreases in compressor performance. The low efficiencies normally observed at low values of rotative speed can generally be attributed to depreciation of stage performance due to off-design operation and stage mismatching rather than to the effects of the unsteady flow imposed by rotating stall. Although the local flow is unsteady when rotating stall exists, the net flow through the compressor is steady, and stable operation is obtained.

The local flow fluctuations, however, are an extremely serious operating problem with regard to fatigue failures of compressor blades. The local fluctuations due to rotating stall impose periodic force variations on the compressor blades. The frequency of these force fluctuations is frequently of the same order of magnitude as the natural frequency of the blades of one of the compressor stages. When resonance of these two frequencies occurs, severe blade vibrations result. Experiment has shown that rotating-stall patterns have sufficiently strong harmonics to incite serious blade vibrations even when the stall frequency is one-half or one-third of the natural frequency of the compressor blades;

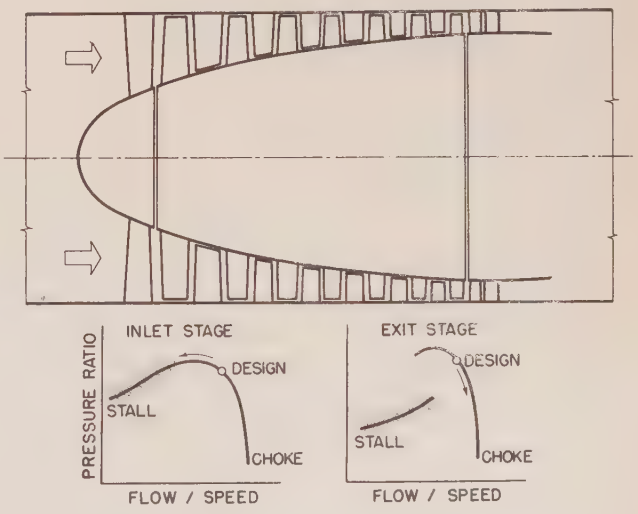

Fig. 5—Longitudinal view through multistage compressor. Inlet- and exit-stage performance is presented in terms of equivalent pressure ratio on graphs

it is believed that rotating stall is the major source of blade failure due to vibrations. Of course, resonance with pressure fields due to structural members such as inlet struts remains as a source of blade vibration. In addition, self-excited blade vibrations due to a phenomenon such as stalling flutter are also possible.

Compressor Stall—In addition to rotating stall due to inlet-stage stall at low equivalent speed, another phenomenon called complete compressor stall is encountered at all speeds. This is also a form of propagating stall and is due to excessive angle of attack on one or more blade rows. These two phenomena are, however, appreciably different in their effects on overall compressor performance. The effect of rotating stall on compressor performance is generally not of such a magnitude as to render the engine inoperable. On the other hand, complete compressor stall represents the minimum flow consistent with usable compressor operation at any given equivalent speed. Complete compressor stall is characterized by a sharp decrease in pressure ratio as shown in Fig. 6. This figure shows the pressure ratio characteristics for a low and a high value of rotative speed. A large decrease in efficiency occurs along with the drop in pressure ratio shown in Fig. 6.

At high values of rotative speed, compressor stall is initiated by abrupt stall of the rear stages. As speed is decreased, complete compressor stall is instigated by abrupt stall of successively earlier stages. At very low speeds, compressor stall is probably instigated by the middle stages. The locus of the compressor stall points for all speeds comprises the compressor stall limit line.

The flow required to unstall the compressor is somewhat higher than that at which stall is initially encountered. Thus, unstalling of the compressor exhibits a definite hysteresis effect. This effect is quite large at low speeds and is quite small at high speeds as indicated in Fig. 6. Therefore, unstalling, particularly at low engine speeds, requires a large reduction in fuel flow.

The complete compressor stall phenomenon is also characterized by a low flow or stalled zone which rotates about the compressor axis. This stall zone is a region of practically zero flow or perhaps even reverse flow which generally extends over the entire span of the blade and covers one-third or more of the compressor circumference. Furthermore, the large deficiency of velocity in the stall zone is maintained throughout the entire length of the compressor. The stall zone rotates in the direction of compressor rotation, but at a speed slightly less than that for rotating stall instigated at low speeds by inlet-stage stall. When compressor stall exists, the local flow is extremely unsteady. However, as for the case of rotating stall, the net flow through the compressor and the torque required to drive the compressor are steady.

The shape of the compressor stall limit line is important in regard to flexibility of engine operation. Experience has indicated a wide variation of stall-limit-line characteristics. Some typical stall limit lines are shown in Fig. 7. This figure is a plot of compressor stall pressure ratio against equivalent weight flow for several compressors. Curve A is typical of a low-pressure-ratio compressor and shows a fairly continuous variation. Curve B is for a high-pressure-ratio compressor and shows a very abrupt dip at intermediate values of equivalent flow. This dip is closely related to stalling of the inlet stages of the compressor. Curve C (Fig. 7) again represents a relatively high-pressure-ratio design. In this case the stall limit line at low speed is approximately linear. At some intermediate speed, the slope of the stall limit line increases appreciably. Thus, a change of slope occurs for curve C, but this stall limit line does not have the severe dip that is indicated for curve B. For speeds above the design value, the slope of curve C is practically zero.

The three stall-limit-line variations shown in Fig. 7 are only representative of those observed for axial-flow compressors. Each compressor has its own stall limit variation. These curves, however, serve to illustrate some of the basic characteristics that are encountered. In general, low-pressure-ratio compressors have rather continuous stall limit variations. High-pressure-ratio compressors have rather severe stall limitations at low and intermediate speeds. Some high-pressure-ratio units have large dips at intermediate speeds; whereas, other units have the double slope characteristics indicated by curve C. Some compressors have a definite leveling out of stall pressure ratio at high speeds; whereas, others continue to increase stall pressure ratio, even at speeds appreciably above the design value. The particular characteristics for a given compressor are determined by the basic stage characteristics and by the matching of stages. Currently, however, the stall limit characteristics of a compressor cannot be analytically determined from the geometry of the design, but must be determined experimentally.

Surge—Compressor surge is a system instability which results in large fluctuations in net flow through the compressor and in torque requirements, whereas, for compressor stall, the net flow and torque are steady. Stability analyses [6] have shown that surge may occur when the slope of the curve of pressure ratio versus flow characteristics becomes positive. For discontinuous multistage characteristics, surge may occur at the compressor stall point. If the slope of the static-pressure ratio characteristic of the compressor becomes positive without the occurrence of complete stall, surge may also occur. In this case, the surge flow fluctuations are small and the resulting surge is not considered to be serious.

When a compressor is operated in a conventional component test facility, violent surge is generally encountered at the compressor stall point. The surge

cycle consists of an alternate stalling and unstalling of the compressor [6] around the hysteresis loop indicated in Fig. 6. Because surge is a function of system instability, the occurrence of surge can be frequently eliminated by altering the characteristics of the external system of the compressor facility. One simple technique of improving system stability is to decrease the volume of the discharge ducting of the compressor. This can be done by locating the discharge throttle of the installation close to the compressor exit. Tests of this type have shown that compressor surge occurs at the compressor stall limit. Thus, the elimination of surge will not improve the operating limits of the compressor because performance at flows below that for stall is, in general, too poor for useful operation.

In an actual engine, the turbine stator is generally choked so that the compressor-discharge system is determined by the compressor-discharge diffuser and combustor volumes. Therefore, the possibility of eliminating surge in the actual engine is somewhat restricted. Because of the variations in unstalling hysteresis,[6] compressor stall may be prevalent at low values of rotative speed and compressor surge is generally encountered at high speeds. However, surge or stall both indicate the stall limit of the compressor, and both represent the limit of usable operation of the compressor.

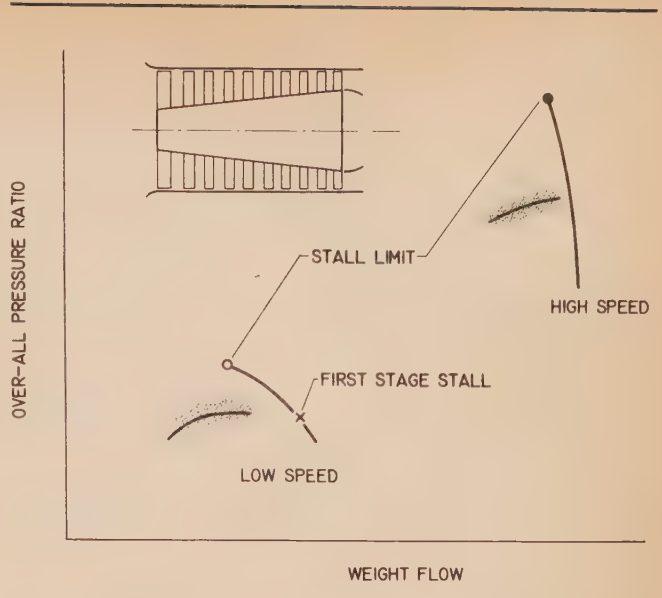

Fig. 6—Multistage compressor performance

Engine Operation

The preceding section has considered the basic compressor-performance-rating parameters and the process by which rotating stall and compressor stall or surge develop in the compressor. The following discussion is concerned with the limits imposed by rotating stall and compressor stall on the acceleration characteristics of the engine. The limitations imposed by compressor stall and rotating stall on the permissible range of flight Mach number and altitude are also considered. The effects of Reynolds number and inlet-flow distortions on compressor performance will be considered in later sections.

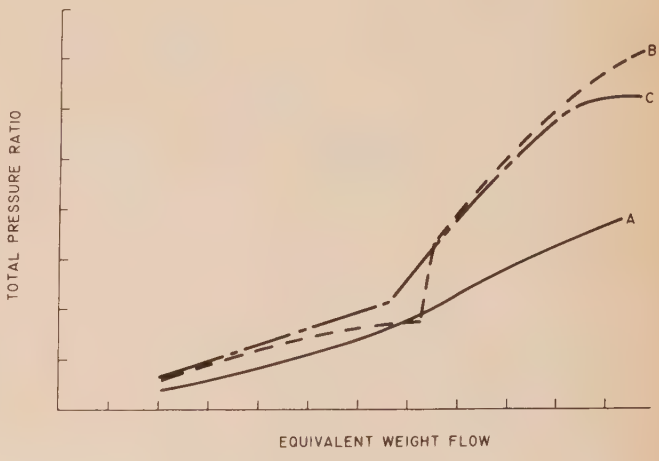

Fig. 7—Typical compressor stall limits

Single-Spool Engine—Engine acceleration: A typical fixed-geometry compressor performance map is presented in Fig. 8 for sea-level conditions. The heavy dash-dot line indicates the steady-state or equilibrium operating line of the compressor when installed in a fixed-geometry engine. This line is determined by the matching of the compressor with the other components of the jet engine. It might be considered that the other components act as a throttle on the compressor to force the compressor to operate at a given point at each speed. The locus of these points is then the equilibrium operating line and indicates the pressure ratio and flow of the compressor at each engine speed.

The design-point pressure ratio of the engine is indicated by the square symbol in Fig. 8. In order to obtain minimum fuel consumption, the turbine and

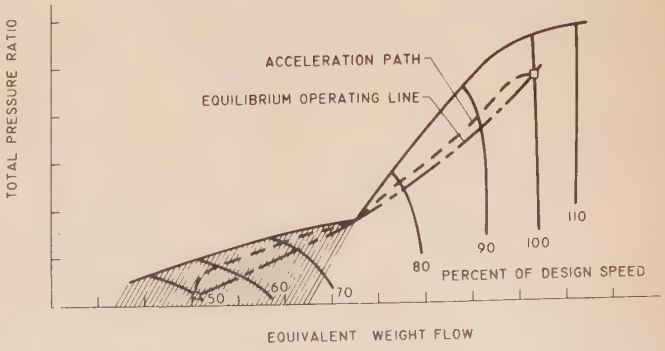

Fig. 8—Compressor operation in engine at sea-level conditions

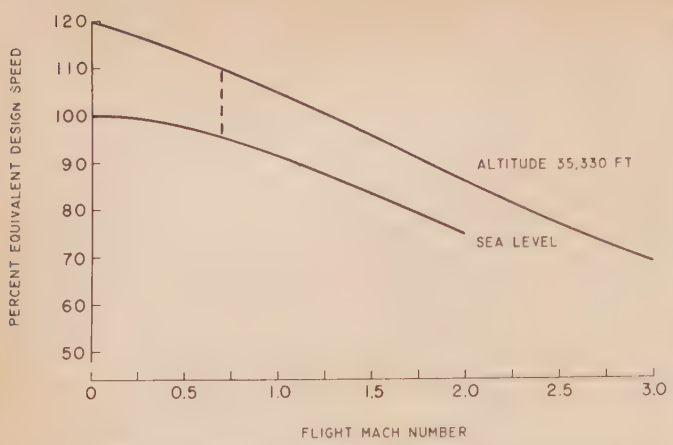

Fig. 9—Effect of altitude and flight speed on engine equivalent speed

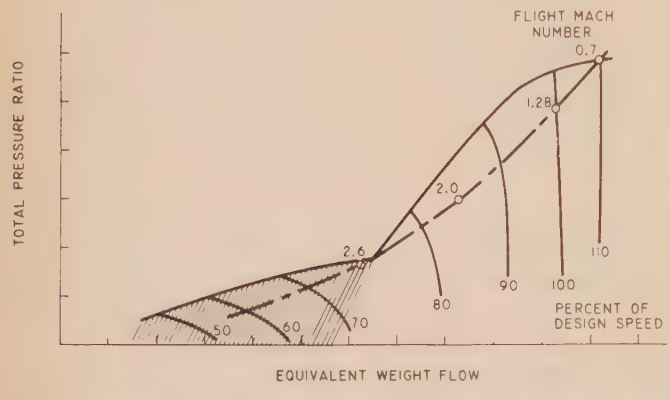

Fig. 10—Effect of flight Mach number at rated rotational speed and 35,330-ft altitude

exhaust nozzle are generally sized so as to cause the compressor to operate at as high a pressure ratio as is consistent with good compressor efficiency. The efficiency contours on Fig. 1 indicate that to achieve maximum compressor efficiency at design speed, the operating point must be relatively close to the stall limit line.

In order to bring the engine up to the design point, the starter runs the engine up to approximately 30 or 40% of design speed. Fuel is then ignited in the combustor, the starter disengages, and the engine speed increases to the idle speed (about 50% of design speed). The idle-speed operating point is indicated by the circled point on the equilibruim operating line (Fig. 8). This point is generally in the rotating-stall region.

Acceleration from idle speed to design speed is accomplished by injecting and burning more fuel in the combustor than is required for constant-speed operation. The resulting increase in turbine temperature forces the compressor to operate along a path such as that shown by the dashed line (Fig. 8). Thus, the acceleration path lies closer to the compressor stall or surge region than the equilibrium operating line. For mechanical reasons, it is, of course, desirable to accelerate as fast as possible out of the rotating-stall region where blade vibrations are probable. For operational reasons, it is also essential that rapid engine acceleration be obtained. Rapid acceleration requires large and rapid increases in fuel flow and turbine-inlet temperature. The magnitude of these fuel and temperature increases is generally limited by the margin between the equilibrium operating line and the stall line. The wider this stall margin, the more fuel that can be added and the faster the acceleration. Of course, in certain cases, it is conceivable that the margin for acceleration may be so great that the boost in fuel flow may be limited by the allowable engine temperature level rather than by the compressor stall line. Such a condition is, however, unusual and would be the exception when we consider acceleration from idle to design speed.

If the stall margin is so small that the acceleration path moves into the compressor stall region, then the engine generally will not accelerate. In such a case, combustion blow-out may result from the unsteady flow condition encountered in the compressor stall region, or an engine failure due to excessive temperatures may result. As has been pointed out earlier, even if the compressor does operate stably at flows below the compressor stall flow at a given speed, the performance in this region is generally so poor that excessive turbine temperatures would be required even for steady-state operation. Thus, any attempt to accelerate through this region would probably result in an engine failure.

For the stall limit line shown in Fig. 8, a reasonable margin exists in the range of 50 to 70% speed; the margin is extremely small at about 75% speed where the slope change in the stall limit line occurs. At speeds above 75% speed, an appreciable stall margin is again available. In compressors that show a large dip in the stall limit line, such as shown on curve B of Fig. 7, the stall limit line may actually intersect the equilibrium operating line. In this case, acceleration to design speed would be impossible without the aid of some variable-geometry features such as exhaust-nozzle adjustment or air bleed.

One of our major problems is, therefore, to provide sufficient margin between the equilibrium operating line and the compressor stall line at all speeds so as to permit satisfactory engine acceleration. The effectiveness of variable-geometry features in alleviating this problem is discussed in literature.[2]

Effect of altitude and flight Mach number: For a fixed-geometry turbojet engine, thrust is controlled by varying the engine speed. Inasmuch as design mechanical speed is the maximum operational speed consistent with allowable material stresses, maximum thrust at any flight condition (altitude and flight Mach number) is achieved at rated mechanical engine speed. Because of the variations in compressor-

inlet temperature with altitude and flight Mach number, the engine equivalent speed $N/\sqrt{\theta}$ for maximum thrust varies as shown in Fig. 9. This figure shows the variation of equivalent compressor speed with flight Mach number for standard sea level and 35,330-ft altitude at rated mechanical engine speed. At static sea-level conditions, the equivalent speed is 100% of the rated value. As flight Mach number is increased at sea level, the compressor equivalent speed decreases. At a flight Mach number of 0.7, the equivalent speed is about 95% of the rated value. Because standard temperature decreases as altitude is increased, the equivalent compressor speed increases from 95 to about 110% of the rated value for a climb from sea level to 35,330-ft altitude at Mach number of 0.7 and rated mechanical engine speed. Above 35,330-ft altitude, the standard temperature is constant, and the equivalent speed also remains constant for a given flight Mach number. As flight Mach number is increased for these high altitudes, the compressor-inlet temperature increases because of ram and equivalent speed decreases. At a flight Mach number of 1.28, the engine is again operating at design equivalent speed; at flight Mach number of 2.0, it operates at about 86% of the design value; and at flight Mach number of 2.6 it operates at 75% of the design speed.

If the fixed-geometry engine is operated at rated mechanical speed, as is the case for maximum thrust output, then these flight Mach number and altitude effects cause the compressor to operate along the dash-dot line shown in Fig. 10. In this compressor performance map, only those flight conditions above the critical Reynolds number are assumed to have been considered. This is, therefore, the same compressor map shown in Fig. 1 for sea-level conditions. The dash-dot line indicates the maximum thrust operating condition of the compressor at altitudes above 35,330 ft for various flight Mach numbers from 0.7 to 2.6. This line is very similar to the sea-level equilibrium operating line that was shown on Fig. 8.

Several problems are indicated here. If the compressor stall limit levels off appreciably at equivalent engine speeds above design, then compressor stall may be encountered during full-thrust operation at low-flight Mach numbers at high altitudes. In addition, because of the ram-temperature effect on equivalent speed, a small compressor stall margin at intermediate and low speeds may cause stall and surge problems at high-flight Mach numbers. Rotating stall may also be encountered at high-flight Mach numbers. For example, at the flight Mach number of 2.6, this compressor apparently would be operating in rotating stall with serious danger of blade vibration and engine failure. Incidentally, the rotating-stall frequency, which determines the blade vibration problems, is a function of rotational speed. The stall frequencies encountered at high-flight Mach numbers will be greater than those encountered at the same equivalent speeds at static sea-level conditions. Therefore, different, and perhaps more serious, blade vibrations may be anticipated at high-flight Mach numbers. Furthermore, the equivalent speeds shown on Fig. 10 are for maximum thrust operation. For any part-throttle operation that may be required, the engine speed will be lower, and the potential of encountering rotating stall at part throttle at high Mach numbers is greater than for full-thrust operation.

Double-Spool Engine—The double-spool engine is shown in Fig. 11. It is composed of two compressors driven independently (except for the aerodynamic coupling) through concentric shafts by two turbines. This type of engine has received a considerable amount of attention in recent years because it offers a degree of freedom in design that may result in some improvement in high-pressure-ratio engine configurations. Its benefits, in so far as freedom from stall problems is concerned, are based on the advantages of low-pressure-ratio versus high-pressure-ratio compressors. These advantages were illustrated previously in Fig. 7. In general, low-pressure-ratio compressors have smooth compressor stall lines and a small speed range over which rotating stall exists. In the 2-spool compressor, however, there are two compressor stall lines that must be avoided.

Engine acceleration: The sea-level equilibrium operating lines for a 2-spool engine are shown on the two compressor maps in Fig. 12. For idle speed (approximately 50% of rated outer-spool speed) the outer compressor operating point lies in the region of

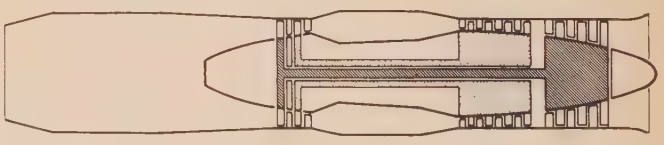

Fig. 11—Two-spool turbojet engine

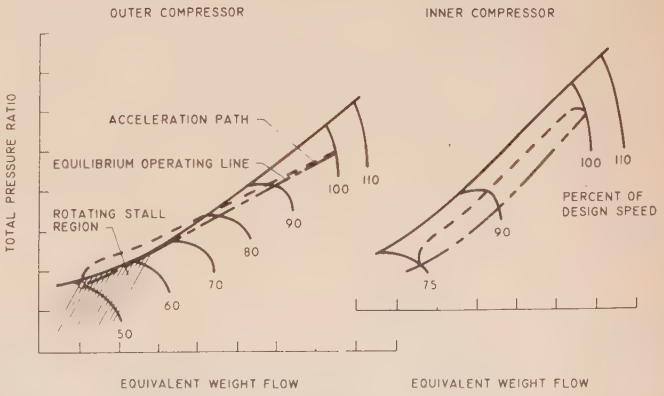

Fig. 12—Compressor operation in 2-spool engine at sea-level conditions

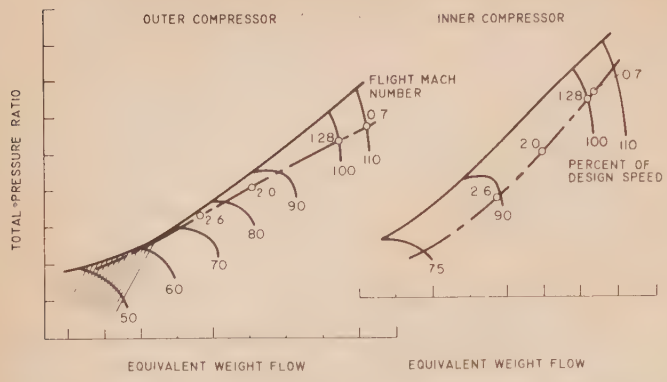

Fig. 13—Effect of flight speed on 2-spool compressor operation

rotating stall. At idle speed, the inner compressor operates at about 70% of rated speed and is out of the region of rotating stall for this particular case. The flow fluctuations induced by rotating stall in the outer compressor, however, may still persist through the inner compressor. As in the rear stages of a single-spool compressor, however, the amplitudes of fluctuations in the inner spool may be quite small. The margin between the equilibrium operating line and the stall limit line for the outer compressor is normally quite small at low engine speeds.

Analyses of the transient characteristics of 2-spool engines have shown that the effect of acceleration is to move the operating point of each compressor closer to the stall limit. This is indicated by the dashed lines that represent typical acceleration paths of each compressor. Therefore, stall margin is required in both compressors in order to achieve rapid acceleration.

Effects of altitude and flight Mach number: At high altitudes and high-flight Mach numbers, the maximum thrust operating point of each compressor is determined by the mode of engine operation. For a fixed-geometry engine with constant outer-spool mechanical speed, the outer-spool equivalent speed varies with flight Mach number as shown in Fig. 9. The maximum thrust operating points are shown on the compressor performance maps in Fig. 13. These are the same performance maps as were shown in the previous figure. As the flight Mach number is increased, the outer-spool operating point moves toward compressor stall. The inner-spool operating point, on the other hand, moves away from the stall line at high-flight Mach numbers but towards it at low-flight Mach numbers. In general, therefore, compressor stall problems may be instigated by either compressor if a wide Mach number range is required. For part-thrust operation, the engine speed must be reduced, and the potential of encountering rotating stall will be increased.

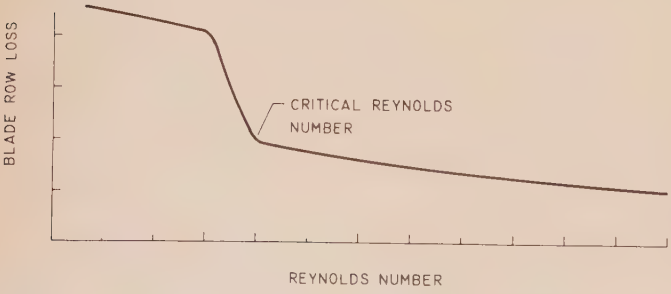

Fig. 14—Variation of blade-row loss with Reynolds number for fixed angle of attack

Effect of Reynolds Number

A conventional compressor performance map is normally obtained for relatively high values of inlet pressure. Increases in flight altitude, however, result in large decreases in compressor-inlet pressure, and a definite deterioration of compressor performance has been noted for operation at extremely low inlet pressures. This loss in performance is attributed to a Reynolds-number effect. Reynolds number is an expression for the ratio of viscous forces to inertia forces and is a measure of the boundary-layer characteristics of the flow process. Inasmuch as the major losses in the compressor arise as a result of viscous forces in the boundary layer, large variations of Reynolds number may be expected to appreciably alter the performance of an axial-flow compressor.

The standard expression for Reynolds number is $\rho Vl/\mu$ where:

ρ = Density
V = Velocity
l = Length
μ = Air viscosity

For single-blade rows or casades of airfoils, the density, velocity, and viscosity are generally taken as inlet conditions, and the blade chord is taken as the characteristic length. The variation of blade-row loss with Reynolds number for a fixed angle of attack is shown in Fig. 14. This figure shows that losses are small at high values of Reynolds number. As Reynolds number is decreased, the losses increase very slowly until the critical Reynolds number is reached. At this point, a sharp rise in loss is experienced. Further decreases again result in small increases in loss. In general, the large rise in loss can be attributed to flow separation that results from imposing a large local pressure rise on a laminar boundary layer, as discussed in the literature.[7]

In a multistage compressor, the value of density,

[7] NACA TN 1941, September, 1949, "Attainable Circulation of Airfoils in Cascade," by A. W. Goldstein and A. Mager.

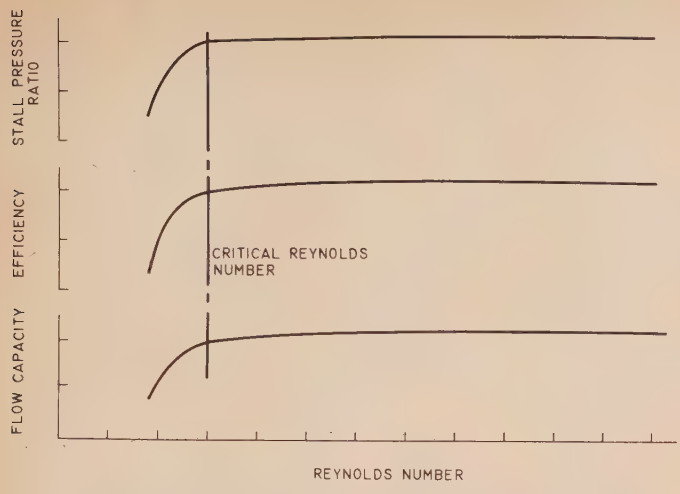

Fig. 15—Effect of Reynolds number on compressor performance for constant equivalent speed

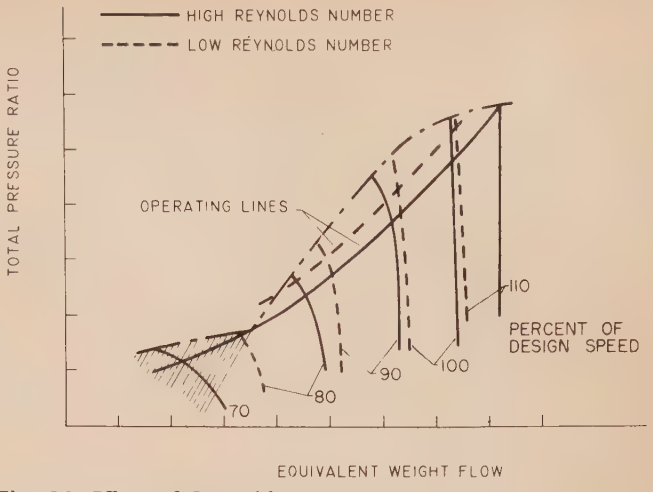

Fig. 16—Effect of Reynolds number on compressor operation in engine

velocity, and chord length vary appreciably from blade row to blade row. Thus, it is difficult to define a characteristic Reynolds number for a multistage compressor. Because inlet stages are believed to be more critical, the Reynolds number for the multistage compressor is generally based on inlet conditions and the inlet-stage dimensions. For a given compressor, the characteristic length is fixed, and the velocity is dependent on the operating point of the compressor. Inasmuch as the variation of density with flight condition is far more significant than the variation of viscosity, increases in altitude decrease Reynolds number. Increases in flight Mach number increase Reynolds number because of ram pressure rise. Thus, at high altitudes, low compressor Reynolds numbers may be expected at low-flight Mach numbers.

Compressor Performance—The effects of Reynolds number on compressor performance are shown in Fig. 15. This figure gives the variation of maximum airflow, maximum compressor efficiency, and stall-limit pressure ratio as a function of Reynolds number for a fixed value of equivalent compressor rotational speed. These curves indicate that decreases in Reynolds number have only a small effect on compressor performance until the critical value of Reynolds number is reached. Further decreases in Reynolds number result in extremely sharp decreases in flow, efficiency, and stall-limit pressure ratio. Similar results are obtained at other values of equivalent speed.

At each inlet pressure below that for critical Reynolds number, a separate performance map is obtained. The effect of a large decrease in Reynolds number on the compressor performance map is shown in Fig. 16. In addition to a shift of the constant speed characteristic lines to lower values of flow, the overall level of efficiency is greatly reduced. Because stall-limit pressure ratio and flow are reduced at a given speed, the stall-limit pressure ratio at a given flow may not be appreciably affected by large decreases in Reynolds number. In addition, the range of speeds over which rotating stall exists may be increased. With respect to engine operation, the large decrease in efficiency and airflow are considered to be the most significant deleterious effects of large decreases in Reynolds number.

Engine Performance—Single-spool engine: The effect of reduction in Reynolds number below the critical value may be qualitatively summarized by considering the engine-operating lines on the compressor performance map shown in Fig. 16. This figure shows the performance of the compressor at high Reynolds number (the solid lines) and low Reynolds number (the dash lines). The maximum thrust operating line is shown for these two Reyonlds numbers as the heavy solid and dash lines. For a fixed-geometry engine and rated actual rotational speed, the operating line of the compressor at low Reynolds numbers will shift to higher pressure ratios, or closer to the compressor stall limit at any given flow. This shift is a result of the large decrease in compressor efficiency. In addition to the effect on the operating line, the Reynolds number effect may also cause the compressor stall line to be shifted to lower pressure ratios at a given flow. Therefore, for those cases in which only a small stall margin is available at sea-level conditions, compressor stall or surge problems may be encountered at the low Reynolds number.

Because the occurence of rotating stall is determined to a large extent by the angle of attack on the front-stage compressor blading, the reduction in flow that has been observed with reduction in Reynolds number produces a rotating-stall condition at a higher speed for low Reynolds numbers than for high Reynolds numbers. Thus, a wider speed range

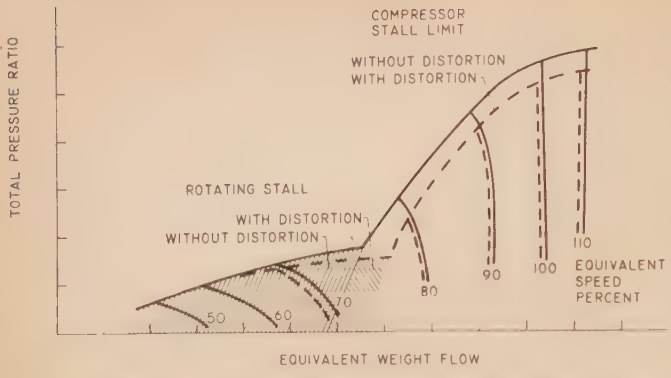

Fig. 17—Effect of inlet-flow distortion on compressor performance

of potential blade vibrations exists at low Reynolds numbers than at high Reynolds numbers.

Two-spool engine: The overall effect of Reynolds number in the 2-spool engine is very similar to the effect noted for the single-spool engine. Because the effect is believed to be concentrated in the inlet stages of the compressor, the performance map of the outer compressor is expected to vary most noticeably with Reynolds number. Smaller effects may be expected for the inner compressor. The change in performance of the outer compressor, however, will alter the match point of the two spools.

Effect of Inlet-Flow Distortions

In normal compressor tests, uniform flow is maintained at the compressor inlet. In an aircraft installation, however, flow distortions frequently exist at the compressor inlet. For subsonic flight these nonuniformities of flow are due to viscous and flow separation losses. For supersonic flight, nonuniformities can also arise from nonuniform shock losses. The severity of these flow distortions is dependent on the flight altitude of the airplane and on the inlet design.[1] Thus, the inlet-flow distortions vary appreciably even for a given installation.

In order to simplify the study of this problem, inlet-flow distortions have been identified as radial and circumferential distortions. Radial distortions are axisymmetric, but have variations of inflow velocity with radius. Circumferential distortions are uniform with radius, and velocities vary around the compressor circumference. In actual installations, however, some combinations of radial and circumferential distortions exist.

Compressor Performance—The effects of inlet distortion vary appreciably for different compressor designs. Some compressors are more adversely affected by radial distortions and others by circumferential distortions. In addition, the deterioration of compressor performance for a given magnitude of distortion also varies. In the following discussion, only the general effects of inlet-flow distortion will be considered without differentiations as to type or magnitude of distortion. The magnitude, and in some cases even the direction of the effect, is determined to a very large extent by the type of distortion, the magnitude of distortion, and the compressor design.

The general effects of inlet-flow distortions on compressor performance are indicated in Fig. 17. In this figure, the performance for uniform inlet flow is indicated by the solid lines and that for a severe inlet-flow distortion by the dotted lines. The upper solid line is the compressor stall limit for uniform flow; that for distorted flow is the lower dashed line. The increase in the region of rotating stall due to distortion is indicated by the change in shading.

As indicated in Fig. 17, inlet distortions may result in a slight decrease in compressor flow capacity. Furthermore, the effects of distortion on compressor efficiency are normally not appreciable. Inlet-flow distortions, however, show severe effects on the stall characteristics of the compressor. Fig. 17 indicates a large decrease in the stall-limit pressure ratio of the compressor. The speed range over which rotating stall exists is also greatly increased inlet-flow distortions.

Engine Performance—Single-spool engine: Generally, the principal effects of distortions of inlet flow are to drop the compressor stall line to lower pressure ratios and to increase the maximum engine speed at which rotating stall exists. These effects are indicated on the compressor performance map shown in Fig. 18. This figure presents the maximum thrust operating line for the fixed-geometry engine operating at rated rotational speed. The general qualitative trend of the distortion effect is indicated in this figure; however, it is conceivable that certain distortions could result in an improvement in performance.

From this figure, apparently the stall margin is reduced and the rotating stall region is increased. Operation between 75 and 80% speed would require operation beyond the stall limit. In addition, the flow distortions often produce hot spots in the turbine which may require derating of the engine in order to

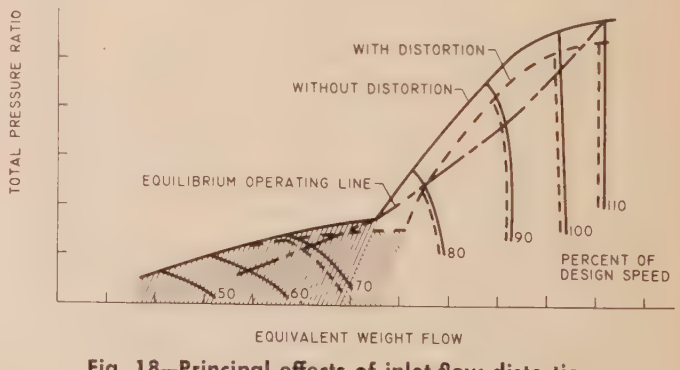

Fig. 18—Principal effects of inlet-flow distortion

avoid turbine failures. Furthermore, the effects of Reynolds number and inlet distortion are compounded in order to limit attainable altitude.

Two-spool engine: The effects of flow distortion are very similar in the 1- and 2-spool compressors. It is reasonable to expect that the outer-compressor performance map of the 2-spool engine will be affected as indicated in Fig. 17 for the single-spool compressor. The effect on the second-spool performance may not be quite so noticeable because of the possible attenuation of the distortion in the outer compressor.

In general, therefore, the 2-spool engine apparently will encounter surge and stall difficulties very similar to those of the single-spool engine. For extremely high-pressure ratios, however, the 2-spool compressor may offer some advantage because of the improved stall-limit characteristics associated with the low-pressure-ratio compressors.

Summary

Two closely related compressor stall phenomena are encountered during off-design operation of axial-flow compressors. Rotating stall results from stalling of the inlet stages of the compressor and occurs at low values of compressor speed. This type of stall does not necessarily produce severe compressor performance deteriorations, but results in unsteady flow that is a serious source of blade failure due to vibration. Complete compressor stall results in a severe drop in compressor performance and represents the limit of usable operation of the compressor. Compressor surge frequently is instigated by the occurrence of compressor stall. Increases in design pressure ratio of the axial-flow compressor increases the range of rotative speeds over which rotating stall exists and also adversely affects the attainable stall-limit pressure ratios at low values of rotative speed.

Reductions of Reynolds number that are encountered at low-flight Mach numbers at high altitude adversely affect compressor performance. The primary effects are a reduction in compressor flow capacity, efficiency, and stall-limit pressure ratio. The major effects of inlet-flow distortions are to decrease stall pressure ratio and increase the range of speeds over which rotating stall exists.

In the single-spool-type turbojet engine, rotating stall is a major source of blade vibration. The compressor stall limit presents a serious problem in regard to engine acceleration. For operations at high altitude, compressor stall may be encountered at low-flight Mach numbers or at extremely high-flight Mach numbers. The limitations imposed by compressor stall are severely aggravated.

DISCUSSION

Suggests Two Solutions to Compressor Stall Problems

—F. Schwoerer
Westinghouse Electric Corp.

THE engine manufacturer wants to design for operation close to the compressor stall limit for two reasons: low specific fuel consumption and low specific weight. He must, however, arrive at a satisfactory compromise between these two known factors and the possibility of stalling the compressor in operation due to such semiunknown factors as: inlet-flow distortions, Reynolds-number effects, engine-to-engine variations, and deterioration of engine performance with time. In the past, the engine manufacturer has not had complete information about these other variables. As a result the industry has had troubles. Today, the industry is making a major effort to evaluate such factors as distortions, and their effect on engine operation. However, I believe that the semiunknown factors I mentioned will always be unpredictable to some degree.

There are then two general approaches to the problem of compressor stall in turbine engines. Borrowing from the control jargon, I would like to christen them "scheduled" and "closed loop" solutions.

The most elementary "scheduled" solution is to design the engine with sufficient margin between the operating line and the compressor stall line, so that all engines produced will be stall-free under all operating conditions throughout their entire lives. This approach is fine except for the resulting penalties in specific fuel consumption and specific weight. Certainly it is the easiest and most obvious approach and is the one most generally followed to date.

We can think of a more sophisticated "scheduled" solution. It might be possible and practical to measure the magnitude of, say, inlet-flow distortion and Reynolds number. If so, the magnitude of these parameters could be fed to the control and could be used to actuate a "trim" on the basic operating line. Thus when the engine was subjected to large flow distortions or low Reynolds numbers, the control could cause the engine to operate on a modified, lower operating line. This general approach might be very profitable. It reduces the penalties imposed by the simpler approach. It is not dependent upon development of a stall sensor.

The ultimate solution is a "closed loop" control actuated by a stall sensor. If a sensor can be developed (and this would be no small accomplishment) the solution can be very direct. We need not anticipate precisely what distortions, tolerances, and the like will result, for the sensor would detect incipient stall. Control action would then move the compressor out of danger.

In looking for a stall-sensing device, the overall performance parameters of the compressor, individual stages, or other engine components have already been thoroughly examined. There has been little success in finding a parameter that gives the desired stall warning. Therefore, I believe we must go beyond these overall parameters and search for detail phenomena that may give us an indication of incipient stall. The occurrence of rotating stall may be the answer. Or the answer may lie in local boundary-layer separation or velocity profiles in critical portions of the compressor. To this end, we need a better understanding of detailed flow phenomena within the compressor. NACA and other

research agencies are making significant contributions along this line. Also called for, I believe, is the development of rugged, fast-response pickups. The hot-wire anemometer, which is now primarily a laboratory instrument, might be used if it can be made sufficiently rugged.

Whatever phenomenon is sensed by the stall sensor must occur before the compressor actually stalls, in order to permit control action to take place. Surge cycles are undesirable and, further, the aerodynamic hysteresis associated with stall means that once the compressor is allowed to stall, recovery may be difficult.

A few remarks on what control action might take place should be mentioned. It is highly desirable that essentially no loss in thrust should result. A temporary increase in fuel consumption should be quite permissible however. One control action that accomplishes these objectives is simultaneous increases in turbine-nozzle area and exhaust-nozzle area.

Discusses Differences of Turboprop and Turbojet Compressor Designs

—L. B. Lisher
General Motors Corp.

THE primary differences between the turboprop compressor design problems as compared to those of the turbojet will be discussed.

Due to the more complete expansion in the turbine, the cycle analysis of the turboprop shows that the optimum pressure ratio is below that of a subsonic turbojet. Using present turbine temperature limits and component efficiencies, a pressure ratio of about 12 to 13 gives about optimum performance for cruise at 35,000 ft for a turboprop. This would be somewhat over the pressure ratio of peak power output but have lower sfc. The Allison T56 operates at a pressure ratio of about 11¼/1 at this condition.

Due to the lower ram, the corrected speeds for the compressor are much higher for the turboprop than for the turbojet as pointed out in Fig. 9. However, the idle-speed requirements for the turbojet are much different. The turbojet idles at about 40% to 50% speed, whereas the turboprop usually idles at a much higher speed, about 95% rated speed for the Allison T56 military engine. Therefore, the required operating range of the turboprop, except for starting, lies in the corrected speed range of 87% to 112% of rated speed for hot-day to high-altitude operation.

Because of propeller problems and size requirements, the turboprop engines are in general much smaller in size than the modern turbojet engines. The present T56 turboprop has a compressor tip diameter of less than 15 in. This small size presents many other associated problems to the compressor designer—such as low Reynolds number and critical radial clearance problems. Due to the high corrected speeds mentioned earlier and the low Reynolds number resulting at altitude, it has been necessary to match the compressor for best performance obtainable at altitude; however, by so doing, the surge curve was naturally worsened. Therefore, the final product usually results in a compromise between these two extremes. The radial seal clearances were found to represent considerable performance losses and also large depressions in compressor surge range; therefore, every effort has been made on the Allison turboprop engines to reduce these clearances to the minimum at both the rotor tips and at the interstage vane seals.

Allison has used compressor bleeds to provide sufficient margin for starting and acceleration in the low-speed range. Perhaps, other methods could be used as mentioned in some of the previous papers—such as variable geometry and inlet blockage. In general, rotating stall occurs only at the high pressure ratios just prior to surge. For the commercial versions of the Allison turboprop, idle speed will be lowered to about 75% speed which will still be above the rotating-stall range.

Now that the major design considerations for a compressor to be used on a typical turboprop engine have been stated, it may be of interest to trace briefly the development history of the T56 compressor. The 14-stage axial-flow compressor was originally designed to have an airflow of 30 lb per sec at a pressure ratio of 8.5/1 for sea-level static conditions with an operating speed of 14,300 rpm.

Initial component testing revealed that design airflow was attained, that the operating pressure ratio, Rc, could be 9.5/1 instead of 8.5/1, and that the compressor had an unusually smooth and favorable surge line for a high-pressure-ratio machine.

Although the compressor map as such appeared encouraging, two conditions unfavorable to engine performance became apparent. First, the peak efficiency occurred at 85% speed which would be bad for altitude performance, and secondly, the airflow tended to level off at slightly over 30 lb per sec, which also hurt altitude performance.

To correct these unfavorable conditions, compressor stages were rematched to better suit the engine requirements. The restaging was done by changing the setting angles throughout the compressor in order to move the match speed to a higher value. Interstage seal leakage was also reduced at the same time.

This restaged compressor improved both altitude and sea level performance by pumping considerably more airflow and had the peak efficiency island moved up closer to normal operating speeds. The surge line at low speeds was, of course, lowered, but good starting and accelerating characteristics were obtained by bleeding air from two of the compressor stages. This compressor was used on preproduction (YT56) engines.

A redesigned first stage taking full account of radial flow considerations was then substituted for the original first stage. This fattened up the surge line at the critical bleed closing speed range (12,000–13,000 rpm) so that the match speed could be moved still higher by resetting the inlet guide vane. This compressor had a peak capacity of nearly 36 lb per sec, and peak efficiency occurred very close to the sea-level operating point. This compressor is used on the T56 engine. A slight increase in midstage bleed was necessary since low-speed surge was adversely affected by these changes.

It is evident that this compressor has been developed to give best performance at altitude cruise conditions—airflow, surge pressure ratio, and efficiency have all been improved by comparison with the original design at the high corrected speeds. It is noted that the gains at these high corrected speeds would have been impossible without some means of offsetting the associated losses in the starting range. Fortunately, the dual starting bleed is so beneficial that starting acceleration is better by far with the T56 compressor than with the original design without bleed.

ORAL DISCUSSION

Reported by J. D. Redding

Westinghouse Electric Corp.

J. S. Alford, General Electric Co.: Is rotating stall sufficiently critical to cause failure of the compressor blades if the first flexual resonance mode of vibration is absent.

Mr. Benser: Not to my knowledge.

Mr. Alford: If there are six to seven segments of rotating stall, will breakage of the blades still occur?

Mr. Benser: Yes, if resonance is present.

Major K. L. Jones, USAF: Has rotating stall been encountered or noted in all compressors tested at NACA Lewis Flight Propulsion Laboratory?

Mr. Benser: Yes.

Major Jones: Are the frequencies of rotating stall such as to always cause failure of the compressor blades?

Mr. Benser: No.

R. Bodemuller, Bendix Aviation Corp.: How accurately can the problem of oompressor stall be predicted with cascade test data?

Mr. Benser: It is extremely difficult to predict from such data whether stall will occur or not.

Mr. Bodemuller: Can the abrupt stall limit be located and eliminated by compressor research?

Mr. Benser: Not entirely, but such research will aid greatly in reducing the probability of stall being encountered.

COMBINING THE FEATURES OF DISC AND SHOE BRAKES

T. H. Thomas, Bendix Aviation Corp.

This paper was presented at the SAE Passenger-Car, Body, and Materials Meeting, Detroit, March 7, 1956.

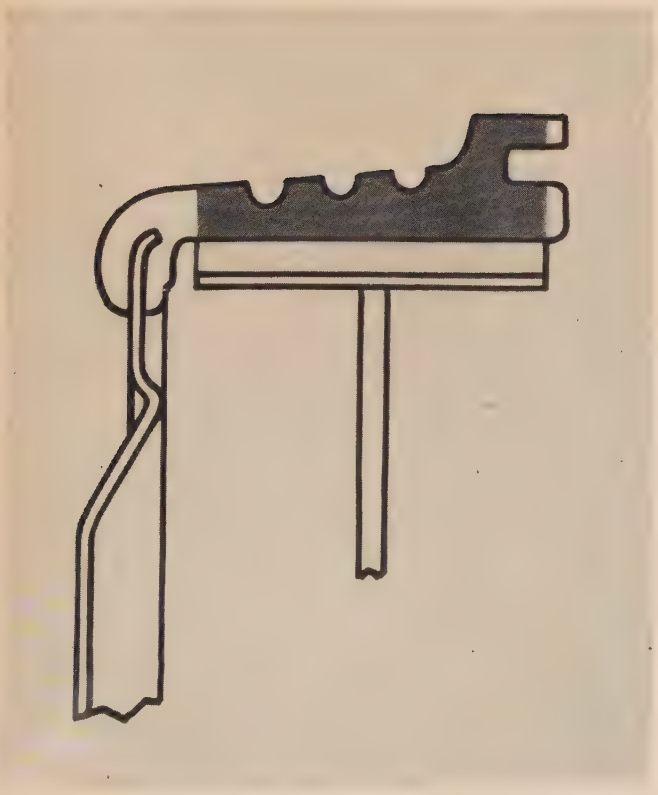

Fig. 1—Conventional drum brake with one rubbing surface

SINCE this is a paper on the new disc-servo brake which is related in principle to both the shoe brake and the disc brake, I would like to review briefly the terminology and principles of these two types of brakes.

Fig. 1 illustrates the shoe brake, which is almost universally used on passenger cars and trucks. The friction-lined shoes expand outward against the inside cylindrical surface of the brake drum. Various shoe actions have been developed to bring the shoes in contact with the drum and to provide self-energizing action to reduce the pressure necessary on the brake pedal.

As an example of this brake Fig. 2 shows a Bendix duo-servo brake such as is used on most passenger cars. The primary shoe is pressed into contact with the drum by hydraulic force originating at the brake pedal. Friction between the primary shoe and the drum moves the shoe sufficiently and thus brings the secondary shoe into contact with the drum. In addition, friction between the primary shoe and the drum exerts a force which presses the secondary shoe against the drum and thus multiplies the braking effect.

This duo-servo brake was originated 25 years ago and has continually undergone refinement. It is doing a very good job and has certainly reached a new high of performance on the 1956-model cars.

THE disc-servo brake was designed to eliminate brake fade. To overcome limitations of existing brake structures, desirable features of the disc and shoe brakes were combined.

The introduction of three friction surfaces, two pads, and a shoe, and the addition of a ball-and-ramp self-energizing mechanism contribute, the author says, to a brake that has greater fade resistance and longer wear of small segments of the brake lining.

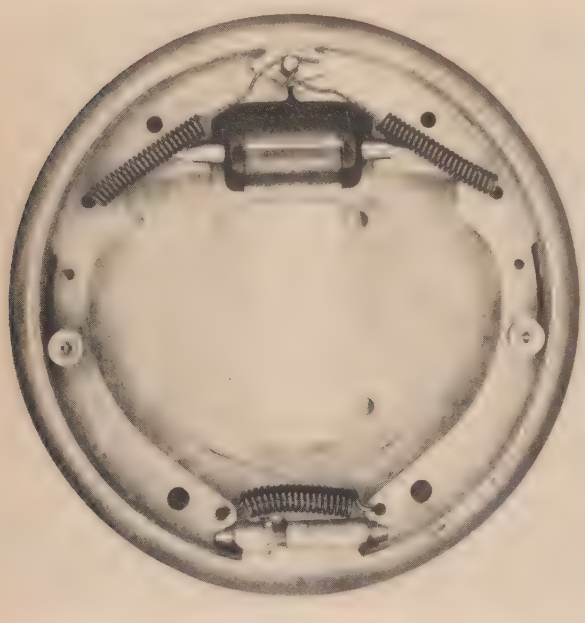

Fig. 2—Shoe brake of the duo-servo type

The present accelerated efforts to develop a substitute type of brake grow out of the fact that, with present engine power, certain limitations of shoe brakes can be demonstrated. For example, it is possible to make enough emergency stops in rapid succession from high car speeds to produce temperatures that tend to expand the drums away from the shoes, resulting in loss of pedal reserve. Extreme temperature can also cause varying degrees of loss of friction in the lining material, resulting in brake "fade."

For comparison the disc brake is illustrated in Fig. 3. This is the "spot" type, or caliper brake. Flat friction-lined pads or "spots" take the place of the arc-shaped shoes. The mating friction surface is a rotating disc. The pads are pressed against either side of the disc (that is, the disc is clamped between two opposing pads). Actuation is usually by hydraulic cylinders.

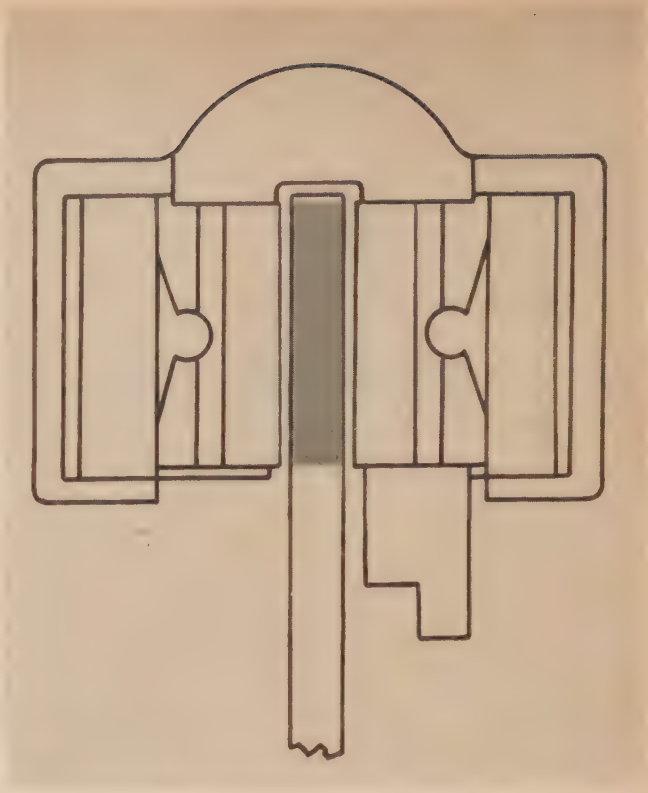

Fig. 3—Caliper disc brake with two rubbing surfaces

Fig. 4 is a picture of a caliper brake used on a sports car. A comparatively small area of lining is swept by a large area of disc surface. This is highly favorable to heat dissipation, thereby avoiding excessive lining temperature; moreover, in this design, drum or disc expansion is not a factor.

This, and similar disc brakes have been used quite successfully in relatively small quantities on sports cars, particularly in England. Brakes, usually with more than one opposed pair of spots, or lining buttons, have been quite successful in racing competition here and abroad.

The exposed disc area, however, brings its own set of problems. Work done years ago in the Arkansas mud, experience with cars standing in the mist and salt-laden atmosphere of the East Coast, and recollections of what calcium chloride and salt combined with slush and grit did to disc brakes on one of the small cars built in the United States a number of years ago have slowed down the adoption of such brakes on American passenger cars.

Brakes of this type are, of course, quite ineffective when compared with most of the present-day shoe brakes, because they do not incorporate any servo or self-energizing feature. It is almost a certainty, therefore, that power braking would be needed with these brakes even if a full-stroke, full-height pedal were used. On the other hand, most of today's passenger cars use power brakes to obtain the advantages of a short-stroke, low-height pedal. Although work is

being done on exposed disc brakes, the job of development for American pasenger cars is a difficult one.

A third brake design, the enclosed disc type, is shown in Fig. 5. The friction pads here expand outward against the inside end surfaces of a brake drum. This has the advantage of a large-swept area. It has the further merits of being self-enclosed and better adapted to self-energizing arrangements which help to avoid excessive pedal pressure.

There is also a disc brake used with outstanding results on airplanes. (See Fig. 6.) A single brake of this type can have the energy-absorbing capacity of all the brakes on 25 or 30 passenger cars. Such brakes are probably the most highly developed machines in use today for converting kinetic energy into heat. This brake attains near-perfection in the job of slowing down and stopping an airplane after it has touched the ground. But in this it has a *specific* job to do, and it cannot be modified to fit the four wheels of automobiles and to do an acceptable job as a passenger car brake.

In the first place, although it is a very effective heat reservoir, it is not intended to be an equally effective heat dissipator. In automotive use, a brake must be a good heat dissipator. Automobile brakes are often used a number of times per minute and the heat they produce during these successive applications cannot be stored. Secondly, most aircraft brakes are not very quiet when they are applied very lightly. This is not of any consequence on an airplane because the brake noises cannot be heard above other noises which the airplane emits while taxiing, but unfortunately for the brake people, the present-day automobile is pleasingly quiet. Although the life of aircraft brake facings and linings is truly phenomenal under very severe operating conditions, neither linings nor rotors of these

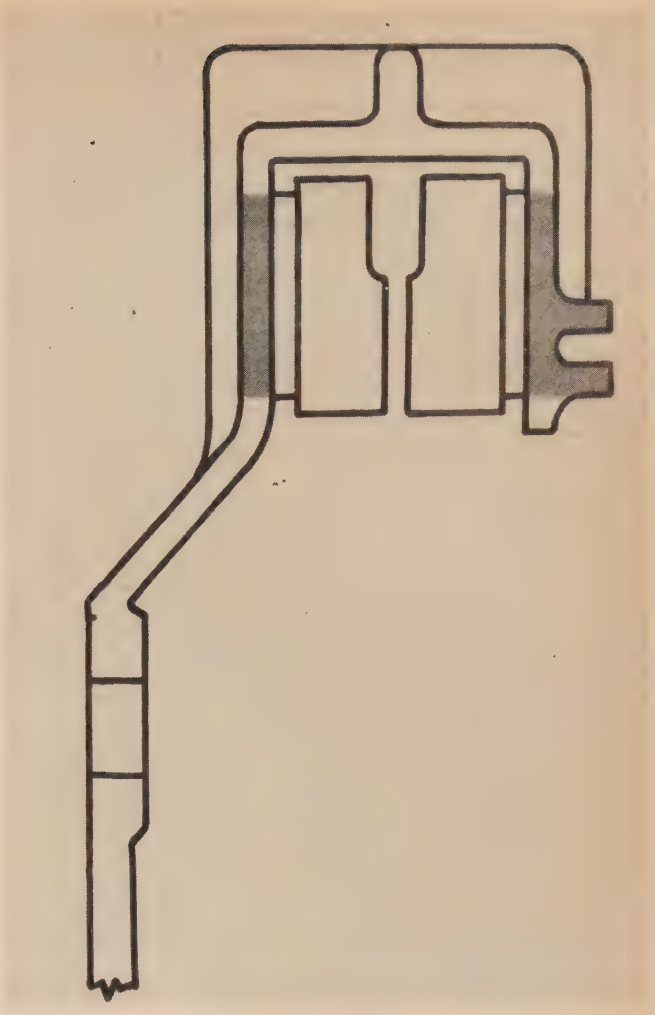

Fig. 5—Enclosed disc brake with two rubbing surfaces

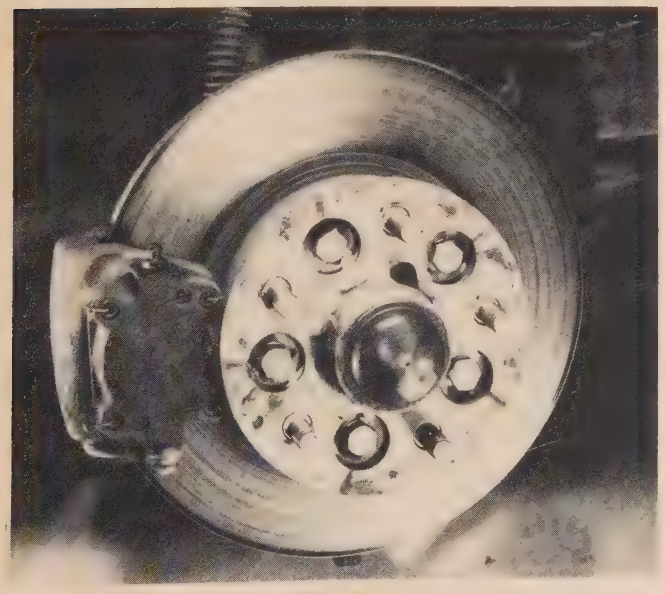

Fig. 4—Caliper-type brake as used on sports cars

Fig. 6—Disc brake of the type used on airplanes

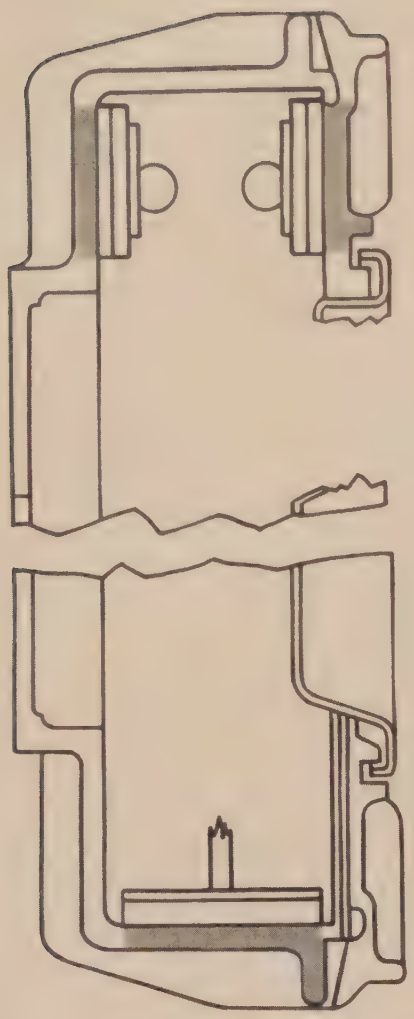

Fig. 7—Disc-servo brake with three rubbing surfaces

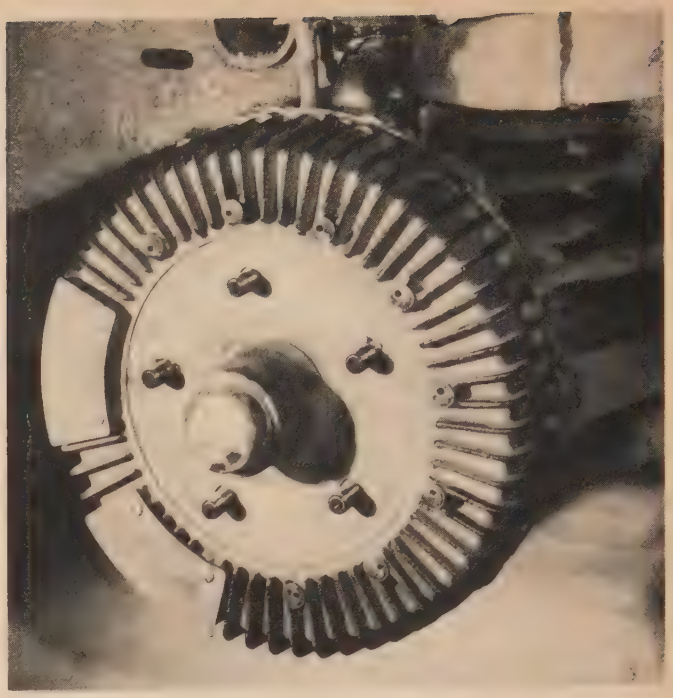

Fig. 8—Disc-servo brake as seen from the outside on a car

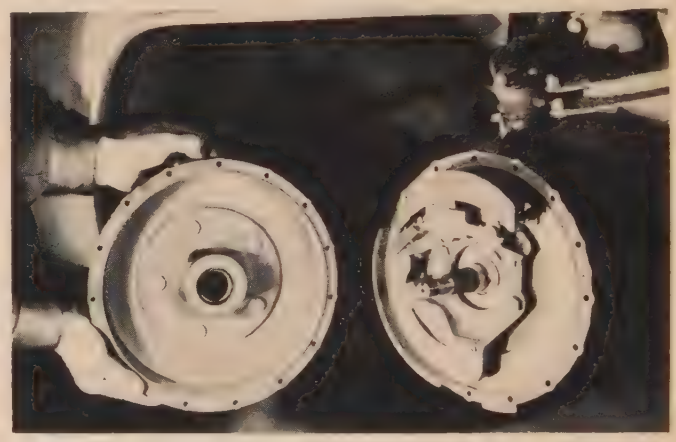

Fig. 9—Outer section of the disc-servo brake removed to show pads, shoe, and friction surfaces of drum

brakes, when adapted to pasenger cars, do very well under normal driving conditions where all but a small fraction of the brake applications are very light ones.

Fig. 7 illustrates this new automotive brake design called the disc-servo. The brake drum is similar to the enclosed disc type in that flat-end surfaces are friction surfaces. In addition, like the shoe brake, the cylindrical surface is also a friction surface. Two friction pads expand outward to bear against the flat end-surfaces of the drum. An arc-shaped shoe, similar to those used in present shoe brakes, moves outward to bear against the inside cylindrical surface of the drum.

There are three friction surfaces: the two flat end-surfaces of the drum and the inside cylindrical surface. The stationary friction units of the pads and shoes are small compared to the large swept area of the drum. Cooling advantages result from spot-disc construction and the three swept surfaces instead of two. We have the self-enclosed feature of the disc and the self-energizing feature and other advantages of the well-proved, highly developed shoe brake. However, there is an inherent improvement in stability resulting from the fact that the serving elements (the pads) do not operate against the same friction surface that is used by the element which they serve (the shoe).

The principle of the action is this: a hydraulic cylinder is placed so that it forces the two pads out-

ward against the drum. This action is self-energizing through a ball-and-ramp mechanism which also tends to spread or force the pads outward against the drum. The spreading action, of course, involves a certain amount of movement of the pads as they tend to be carried around with the drum. A linkage connects the pads with the shoe. As the pads move around slightly with the drum, the motion is transmitted through the linkage so that the shoe moves into contact with the drum. Following that, the force of the friction between the pads and drum is transmitted through the linkage to press the shoe against the drum, and the work of braking is divided almost equally between the pads and the drum.

Note that the force from the driver's foot serves to bring the pads into contact with the drum. A good share of the force for pressing the pads against the drum is furnished by the self-energizing action of the ball-and-ramp mechanism. Moreover, the application of the *shoe* is entirely a servo action; there is no connection between the foot pedal and the shoe. This servo-action bears a resemblance to the shoe action we reviewed in the duo-servo brake. It is a case of utilizing the friction of brake action to furnish power for brake application. That is, we are utilizing the momentum of the vehicle itself to furnish force to stop the vehicle.

Fig. 8 shows the way the disc-servo brake looks from the outside on a car; the front wheel has been removed. The drum serves as a protecting housing for the mechanism. Cooling fins help dissipate heat.

In Fig. 9 the outer section of the drum has been removed to show the pads, the arc-shaped shoe, and the friction surfaces of the drum.

In Fig. 10, the major parts of the brake are labeled, to establish terminology. They are: pads, relay lever, reverse link, adjusting link, and shoe.

Fig. 11 shows the wheel cylinder, which is located outside the brake itself and the levers which connect the wheel cylinder piston to the left-hand end of the pads. As the wheel cylinder-piston force is applied to the left-hand ends of the pads, the pads are spread apart. Their left-hand ends, therefore, contact the disc surfaces of the housing, and the pads are carried around slightly with these surfaces.

Fig. 12 shows the location, and the construction, of the mechanism which spreads the *right-hand* ends of the pads. It is a ball-and-ramp arrangement connecting the pads with the upper end of the relay lever so that, as the pads move in either direction (to the left in this illustration) two things happen. First, the right-hand ends of the pads are spread so that the pads engage the surfaces of the drum at *both* ends, and consequently along their entire length. Second, a force equivalent to the braking force developed by

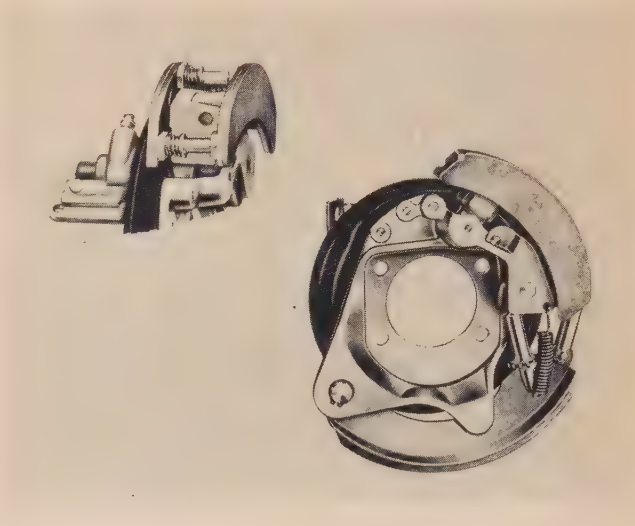

Fig. 11—View of wheel cylinder and levers connecting wheel cylinder piston to left-hand end of pads

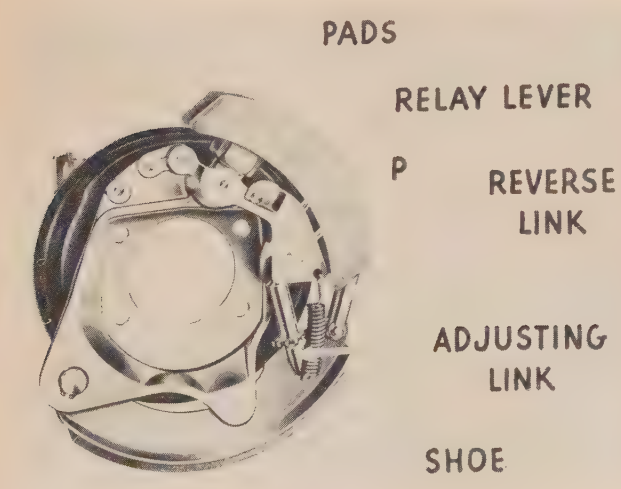

Fig. 10—Diagram of the disc-servo brake

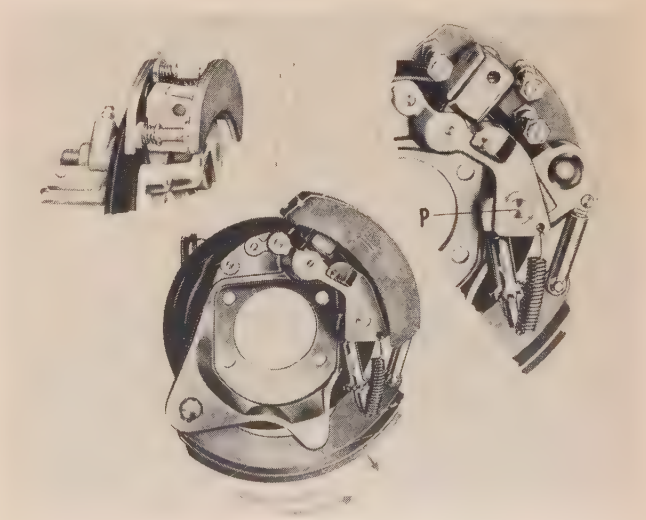

Fig. 12—Location and construction of mechanism spreading right-hand ends of pads

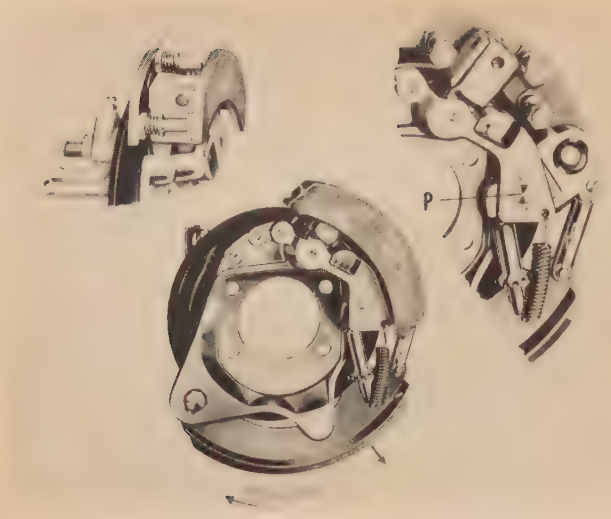

Fig. 13—Diagram showing direction of movement of parts during braking while traveling in reverse

the pads, is applied to the upper end of the relay lever tending to make it revolve counterclockwise about point "P" which is a stationary pivot.

The tendency of the relay lever to rotate counterclockwise causes the adjusting link to exert a downward force against the right-hand end of the shoe. This presses the shoe into the drum with a force that is proportional to the amount of braking force being exerted by the pads.

It will be seen that, in this case, the shoe is working as a *trailing* shoe. Trailing shoes operate consistently and, therefore, are relatively unaffected by such things as changes in lining coefficient. However, they require application of a considerable force, compared with leading shoes, for equivalent brake performance. This large force requirement is considered disadvantageous when the work (force and distance) must be supplied from a brake pedal which is foot-operated. With trailing shoe brakes the tendency is to require considerable pedal pressure or considerable pedal travel (sponginess) or both, unless a rather large power brake unit is used.

In this case, however, the force delivered to the trailing shoe by the pads is multiplied (about 3/1) by the relay lever, and both the force and distance components of the work required to actuate the shoe come from the work which must be done by the pads to decelerate the car. It is worth emphasizing that none of the work required to actuate the shoe is supplied by the hydraulic brake system; it is all obtained from the servo effect of drum rotation and friction. Actuation of the shoe, which does about 55% of the braking, is therefore obtained at no expense to pedal pressure or pedal travel.

The rear brakes are equipped with a bell-crank lever and a connection through which the applying levers can be moved mechanically. The bell-crank is connected to the parking brake cable. The left-hand ends of the pads can thus be spread *mechanically* to apply the brake for emergency stand-by and parking, just as it is applied *hydraulically* for service braking.

Direction of movement of parts during braking while traveling in reverse is shown in Fig. 13. Rotation of the drum is, in this case, clockwise. The pads are therefore carried in a clockwise direction, rotating the relay lever about point "P" in a clockwise direction. The thrust from the lever to the brake shoe is delivered through the reverse applying link, and the shoe is operating as a *leading* shoe. To compensate for the action of the shoe as a leading shoe, and to maintain proper work distribution between the pads and the shoe, the lever ratio for reverse braking is approximately 1/1 rather than 3/1. Thus, the *overall* effectiveness is essentially the same in either forward or reverse directions of car travel.

Fig. 14 has been prepared by averaging data taken by means of thermocouples located in brake drums and housings at a distance of 0.090 from the rubbing surface in each case. They show that, when conventional 11-in. brakes are used, the temperatures obtained during successive stops from 70 mph at 1-mile intervals rise, until they begin to level off after 12 stops at a little over 700 F. When the same test is conducted in the same car under identical conditions, but with disc-servo brakes instead of conventional shoe brakes, the temperatures tend to level off at approximately 500 F after 17 or 18 stops. There is, therefore, a difference of slightly more than 200 F between the temperatures obtained under identical conditions with the two brakes. This, of course, contributes to the fade resistance of the brake, which is exceptionally good, and also to the ability of relatively small segments of brake lining to stand up for surprising periods of time.

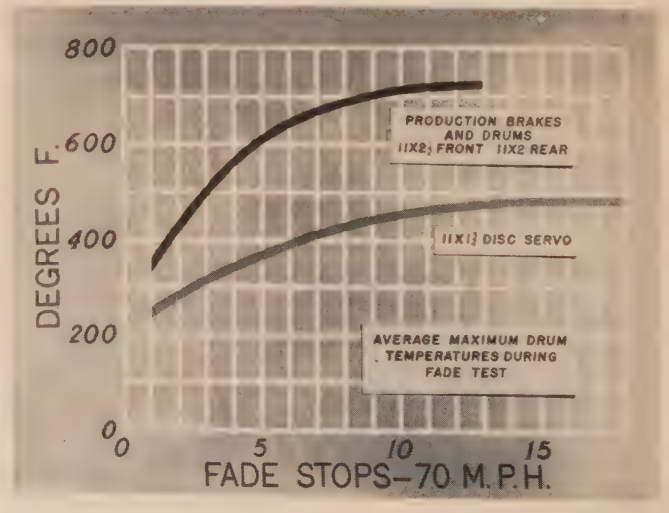

Fig. 14—Comparison of temperatures of conventional and disc-servo brakes obtained during stops from high speeds

We believe the disc-servo brake, combining as it does a number of desirable features of both disc brakes and shoe brakes, provides a very good measure of all the improvements that are agreed worth while. At the same time, it meets almost all the requirements which have meant good, accepted practice in the field of automotive brake design for the past 25 years.

DISCUSSION

Disc-Servo Brakes Noteworthy, But Need for Adequate Warning of Brake Failure Persists

—H. P. Hayes
Ford Motor Co.

MR. THOMAS suggests a combination of an enclosed disc and a drum brake that has relatively high energization to beat brake fade.

The disc-servo brakes combine an enclosed outward expanding disc brake with an energized shoe operating on a conventional drum surface. This design is noteworthy in several respects. First, there is no pedal travel or fluid displacement used to actuate the conventional drum shoe. Secondly, the brake fluid and wheel cylinders have been moved outside the brake proper to eliminate the possibility of boiling brake fluid under severe use.

I believe that one of the brake designer's basic objectives, whether he be working with disc brakes or drum brakes, should be to develop a brake and lining combination that will give the driver warning that he *has been* or *is* using the brakes beyond their ability to continue. In other words, no instantaneous failure of any nature can be tolerated. I am thinking especially of structural failures and brake fluid boiling. I believe the driver has to be warned.

Sintered Metallic Linings Found Suitable Friction Material for Brakes

—H. M. T. Harris
American Brake Shoe Co.

FROM the information at hand, it would appear that conventional drum brakes, similar to those which we know today, will be with us for a number of years yet. More is being done continuously on development of friction materials for use on these brakes. Selection of friction material for the drum brake has often been a compromise from the standpoint of fade resistance and heat stability in order to meet other brake requirements. A more direct approach is the selection of friction material with exceptionally good fade resistance and the engineering of the brake and drum to take advantage of this important characteristic. R. E. Spokes has suggested that "much can be gained from experience in improving liners for heavy duty application." He points out that more can be done in the selection of minerals as friction-enhancing agents if the drum and brake lining are considered as a friction pair, and thus attention is also given to the metallurgy of the drum.[a]

[a] "New Lining Materials," by R. E. Spokes. Presented at SAE Golden Anniversary Passenger-Car, Body, and Materials Meeting, Detroit, March 3, 1955.

Table A—Thermal Capacity of Present-day Brake

Stop No.	Average Peak Drum Temperature (Thermocouple per SAE Specification)	
	Composition Brake Lining, F	Sinter-Metallic Brake Lining, F
1	497	403
2	626	580
3	738	710
4	830	793
5	867	862

In addition, possible use of sintered metallic brake linings should be included in reengineering of the drum brake. Continuous progress is being made in the improvement of the frictional stability of this type of material which can be made fadeproof. Improvement of thermal capacity of a present-day brake when using this lining is shown in Table A of 100-mph stops made at 1½-min intervals on a dynamometer.

From the above, it is apparent that the increase of thermal capacity of the sintered metallic linings is an advantage for the first few stops. Beyond that, the entire mechanism is saturated with heat.

In conclusion, although it is recognized that a tremendous effort and cost goes into any brake change, it is believed that the above is the immediate answer to safer brakes.

High-Temperature Problems Still Remain for Brake Engineers' Solution

—J. T. Ball
Chrysler Corp.

ALTHOUGH brake performance under high energy input has been proved over the years, high brake temperatures still remain and will remain the basic problem of the brake engineer. The increased use of automatic transmissions, greater car weights, more freeways, and many other factors have had an adverse affect on braking systems. Car activity, as an example, has been greatly increased. Within the past 10 years the time required for a car to accelerate to 70 mph has been approximately halved, greatly decreasing the cooling time of the brakes. Also it can be assumed readily that the adverse factors affecting braking systems will continue to multiply.

Generally speaking, there has not been too much difficulty in obtaining the required brake output during one stop. However, for a series of high-speed stops or under conditions of prolonged heat input lining fade, drum expansion, stability, and vapor lock have become serious problems.

Regardless of disc or drum design, *mass* is the prime requisite for heat energy storage during a single stop. Under the steady state or prolonged heat input conditions, cooling or heat dissipation is the basic consideration.

While most of the problems concerning wheel brakes consist mainly in means of decreasing lining fade and increasing pedal reserve, some of the other characteristics which become apparent in the brake system under very high temperatures are just as serious. For example, the customer is just as conscious of brake instability as he is of lining fade. The noise problem must not be underestimated. Excessive lining wear, damaging of rubber parts in the brake system, and brake drum spotting, while not as evident to the average driver, are certainly items which must be corrected. Brake fluid vapor lock and the required raising of the boiling point bring out another problem caused by higher brake temperatures.

Cost will be a very important factor in the design of a disc brake since the complexity will be greater as well as the requirement for more accurate machining.

The job of the brake engineer remains, therefore, to challenge the laws of physics by putting bigger and better brakes into hotter and smaller locations.

THIS paper describes the operational and performance characteristics which will make the DC-8 jet transport capable of economical operation over both long- and short-range routes of reasonably high traffic density.

It is shown, for example, that the effects of operation with one or two engines inoperative, the effect of high ambient air temperatures, and effects of flying at nonoptimum altitudes, while different from reciprocating-engine aircraft, will permit easy integration of the jet transport into existing airline operations.

Furthermore, the jet transports will have such short take-off distances when carrying fuel sufficient for short and moderate ranges that most existing airport runways, now used for DC-6B and DC-7 operation, are of sufficient length for jet-transport operation.

THE . . . VERSATILE

SINCE the first appearance of the jet engine on the aviation scene, there has been a great tendency to regard the jet airplane as a special purpose vehicle. The special purpose to which it was allegedly limited has varied with time. In the earliest days, the jet airplane, particularly if applied to transport work, was supposed to be good only for short ranges because jet engines had a very high fuel consumption. As jet-engine fuel consumption decreased and as aerodynamic and structural knowledge necessary to build airplanes capable of efficient flight at high subsonic Mach numbers was acquired, the range ability of the jet airplane rapidly improved. In the last few years it has been shown that the jet airplane is capable of flying economically the longest airline routes in the world. It then became fashionable to assume that the jet was economically efficient only when it flew at very high speeds for long ranges. This type of operation led to an overall block speed almost as high as the cruising speed because the cruising portion comprised almost the entire flight. The slower parts of the flight, like the climb, descent, and maneuvering around the airport, were trivial compared to the long cruise period. It allegedly followed that the efficiency of the high-speed jet airplane suffered greatly at lesser ranges and that anything under 1000 or 1500 miles fully belonged to the field of slower turboprop aircraft.

Early studies of jet-transport characteristics indicated that special operational practices would be necessary because of severe range losses associated with nonoptimum cruising altitudes, partial engine operation, and nonstandard temperature at the cruising altitudes. The impressions left by these studies have not yet been erased.

Recent studies have shown that the jet transport as it has finally evolved in the Douglas DC-8 retains its long-range speed and cost advantages at both moderate and short ranges. Furthermore, the jet transport is actually quite flexible with regard to cruise altitude, temperature, and partial engine operation. The purpose of this paper is to illustrate the present operational characteristics of the jet transport at short, medium, and long ranges. Some of the changes in the engines and aerodynamic characteristics of the airplane which have contributed to the excellent performance of the jet transport will be indicated.

Early Limitations on Range

First, let us discuss the reasons why the jet was limited in its earlier development. The primary problem has always been fuel consumption. The basic fuel flow of the jet transport was high even at its most favorable altitudes which were from 35,000 to 45,000 ft. At lower altitudes the specific fuel consumption was much higher. Furthermore, when the maximum endurance flight condition was studied to establish holding fuel requirements, we found that the specific fuel consumption at low percentages of maximum thrust was much higher than at the maximum thrust. So it was necessary to start with a basically high fuel consumption at optimum cruise altitudes, increase it markedly at lower altitudes, and then add a further large increase at reduced thrust.

Another problem was the large loss in thrust when temperature was above standard. This loss in thrust would reduce both the speed and range of a jet airplane equipped with engines just large enough for optimum standard day operation. Since the average

JET ...TRANSPORT

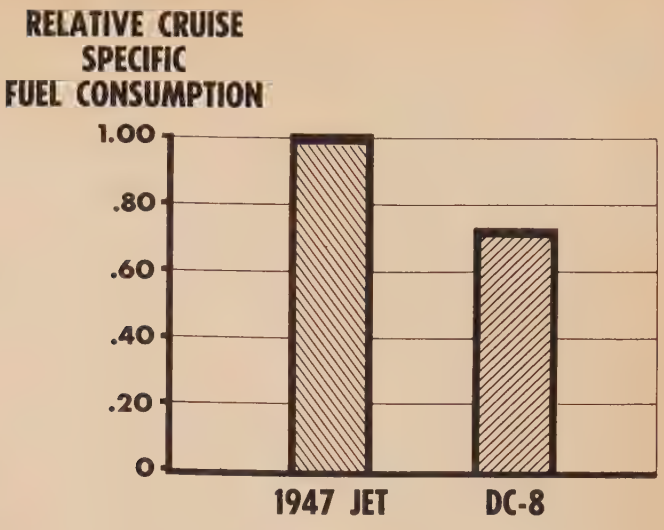

Fig. 1—Comparison of relative cruise specific fuel consumption 1947 jet and DC-8

R. S. Shevell, Douglas Aircraft Co., Inc.

The paper was presented at the SAE National Aeronautic Meeting, New York, April 11, 1956.

temperatures at cruise altitude are above standard over many of the airline routes, this condition needed careful study.

Let us examine some of these items that affect jet-transport performance, see how they have changed with time, and study the comparable items for reciprocating-engine aircraft. We will use for comparison a study of a jet transport carried out in 1947.[1] For our modern jet we will use the DC-8. The DC-8 will be available with several types of engines, among them the Pratt & Whitney J-57 engine, the larger Pratt & Whitney J-75 engine, and the Rolls Royce Conway engine. We will refer mostly to the DC-8 with J-75 engines. However, where the J-57 version has significantly different characteristics, we will mention these differences. The Conway version will be quite similar to the J-75-powered airplane. The DC-6B and the DC-7, or DC-7C, will be shown as representative of the most economical and the fastest of present-day airplanes, respectively.

Long-Range Ability

The first parameter deserving study is the general level of cruising specific fuel consumption. Fig. 1 shows the relative cruising specific fuel consumptions of our 1947 jet-transport study airplane and the DC-8. We can see that there has been a 38% improvement in specific fuel consumption. Another factor that enters into the range efficiency of the jet is the speed. The airplane in our 1947 jet-transport study had a cruising speed of 428 knots compared to the 480-knot cruising speed of the DC-8. The range of a jet airplane is given by the equation:

$$R = \frac{V}{c_j}\frac{L}{D}\log_e\left(\frac{1}{1-\frac{W_F}{W_{TO}}}\right)$$

where:

R = Range, nautical miles
V = Airplane speed, knots
c_j = Specific fuel consumption, lb/lb/hr
L/D = Ratio of airplane lift to drag at cruise condition
W_F = Fuel weight, lb
W_{TO} = Take-off weight, lb

For a given ratio of fuel weight take-off weight, the range depends on $\frac{V}{c_j}\frac{L}{D}$, called the range factor. $\frac{V}{c_j}$ may be considered as a measure of the overall

[1] *Journal of the Aeronautical Sciences*, Vol. 15, March, 1948, pp. 133–143: "Operational Aerodynamics of High-Speed Transport Aircraft," by R. S. Shevell.

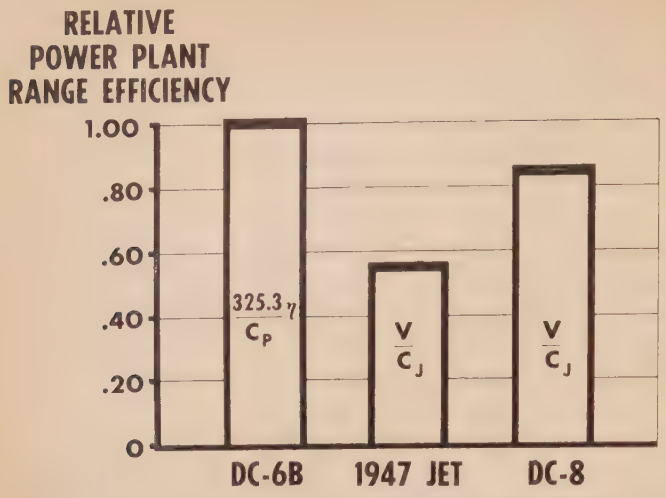

Fig. 2—Comparison of relative powerplant range efficiency in DC-6B, 1947 jet, and DC-8

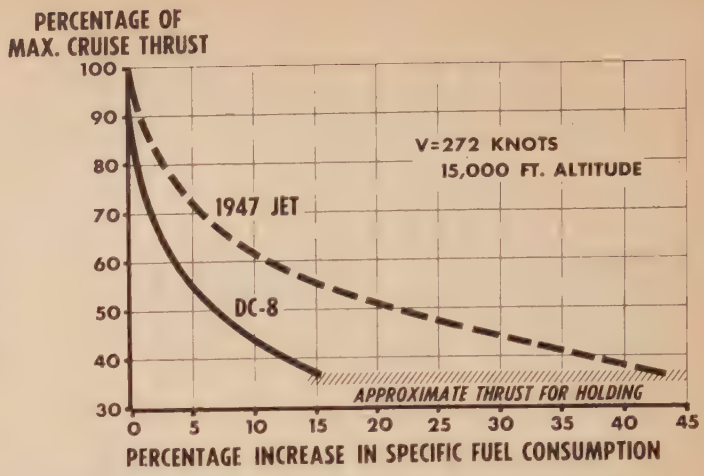

Fig. 4—Effect of reducing cruise thrust on specific fuel consumption

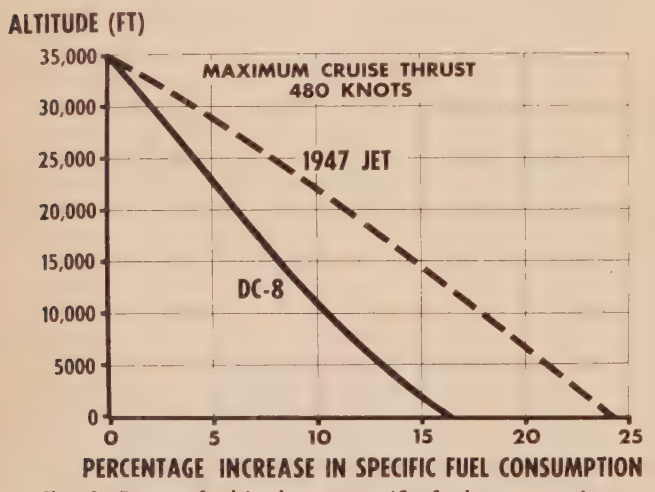

Fig. 3—Effect of altitude on specific fuel consumption

efficiency of a jet engine, while $\frac{L}{D}$ is a measure of the aerodynamic efficiency of the airframe.

The comparable range equation for the propeller-driven airplane is,

$$R = 325.3 \frac{\eta}{c_p} \frac{L}{D} \log_e \left(\frac{1}{1 - \frac{W_F}{W_{TO}}} \right)$$

where:

η = Propeller efficiency
c_p = Specific fuel consumption, lb/bhp/hr

A basic range efficiency comparison between a jet engine and a reciprocating engine may be made by comparing the terms $\frac{V}{c_j}$ for the jet with $325.3 \frac{\eta}{c_p}$ for the reciprocating airplane. Fig. 2 shows a comparison of this factor for the DC-6B, the 1947 jet, and the DC-8. It is seen that the jet value has improved from 55% of the DC-6B value in the 1947 design to 85% in the DC-8. The DC-8 has about the same ratio of lift to drag as the DC-6B so that these figures also represent the relative range for a given ratio of fuel weight to take-off weight. The improvement in this range factor during the development period of the jet transport has been 55%.

Although the value of the range factor for the DC-8 is still 15% less than for the DC-6B, the DC-8 is able to fly much longer ranges than the DC-6B because of its higher fuel-weight/gross-weight ratio. This ratio is about 42.2% for the DC-8 with full over-water coach payload compared to about 25.5% with full coach payload for the DC-6B. Although it is not the purpose of this paper to discuss the structural and weight factors that make this possible, it can be said that roughly half the gain is due to the reduced percentage of take-off weight absorbed by the weight of jet engines compared to the ratio of weight of reciprocating engines with propellers to take-off weight. The remainder of the gain in fuel-weight/gross-weight ratio is due to many complex interacting factors including increased take-off wing loading, availability of large wing fuel volume which permits increasing fuel weight with a relatively small structural weight penalty, and the fact that the weights of many items of an airplane do not increase as rapidly as the gross weight.

Not all the fuel carried by an airplane can be used for covering the scheduled distance since a substantial quantity of reserve fuel must be carried. This reserve quantity is large for a jet. The fuel used for holding over a weathered-in airport is an important part of the reserve. Although the jet airplane has attained a reasonable fuel consumption per mile, it still retains a high fuel flow per hour. If the jet airplane is flown slowly at low altitude, the fuel flow per hour is as great as or greater than it is during

long-range cruise at high speed and altitude. Therefore, holding fuel might be something of a problem.

As altitude is reduced, the thrust available from the engine increases. At low altitudes very low percentages of rated thrust and engine speed are required to maintain flight and the specific fuel consumption increases. It was once thought necessary to hold at high altitudes, or perhaps to use only two or three engines so they could be operated at a higher percentage of rated thrust. Although the trend toward higher specific fuel consumptions still exists at low thrust and altitudes, the modern engine has much more versatility in this respect than its 1947 ancestor. Fig. 3 shows a comparison of the variation with altitude of specific fuel consumption for the DC-8 engine and for the 1947 jet engine. The effect of altitude is much less in the modern engine. Similarly, Fig. 4 shows that the increase in specific fuel consumption of the DC-8 powerplant as the thrust is reduced is much less than for the earlier engine. The result is that while the holding fuel required still increases as altitude and engine speed are reduced, this characteristic is much more favorable than in early studies. Fig. 5 shows how holding fuel per hour for the DC-8 varies with altitude. An increase of 15% results from reducing the holding altitude from 30,000 to 15,000 ft, and another increase of 18% in going to sea level. Although it is always preferable to have the jet airplane hold at high altitude, it is by no means essential.

Nevertheless, the quantity of holding fuel required per hour is still large. On the DC-8 with J-75 engines, 9000 lb per hr are required for holding at an altitude of 15,000 ft. The DC-6B requires 1420 lb per hr for holding at 10,000 ft. A typical total reserve fuel requirement is 1-hr holding plus 260 nautical miles of cruising. For the jet, 260 miles includes a climb from sea level to 30,000 ft, cruising at 30,000 ft with long-range procedure, and a descent to sea level. The 260-mile flight for the DC-6B includes climbing to 10,000 ft and flying with long-range procedure at that altitude. The DC-8 reserve fuel is 16,400 lb and the DC-6B is 3800 lb.

The reserve fuel problems of the jet transport are solved by designing the airplane to take off and carry the additional reserve fuel. To evaluate the magnitude of this requirement, the reserve fuel can be examined as a percentage of the take-off weight. The reserve fuel is 5.7% of the maximum take-off weight for the DC-8 and 3.6% of the maximum take-off weight for the DC-6B. Although the jet requirement is larger, the difference is only 2% of the take-off weight. When viewed in terms of the total lifting ability of the jet transport, it is clearly just another design requirement. It is true, certainly, that extended holding requirements will have more severe effects on the jet than on reciprocating-engine airplanes. The jet can carry sufficient reserve for all normal requirements, but improved landing aids may be required to avoid increased diversions to alternate airports. It does not seem too much to expect that by the time these airplanes are in regular operation, progress will have been made in this direction.

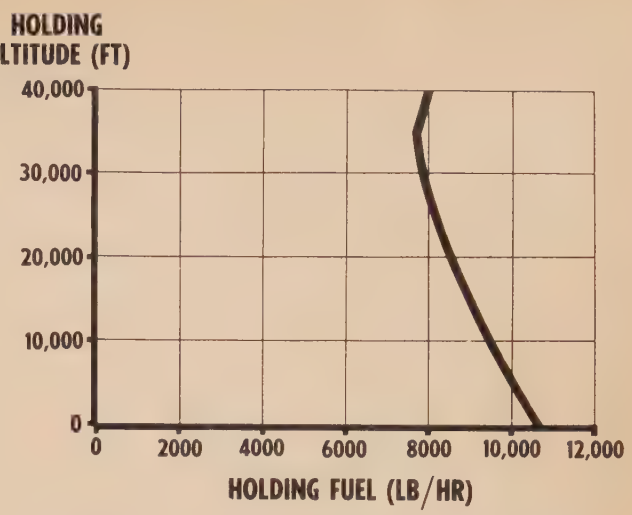

Fig. 5—Effect of altitude on holding fuel in DC-8 with J-75 engine

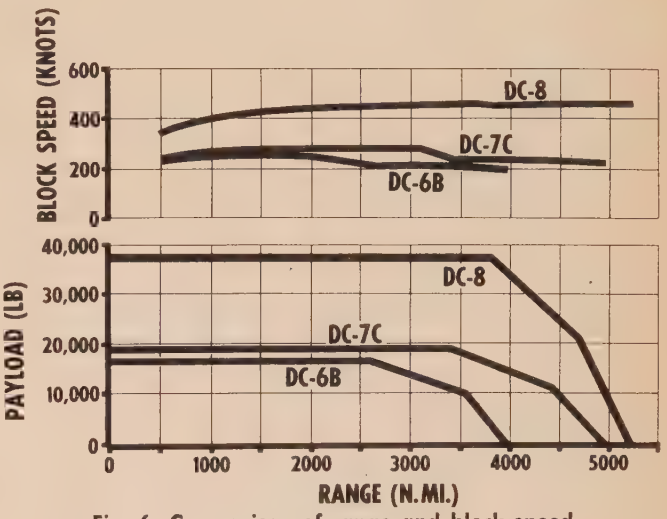

Fig. 6—Comparison of range and block speed

The overall effect of the aerodynamic range, fuel-weight/gross-weight ratio, and reserve fuel factors is seen in Fig. 6 which compares the range with coach interior of the DC-8 with that of the DC-6B. The DC-7C, long-range successor to the DC-6B, is also shown to indicate the greatest capability shown by reciprocating-engine aircraft of the DC-7 series. The DC-8 has a range with full payload exceeding that of its predecessors. The block-speed advantage is about 160 to 190 knots. It is clear that the long-range problem of the jet transport is well in hand.

Effect of Cruising Altitude

Another well-known basic characteristic of the jet airplane is that high-altitude flight is necessary to obtain both the low drag and the high speed that lead to efficiency. An airplane designed for long range

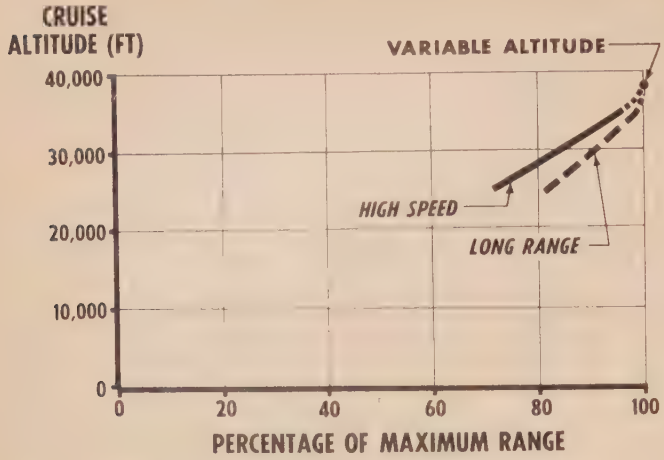

Fig. 7—Effect of altitude on maximum range of DC-8 with J-75 engine

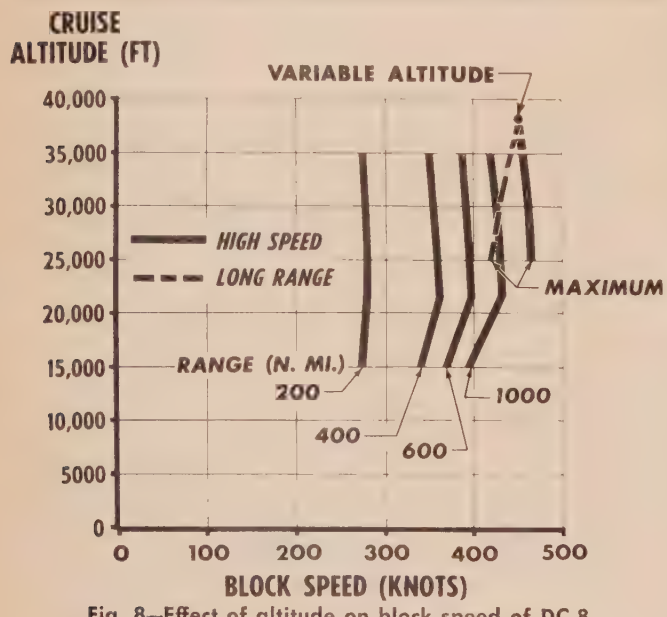

Fig. 8—Effect of altitude on block speed of DC-8

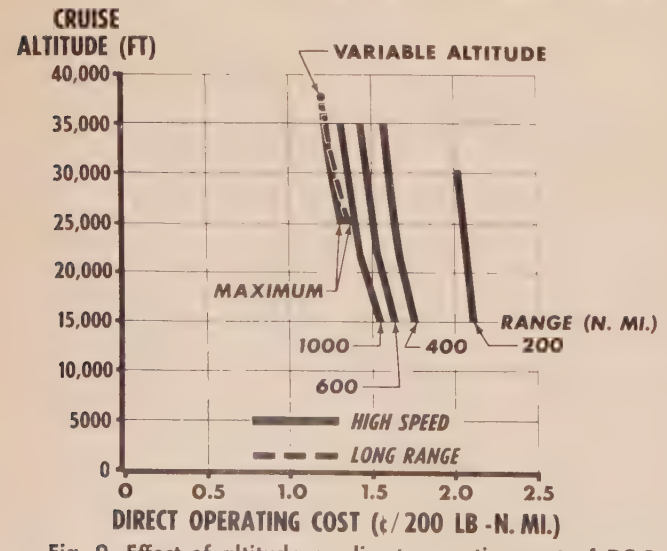

Fig. 9—Effect of altitude on direct operating cost of DC-8

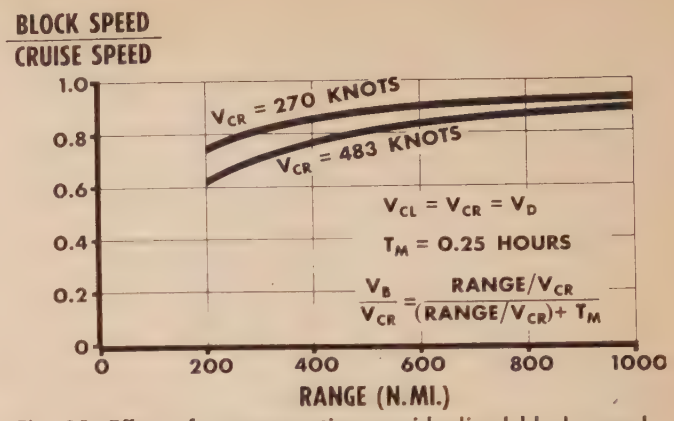

Fig. 10—Effect of maneuver time on idealized block speed

must fly at an angle of attack that gives close to its maximum lift/drag ratio. With a given angle of attack, airplane weight, and a given altitude, the speed is automatically fixed. If the airplane is to go faster, it must go higher to less dense air to maintain the same angle of attack and, consequently, the desired lift/drag ratio.

The maximum range will be obtained by increasing the altitude and thus the speed until the drag divergence Mach number is reached. This is the Mach number above which wasteful increases in drag due to compressibility are incurred. With wing loadings required by commercial take-off and landing field length requirements, the altitudes at which the best lift/drag ratios can be obtained at the drag divergence Mach number vary from about 28,000 to 42,000 ft. The optimum altitude will vary as the weight of the airplane changes during the flight.

When maximum range is a prime requirement, it is necessary to fly fairly close to the optimum altitude in order to get the coincidence of maximum speed limited by the drag divergence Mach number and the optimum angle of attack. It turns out, however, that moderate deviations of 2000 to 5000 ft from the optimum altitude are not too serious, even for the long-range operation. In Fig. 7 we see the difference between the maximum range variable altitude flight path operation, in which the airplane is allowed to change altitude as it burns fuel, and various constant altitude operations. The loss of range is not too serious if the constant altitudes are selected close to the mean altitude of the variable flight path. Other studies have shown that using step-altitude operation, in which we fly at a constant altitude for the first portion of the flight and then change to another altitude, the range is within 1 to 2% of the maximum range shown with optimum variable altitude. If we attempt extreme deviations in altitude from the optimum, we do get severe losses in range. As Fig. 7 shows, if we drop to 25,000 ft and maintain high-speed flight, the loss in range is 28%. The speed increases at the lower altitude. The range loss can be reduced at the low alti-

tudes by reducing speed to the long-range speed at each constant altitude as shown by the dashed line. At 25,000 ft, for example, the range at the long-range speed is 18% less than the optimum. It can be concluded that the jet transport can tolerate moderate deviations from the optimum altitude even when it is engaged in rather long-range operations.

At shorter ranges fuel consumption is important only in so far as it affects operating costs. The take-off distances of the jet transport are quite low at the light gross weights required for short and moderate ranges so that the fuel load is not important in this respect. At lower altitudes we will incur significant fuel consumption increases. The effects of altitude on block speed and direct operating costs have been studied at ranges of 200 to 1000 nautical miles as well as at maximum range.

The block speed effects are shown in Fig. 8. The effect of altitude on speed is relatively minor. The curves are based upon cruising at the drag divergence Mach number except for the dashed curve representing long-range operation at each altitude. The block speeds are improved at the lower altitudes with high-speed operation. At altitudes below 20,000 ft the cruise speed is restricted by the structural placard speed and the block speed is reduced. The long-range operation curve shows the speed loss suffered if maximum range must be attained at the lower altitudes.

In Fig. 9 altitude is shown to have a relatively small effect on direct operating costs. These costs, calculated from the 1955 ATA cost equations,[2] are based on a domestic first-class interior. The increased block speeds at lower altitudes tend to counteract the additional fuel cost. The shorter the range, the less the increase in cost resulting from reducing the cruise altitude. At 400 nautical miles range the cost is increased only 7% by flying at 20,000 ft instead of 35,000 ft while the speeds are increased 2%. At ranges close to maximum range, altitude affects operating cost to a larger extent; but even when the altitude is reduced from the optimum to 30,000 ft, the effect on cost is only 5% with high-speed operation. This asumes, of course, that full payload can be carried at all altitudes. If the payload must be reduced to fly the range at the reduced altitude, then the effects on costs are very large. An additional penalty of about 2% is incurred even if full payload is carried when long-range operation must be used at the low altitudes. It can be concluded that the jet transport can tolerate large altitude variations when fuel is not critical. At short ranges cruise altitude has very small effects on block speed or cost.

As a result of these considerations we can conclude that when the jet transport is used for short-range operation, it has great flexibility with regard to altitude. Contrary to early concepts it does not have to dash up to 40,000 ft, but it may be scheduled at any altitude between 20,000 and 40,000 ft according to the dictates of wind, weather, and the requirements of airways traffic control.

Effect of Range on Block Speeds and Operating Costs

How does the basic operating cost of the jet airplane, and particularly the variation of this cost with range, compare with present aircraft? The major characteristic that tends to reduce the jet's efficiency for short range may be seen by studying the following simplified equation for direct operating cost:

$$\$/\text{mile} = \frac{(\$/\text{hr})_{\text{fixed}} + (\$/\text{hr})_{\text{fuel}}}{\text{block speed}}$$

The basic cost per hr consists essentially of two parts. The first is a constant since we assume that for a given airplane the cost per hr for engine and airframe maintenance and depreciation, crew pay, and insurance are independent of the speed. The second part, the fuel cost per hr, does vary with speed and altitude. However, as a first approximation, it might be said that the fuel cost per hr does not change a great deal from one flight condition to another. It is definitely not a large change compared to the total hourly cost, so that the numerator at the first approximation is a fixed number. The cost per mile is then dependent on the block speed.

Block speed, the overall speed from leaving the loading gate to arriving at the gate at the destination, includes certain fixed time losses due to taxiing, take-off, maneuvering around the airport, landing, and certain regions of reduced velocity in climb and descent. On a very long flight the maneuvering time and the time spent in climb and descent are very small compared to the cruise time and have a relatively small effect on the block speed. The block speed for a very long flight approaches the cruising speed. On a very short flight the 15-min time usually assumed for the maneuver, take-off, taxi and landing may be a large part of the total flight time and is completely wasted time in so far as covering distance is concerned. The effect on reducing the block speed is very large. From the equation we can see that the cost per mile will be much increased. This is characteristically true of all airplanes. But with a faster cruising speed, the basic flight time between two cities is shorter, and the percentage of total time that must be attributed to this maneuvering time is larger. Furthermore, there is a question of whether the climb and descent have a greater influence on the block speed of a fast airplane than on a slow one.

Some insight into the variation of block speed with range can be gained by a brief analysis. Defining:

T_B, D_B, and V_B = Block time, distance, and speed
T_M = Maneuver, taxi, take-off, and landing time
T_{CL}, D_{CL} and V_{CL} = Climb time, distance, and speed

[2] *Air Transport Association*, 1955: "Standard Method of Estimating Direct Operating Costs of Transport Airplanes."

T_{CR}, D_{CR} and V_{CR} = Cruise time, distance, and speed

T_D, D_D and V_D = Descent time, distance, and speed

Then:

$$T_B = T_M + T_{CL} + T_{CR} + T_D$$
$$D_B = D_{CL} + D_{CR} + D_D$$
$$V_B = D_B/T_B$$

Since:

$$V_B = \frac{D_B}{T_B} = \frac{D_{CL} + D_{CR} + D_D}{T_B} = \frac{V_{CL}\,T_{CL} + V_{CR}\,T_{CR} + V_D\,T_D}{T_B}$$

it can be shown that:

$$\frac{V_B}{V_{CR}} = \left(\frac{V_{CL}}{V_{CR}}\right)\left(\frac{T_{CL}}{T_B}\right) + \frac{T_{CR}}{T_B} + \left(\frac{V_D}{V_{CR}}\right)\left(\frac{T_D}{T_B}\right) \quad (1)$$

In an ideal case in which $T_M = 0$, and $V_{CL} = V_{CR} = V_D$, $\frac{V_B}{V_{CR}} = 1.0$.

By making certain simplifications it is possible to establish the relative effect of the various terms in equation (1). Suppose we assume an idealized case where $V_{CL} = V_{CR} = V_D$. Then equation (1) becomes

$$\frac{V_B}{V_{CR}} = \frac{T_{CL} + T_{CR} + T_D}{T_B} = \frac{T_B - T_M}{T_B} = \frac{\frac{D_B}{V_{CR}}}{\frac{D_B}{V_{CR}} + T_M} \quad (2)$$

This equation, which shows the importance of maneuver time on V_B, has been evaluated for two cruising speeds in Fig. 10. The usual value of T_M = 0.25 hr has been used. The cruise speeds were chosen to represent the DC-6B at 15,000 ft and the DC-8 at 35,000 ft, altitudes which are close to optimum for short-range operating cost. The percentage loss in block speed due to maneuvering time is greater for the higher speed airplane at any given range. At 400 miles, for example, the DC-8 loses 10.5% more in block speed due to this term than does the DC-6B.

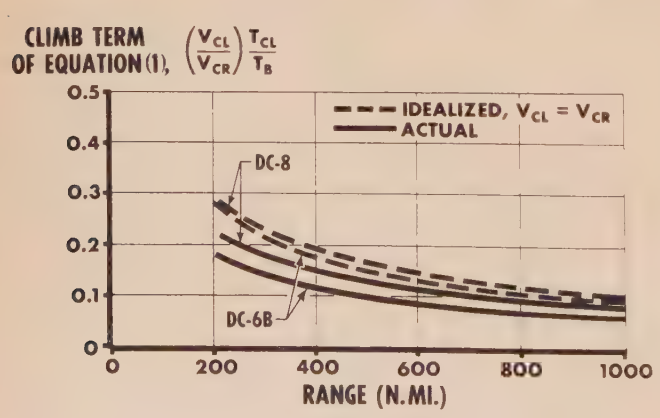

Fig. 11—Effect of range on climb term of equation (1)

If we evaluate equation (1) for a particular airplane and range, each of the three terms has a particular value. If V_{CL} is less than V_{CR}, or if V_D is less than V_{CR}, V_B/V_{CR} will be diminished. If $V_{CL} = V_D = V_{CR}$, only the maneuver time will adversely affect block speed. To determine the relative importance of climb and descent on block speed, the actual values of each term of equation (1) have been compared to the idealized value it would have if $V_{CL} = V_D = V_{CR}$. Values of T_{CL}, T_{CR}, and T_D are the calculated values for each airplane.

Fig. 11 shows both the actual and idealized ($V_{CL} = V_{CR}$) values of the climb term plotted against range for both the DC-6B and DC-8. Both airplanes lose block speed due to climb since the actual value of $\frac{V_{CL}}{V_{CR}}\left(\frac{T_{CL}}{T_B}\right)$ is less than $\frac{T_{CL}}{T_B}$, the value when $V_{CL} = V_{CR}$. At 400 miles, for example, the DC-6B has a value of $\left(\frac{V_{CL}}{V_{CR}}\right)\left(\frac{T_{CL}}{T_B}\right)$ of 0.113 compared to $\frac{T_{CL}}{T_B} = 0.175$, a loss of 0.062. This is a decrease of 7.2% of the value of $\frac{V_B}{V_{CR}}$ which would occur if $V_{CL} = V_{CR} = V_D$. The comparable number for the DC-8 is a loss of 5.2% due to the climb. Thus the DC-8 loses less, by per cent, in the climb segment than the DC-6B by 2%.

The descent portion shows an opposite effect. Fig. 12 shows the actual and ideal ($V_D = V_{CR}$) values of the descent term of equation (1) plotted against range. Since the DC-6B has a descent speed greater than the cruise speed, the DC-6B actual block speed is improved by the existence of a descent as compared to continued cruising. The DC-8 descent speed is limited by a structural placard to speeds below the cruising speed at the lower altitudes. At 400 miles, the DC-6B has a value of $\frac{V_D}{V_{CR}}\left(\frac{T_D}{T_{CB}}\right)$ of 0.085 compared to $\frac{T_D}{T_B}$ of 0.079. This is an increase of 0.7% in the value of $\frac{V_B}{V_{CR}}$ corresponding to the reference condition with $V_{CL} = V_{CR} = V_D$. The DC-8 shows a decrease in $\frac{V_B}{V_{CR}}$ of 1.9% due to descent and thus loses more in descent than the DC-6B by 2.6%.

It should be noted that the influence of T_M is strictly dependent on total block time, not on range. The hypothetical case with constant speed is studied here with the same block time as the actual case. The range if the speed were constant in all segments would be slightly greater than 400 miles. The basic significance of the comparison is not appreciably changed, however.

These comparisons can be summarized as follows:

1. The DC-8 has a greater decrease in block speed due to descent than does the DC-6B, but the effect of climb is more favorable to the jet. The total effect of climb and descent on block speed is the same for

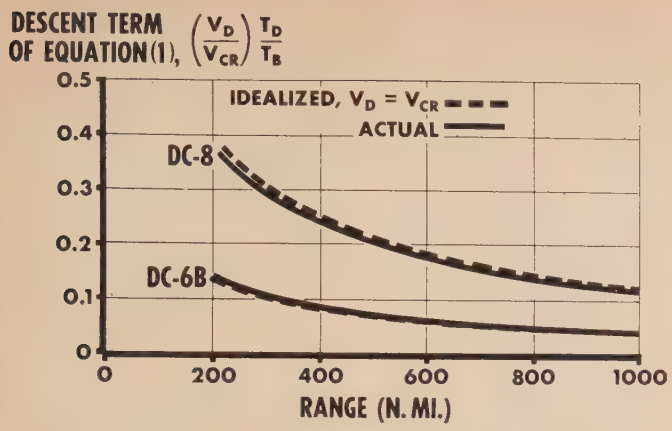

Fig. 12—Effect of range on descent term of equation (1)

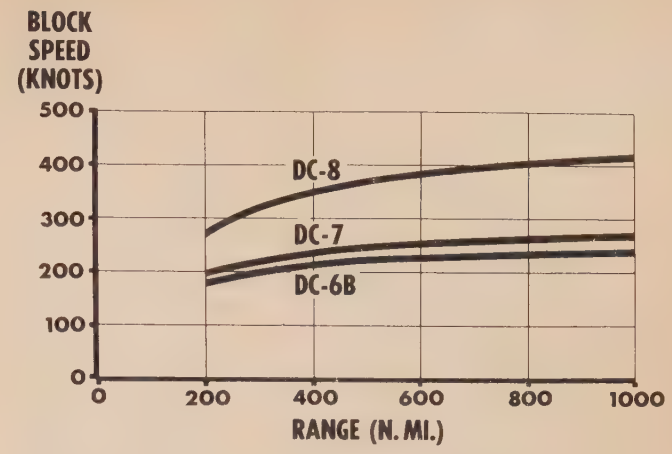

Fig. 13—Comparison of block speeds

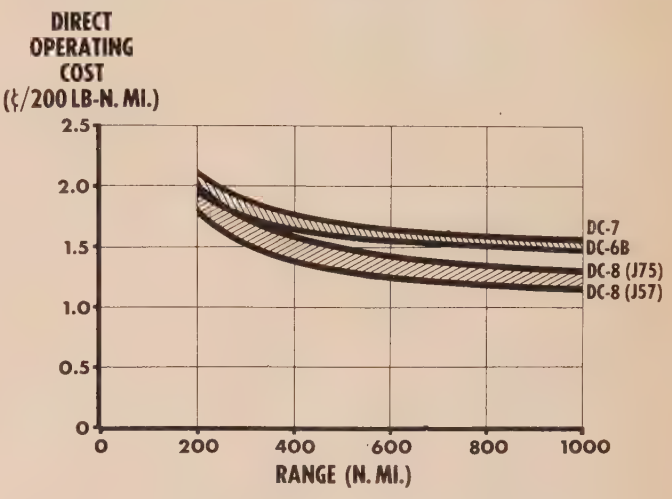

Fig. 14—Comparison of direct operating costs

the DC-6B and the DC-8. At 400 miles range the effect on both types of airplanes is a block-speed reduction of about 7%.

2. The larger percentage loss in block speed at short range associated with high-speed airplanes is entirely due to the maneuvering time loss on and around the airport.

3. Since maneuvering time has a greater effect on high-speed airplane performance, improvement in airport traffic control and check-out practices, will be more beneficial to jet transports than to present-day airplanes. Similarly, delays of this kind will be more costly.

The final effect on the block-speed comparison of the DC-8, the DC-6B, and the DC-7 is shown in Fig. 13. The DC-8 maintains a large speed advantage down to very low ranges although the advantage at shorter ranges is smaller, of course, than at longer ranges.

The direct operating costs are compared for these airplanes in Fig. 14. Both the J-57 and J-57 powered DC-8 airplanes are shown. The 1955 ATA equations are used for the cost calculations. Engines and airframes are depreciated over a 7-year period except for the DC-8 airframe for which a 10-year depreciation period has been used. Engine overhaul periods assumed are 750 hr for the DC-8, 1000 hr for the DC-7 and 1250 hr for the DC-6B. Payloads are based on first-class interiors. The DC-8 carries 118 passengers and 32,970 lb of total payload while the DC-6B carries 58 pasesngers and 15,010 lb of total payload.

The engine maintenance cost equations, based on engine weight and first cost, show the J-75 DC-8 to have a considerably higher operating cost than the J-57 version based on equal overhaul periods. Because the larger J-75 engine will operate at a relatively low percentage of its rated thrust, it may demonstrate a longer period between overhaul. For this reason the J-75 may not actually have the cost increase attributed to it by the ATA cost equations.

The cost curves show that both jets remain cheaper than the most economical reciprocating-engine airplane at all ranges down to 280 nautical miles. At 1000 miles range, the J-75 DC-8 is 12% cheaper than the DC-6B and 16.5% cheaper than the DC-7. The J-57 DC-8 has an additional cost advantage of 11% giving a total cost advantage over the DC-6B of 22% and over the DC-7 of 26.5%.

It is not intended to suggest that the DC-8 is the ideal short-range airplane. The DC-8 has been designed with a gross weight and take-off ability to carry fuel for the intercontinental routes in the overwater version and transcontinental nonstop operation in the domestic version. These data show, however, that where the passenger traffic will justify an airplane the size of the DC-8, a large jet transport can be economically operated over almost any existing airline route. If we were to concentrate on a short-range airplane powered by jet engines, the operating costs could be improved even further in an airplane of DC-8 size. A smaller jet transport designed specifically for short routes could have operating costs approaching those of the DC-8. The gains in operating costs accruing from designing for a particular range

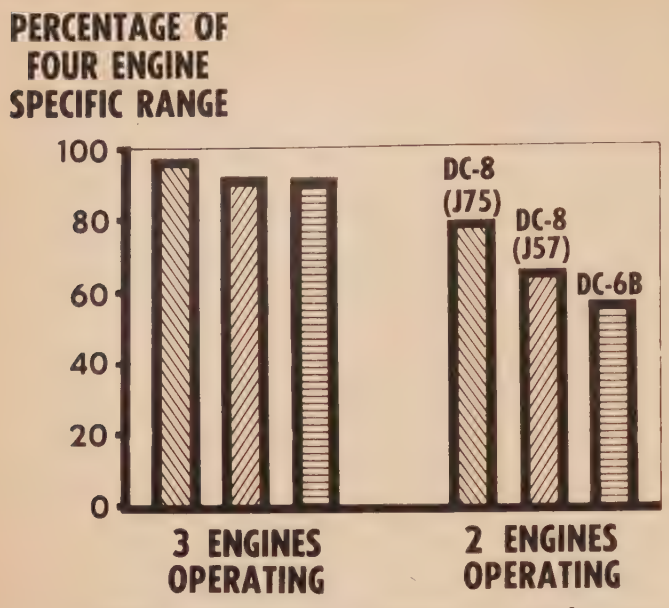

Fig. 15—Effect of partial engine operation on specific range

will tend to balance the higher costs inherent in smaller airplanes.

Partial Engine Operation

The problem of partial engine operation on a jet transport has long been viewed with some trepidation. Operation at less than the desired total cruise thrust can be achieved only by decreasing the altitude to increase the available thrust. Low-altitude, reduced-speed operation of a jet after engine failure once appeared as a serious threat to the safe overwater operation of such an airplane.

Fortunately, a modern design does not suffer particularly in this manner. By selecting the engine with more than adequate cruise thrust, under standard conditions, we can increase the power per engine when only three engines are operating. In the case of the overwater version of the DC-8 with J-75 engines, the three-engine speed and altitude with maximum continuous thrust are only slightly less than the normal four-engine speed and altitude. The altitude loss is 800 ft, and the speed decrease is less than 2%. A small increase in specific fuel consumption occurs at this higher power rating, and a small penalty results from the drag due to the rudder and aileron deflection necessary to counteract the asymmetric thrust. The loss in specific range is only 3% as shown in Fig. 15.

With two engines inoperative, the two remaining engines at maximum continuous thrust cannot drive the airplane nearly as high and as fast as four engines with normal cruise thrust. In addition, if two engines fail on the same side of the airplane, there is considerably greater rudder drag. The loss in altitude with two engines at maximum continuous thrust compared to the normal long-range cruising with all engines operating is about 14,000 ft. The speed decreases 108 knots or 23%, and the loss of specific range is 19%.

Fig. 15 also shows a comparison of these partial engine characteristics with those of a DC-6B. The DC-6B effects depend to some extent on the weight and altitude combination studied. An average cruising weight has been used at 10,000-ft altitude for the DC-6B three-engine operation and 5000 ft for the two-engine operation. It is clear that the jet is actually superior for partial engine operation. With three engines operating, both airplanes show about the same specific range decrease. With two engines operating, the jet loses only 19% whereas the DC-6B would lose 43%. The DC-6B shows such a large effect because two-engine operation requires auto-rich powers even at the low cruise altitude assumed.

The DC-8 with the smaller J-57 engine will show somewhat greater losses in range due to partial engine operation especially in the two-engine case where the speed loss is severe. The reduction in specific range will be about 8% for three-engine operation and 34% for two-engine operation. Even this large loss is more favorable than the conventional piston engine characteristics of the DC-6B.

Temperature Effects on Cruising

Variations of ambient temperature can have an important effect on the long-range cruising performance of an airplane. Fig. 16 shows the variation of specific range, or miles per pound of fuel, and cruise speed with temperature for the long-range condition for the DC-8 equipped with either the J-75 or the J-57 engines and for the DC-6B.

The reciprocating-engine airplane has been assumed to be flying at its long-range condition at 10,000-ft altitude. Under this condition the maximum cruise power is not usually being used. The power loss due to temperature has only the small effect of requiring a higher cruise engine speed and a slightly

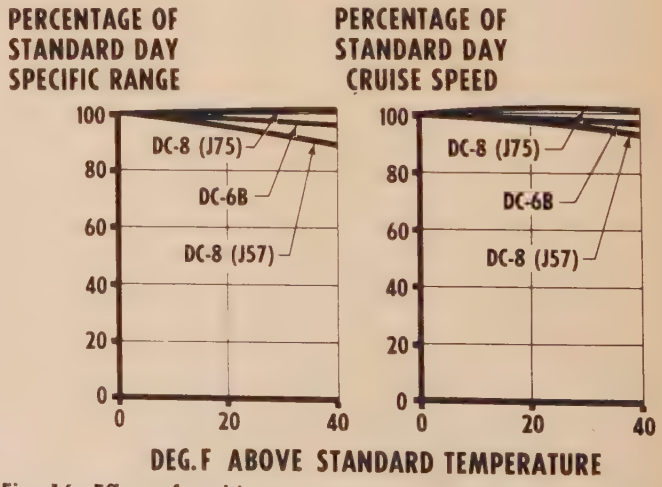

Fig. 16—Effect of ambient temperature on long-range cruising

higher specific fuel consumption. The additional drag associated with the increased cooling required reduces the speed and the range, however.

The DC-8 with the J-75 is not thrust limited but flies under standard temperature conditions with considerably less than the maximum cruise thrust. Therefore, although the maximum cruise thrust is reduced on a hot day, there is still sufficient thrust to fly at the same desirable Mach number up to between 20 and 30 F, above the standard atmospheric temperature. For the long-range case with variable altitude, the airplane would fly at the same angle of attack, Mach number, and altitude as on a standard day. The speed is increased when the airplane flies at the same Mach number with a higher air temperature. Therefore, an increase in specific range might be expected. It turns out, however, that this increase is generally canceled by an increase in specific fuel consumption so that the range is not significantly affected by ambient temperature in a nonthrust-limited airplane. The effects are the same when cruising at constant altitude on a hot day with the nonthrust-limited airplane.

If the jet airplane is thrust limited on a standard day, as is the J-57 powered DC-8, there will be a hot-day speed and altitude reduction and an associated loss in specific range along the optimum long-range flight path. At constant altitude the thrust-limited airplane will usually fly at a reduced speed on a hot day and with an increased specific range. The latter occurs since, in general, the standard day cruise procedure will have been at a speed well above the best range speed. The thrust limitation imposed by increased temperature moves the airplane flight attitude toward the long-range condition for that altitude, that is, a lower equivalent airspeed and a higher angle of attack closer to the optimum lift/drag attitude.

Fig. 16, which illustrates only the long-range condition, shows that increased ambient temperature has a small favorable effect on the DC-8 with J-75 engines. The range with temperatures 20 F above standard is increased ½ % while the speed increases 2½ %. The DC-8 with J-57 engines suffers a 4½ % range loss and a 2% speed loss under these conditions. The DC-6B lies in an intermediate position, having a range loss of 1½ % and a speed loss of 1%. The long-range versions of the DC-8, being equipped with engines of the J-75 size, are less sensitive to temperature than reciprocating-engine aircraft.

These temperature characteristics may have an important influence on operating costs when the average cruising air temperatures are above standard. Fig. 14, which showed a considerably higher operating cost for the J-75 powered DC-8 than for the J-57 version, is based on standard temperature. With temperature 20 F above standard the J-75 airplane shows a speed gain of 2½ % compared to a 2% speed loss for the J-57-equipped airplane. This is a relative speed gain of 4½ %. The operating costs are nearly inversely proportional to cruise speed so that the difference between the cost per mile of the J-75 powered DC-8 and the J-57 powered DC-8 is substantially reduced with cruising air temperatures above standard.

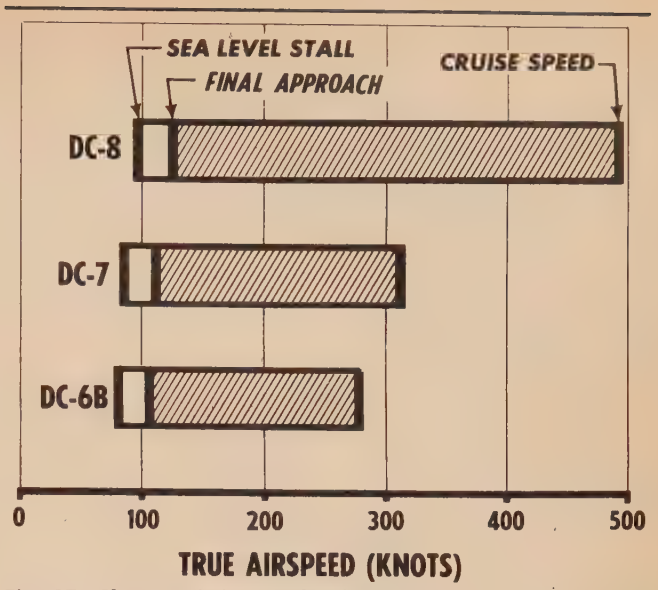

Fig. 17—Characteristic speeds of DC-6B, DC-7, and DC-8 with J-75 engine

Characteristic Speeds

Another aspect of jet-transport performance that deserves mention is its extremely wide speed spectrum. Although the normal cruising speed has been increased by 75% compared to the DC-6B, and 57% compared to the DC-7, the final approach speeds are increased by only 20% and 14%, respectively. Fig. 17 shows the landing stall speed, normal approach speed on the final leg of the landing pattern, and the normal cruise speed of the DC-6B, DC-7, and the DC-8 with J-75 engines. Versatility in terms of the difference between the minimum and maximum speeds is clearly shown in the jet transport.

Take-Off and Landing Performance

Thus far we have discussed the flight and economy characteristics of the jet transport without reference to airport runway length requirements. There has been much discussion of the long take-off distances of jet transports. These discussions have often generated more heat than light. While it is true that the required take-off runway lengths of jet transports are close to 10,000 ft at the maximum gross weights of these airplanes, it is often forgotten that these maximum weights are used only for the longest intercontinental flights. As the range is lessened, the take-off gross weight of the jet transport is greatly reduced due to the lower fuel requirements. The take-off distance reduces faster than the square of the take-off weight.

Fig. 18 illustrates the take-off and landing runway

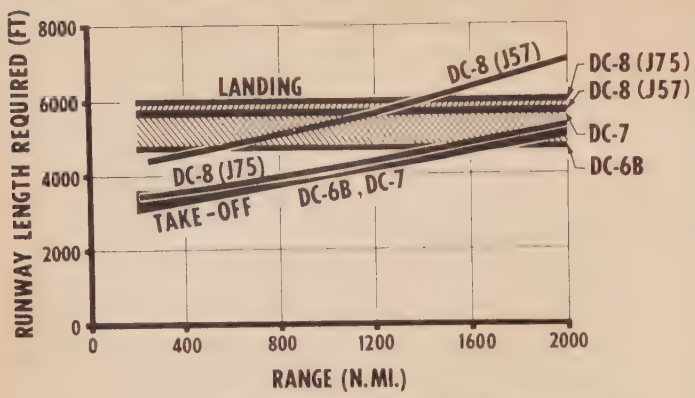

Fig. 18—Take-off and landing runway requirements

length requirements, for ranges up to 2000 nautical miles, of domestic first-class versions of the DC-8 with J-75 engines, the DC-8 with J-57 engines, the DC-6B, and the DC-7. The take-off distances of the J-75-powered DC-8 are comparable to those of the DC-6B and the DC-7. The DC-8 with J-57 engines has longer take-off runway length requirements than these airplanes, but even this airplane requires only a 7000-ft runway for a 2000 nautical mile range. Below 1400-mile range the take-off runway requirement is less than the landing runway requirement. The landing runway requirement is only 300 ft greater for the heavier of the two DC-8 versions shown than for the DC-7.

The runway-length requirements of any particular airport are dictated not only by the types of airplanes using these runways but also by the airline distances these airplanes must fly after take-off. When the runway requirements of the jet transport are examined in this manner, it is found that most existing airport runways, now used for DC-6B and DC-7 operation, are of sufficient length for jet transport operation. Even at the intercontinental terminals where the highest take-off weights will be used, runways often now exist which are either adequate or need only moderate extension.

It is interesting that the longer take-off distances of the jet airplanes at high weights are basically due to excellent flight performance. The maximum allowable take-off weight of commercial airplanes is usually set by the ability to climb and maneuver satisfactorily after the failure of an engine. The jet transport can meet these requirements at such a high weight that the distance to accelerate to flying speed is long. But once the airplane lifts off the ground its reserve of performance exceeds that of propeller-driven airplanes.

Conclusion

The jet transport emerges as a practical economical airplane at long, moderate, or short ranges. Aerodynamic and powerplant improvements during the past 10 years have solved early questions of fuel economy, of ability to fly efficiently at altitudes other than the optimum, of the effects of nonstandard ambient temperature, and of performance after engine failure. Take-off and landing runway-length requirements are reasonable. Only holding fuel remains as significantly different from present aircraft, and even this is adequately handled in the modern jet transport. Improved traffic control and landing aids will be desirable, however, to avoid increased diversions of transport aircraft to alternate airports since the jet will be less able to circle over a weathered-in or congested airport than piston engine aircraft.

It has usually been true during the last 20 years that when new larger aircraft are introduced into the airline field, they are placed into the longest range service. Existing older, slower, smaller equipment is then relegated to shorter routes. After a period of time, the larger equipment generates new traffic and this plus the normal growth in air travel forces the use of the larger equipment on the shorter routes. In this capacity the first-line airplanes usually show their versatility and ability to do the shorter range jobs better than their predecessors. We believe that the operational characteristics described in this paper justify the expectation that the same pattern will be followed by the jet transport. This type of airplane has such great speed and comfort advantages that it will ultimately permeate all airline routes where traffic density justifies this size of airplane. On such routes, the jet transport will prove to be fast, economical, and practical at all ranges.

ORAL DISCUSSION

Reported by W. H. Spannuth

Trans World Airlines, Inc.

R. D. Kelly, United Air Lines, Inc.: The long-range jet transport could also be used economically for short-range operation, but there would be a slightly higher cost than with an airplane designed for short ranges. Large jets may not be so satisfactory on short hauls; although the cost may be satisfactory, utilization would not be, and gain in block speed would not be as significant as in the longer range operation. For a given fleet it would be more advantageous to use the large jet transport on the long haul in paying its way.

Manufacturers should hold weight empty down to maintain the present excess of power, rather than taking advantage of this excess and building a heavy airplane.

Mr. Shevell: If the large transport can be filled with passengers, it will not be unfavorable cost-wise at short ranges.

M. G. Beard, American Airlines, Inc.: Why would a domestic operator use the J-75 in lieu of the J-57 engine?

Mr. Shevell: I cannot officially speak for the airlines, but I believe that they prefer the J-75 because: 1. The larger engine could be operated at reduced percentage of rated thrust and thus reduce the maintenance costs; 2. There is no speed or cost penalty related to hot-day operation (this factor depends, of course, on particular operations and routes); and 3. The J-57 has less thrust margin for operation on hot days or after engine failure.

Mr. Beard: Why won't manufacturers raise V_{ne} to obtain faster descents?

Mr. Shevell: V_{ne} had been increased over original proposals by manufacturers, and such increases naturally involve structural problems with attendant weight increase. Further increases in the permissible descent speed are considered to be unprofitable. Descent time would be affected only below 22,000 ft of altitude.

BRAKE RATINGS FOR AUTOMOTIVE VEHICLES

R. L. Wehe, Cornell University

This paper was presented at the SAE Annual Meeting, Detroit, Jan. 9, 1956.

THE determination of a method for rating automotive vehicle brakes has long been a subject of study by brake engineers. A number of seemingly different methods have been developed and some are in general use. However, many brake engineers have believed a better method could be developed. With this thought in mind, a project was started in the Machine Design Department at Cornell University in the summer of 1953.

In automotive vehicles, the brake functions to control motion by converting kinetic and potential energy to heat which is then absorbed and eventually dissipated. This heat absorption and dissipation problem is clearly illustrated in an analogy first presented by the Research Committee of the SAE in the 1922 *SAE Journal*. The problem of absorbing and dissipating energy was likened to a leaky bucket shown in Fig. 1.

With an empty bucket, a large quantity of water may be dumped in at one time without overflowing. Before the next large quantity is dumped in, enough time must elapse to permit most of the water to leak out or the water will overflow. However, it is not necessary that the bucket become empty before smaller amounts of water can be added, either inter-

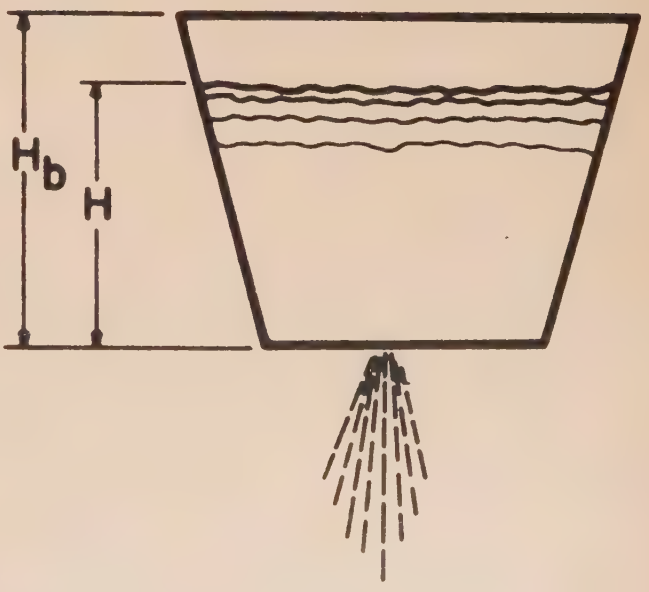

Fig. 1—Hydraulic analogy to heat absorption and dissipation in brakes

THREE factors involved in braking are energy conversion rate, heat storage capacity, and heat dissipation rate. These are used by the author in proposing a method for rating brakes.

Ultimately a brake rating system will involve many variables, some of which must be determined from experimentation.

The present rating method of gvw per square inch of bake lining area would form a good basis for rating brakes if certain limitations and corrections are made.

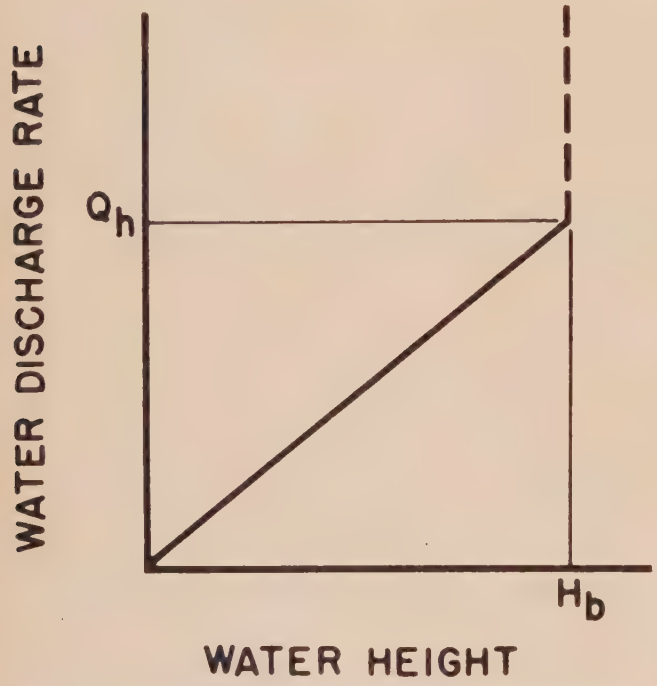

Fig. 2—Conditions for stable operation of the hydraulic analogue

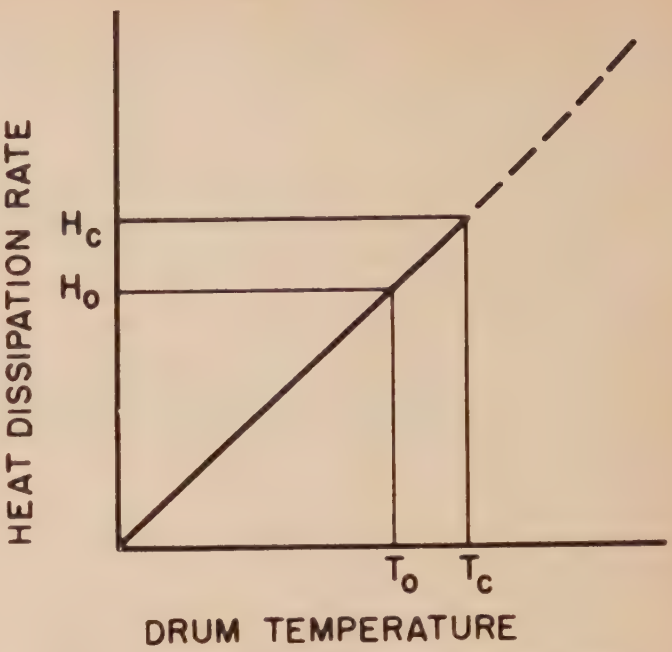

Fig. 3—Variation of heat dissipation rate with drum temperature

mittently or continuously. As shown in Fig. 2, the leaking or discharge rate will increase with the height of water in the bucket. If the rate of adding water is less than the critical value of Q_h in Fig. 2, a stable operating height would be reached with water discharge rate equal to the rate of adding water.

Similarly, if brake drums are cool enough, a severe stop may be made without the brake drum temperature reaching critical values. Also, time between severe brake applications must be sufficient for the drums to cool enough so that added heat will not cause temperatures to rise to critical values. With intermittent, moderate brake applications such as occur in some transit services, it is not necessary for the drums to cool completely between applications. The rate of heat dissipation from a brake drum will increase with drum temperature, giving a relationship such as shown in Fig. 3. If the rate of adding heat is less than the critical value of H_c of Fig. 3, then a stable temperature will be reached where heat dissipation rate equals heat input rate. If heat is added at a rate in excess of the critical rate, excessive drum temperatures will occur, with resulting rapid lining wear rate and brake fade.

This analogy points up two distinct properties a brake system must possess. It must be able to absorb all the kinetic energy of the vehicle in an emergency stop and must be able to dissipate heat fast enough so that the vehicle can be operated on a satisfactory schedule. The three factors involved in braking are, then: energy conversion rate, heat storage capacity, and heat dissipation rate.

In developing energy conversion rates, two cases were selected on the basis of having been the cause of many brake troubles. These cases are first, a vehicle on a long downgrade where gravity is the lone supplier of energy and second, a transit bus making a large number of stops per hour.

A vehicle descending a long hill is losing potential energy which must be converted either into heat or kinetic energy. Power generated on descent by this conversion of potential energy is expressed by: equation (1).

$$\text{hp}_d = \frac{\text{gvw} \times \%\ \text{grade} \times \text{mph} \times 1.467}{100 \times 550} \qquad (1)$$

This quantity can also be determined from Graph 2, p. 1015, 1956 SAE Handbook. In most trucks a large portion of this energy is absorbed in the rolling resistance of the tires, hp_r, air resistance, hp_a, chassis friction, hp_f, and engine motoring power, hp_m. Any energy not absorbed by these must be absorbed by the brakes. The relations for a particular truck are shown in Figs. 4 and 5. Fig. 4 shows the relative energy developed and absorbed by each of these means. Fig. 5 shows power which must be absorbed by the brakes. This power is the total energy conversion rate for the brakes on the vehicle, hp_{tb} which can be expressed by:

$$hp_{tb} = hp_d - (hp_r + hp_a + hp_f + hp_m) \quad (2)$$

To obtain the energy conversion rate for any one brake, equation (2) is multiplied by $R/100$, where R is the percentage of total braking done by that brake. Thus:

$$hp_b = [hp_d - (hp_r + hp_a + hp_f + hp_m)]\, R/100 \quad (3)$$

Fig. 6 shows a simplified extension of Fig. 5 to illustrate that for certain load conditions there are speeds which cannot be maintained by this vehicle, while a higher speed may be possible if the vehicle can be held under control. This results because energy absorped by rolling and air resistance is increasing faster than energy conversion due to descent. The point where the curves cross the zero energy conversion rate represents the vehicle terminal speed for that set of operating conditions, although it should be recognized that most terminal speeds for large trucks with rated loads are impossible for other reasons. Many light trucks, buses, and automobiles are able to descend at terminal speeds, and therefore, without brake applications.

Values of power consumed in rolling resistance, air resistance, and chassis friction were obtained by following the procedure outlined in the "Truck Ability Prediction Procedure," p. 1010, 1956 SAE Handbook. Engine motoring power was estimated, though it may be experimentally determined by motoring the engine with throttle closed and ignition on.

Energy conversion rate for bus brakes depends on maximum speed attained between stops and number of brake applications per unit time. To determine energy conversion rate the first step would be to get the proposed schedule, or to determine the best schedule the bus could make. From this schedule the most severe portion of the operation is taken and energy conversion rate based on this. The most severe portion of the operation is that which will deliver most power to the brakes. One method for finding the portion to be considered would be to take a definite time period—say five minutes—during which the sum of the squares of the maximum speed attained between each stop divided by the total time elapsed

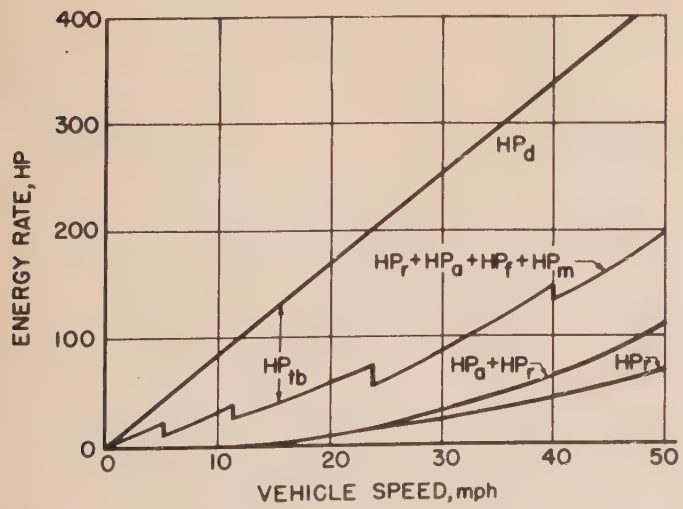

Fig. 4—Energy curves for a tractor semitrailer of 50,000 gvw on a 6.4% grade

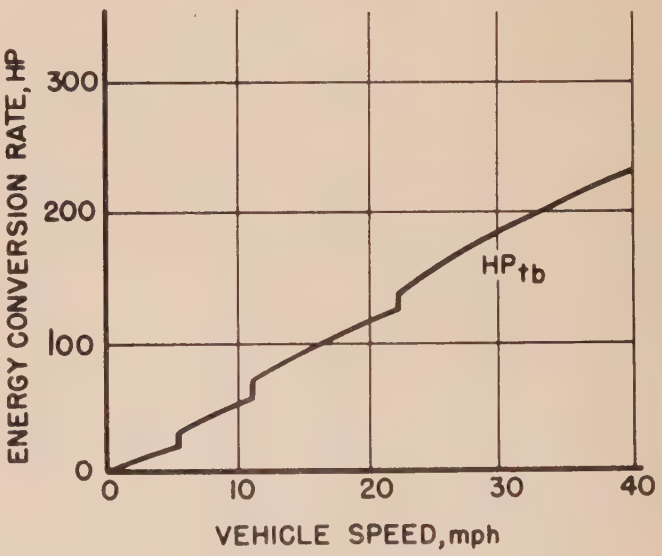

Fig. 5—Energy conversion rate versus vehicle speed

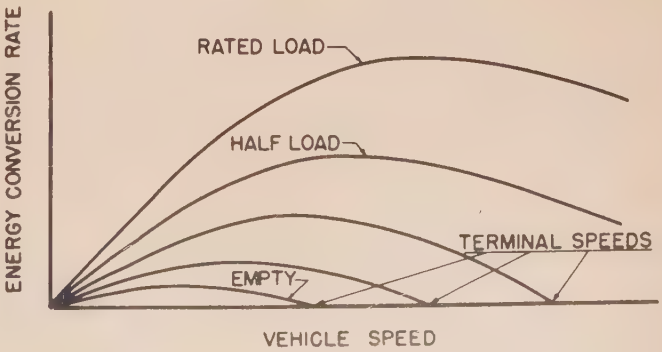

Fig. 6—Energy conversion rate versus vehicle speed with several loading conditions

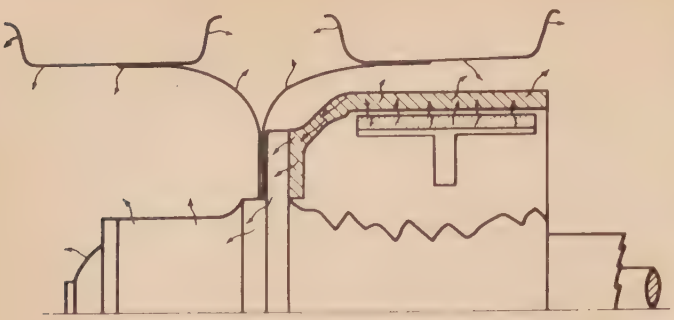

Fig. 7—Paths for heat dissipation from the brake-lining interface

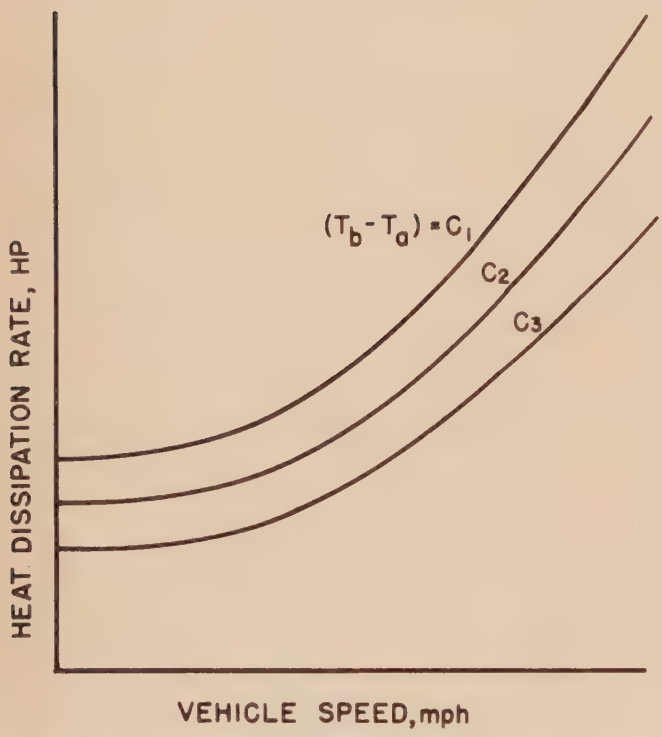

Fig. 8—Heat dissipation rate versus vehicle speed for several different loading conditions

is a maximum. The energy conversion rate of the brakes is then expressed by:

$$hp_{tb} = \frac{gvw}{2 \times 32.2} \times \frac{\sum (V_{max})^2 \times (1 + a)}{t} \times \frac{(1.467)^2}{550} \quad (4)$$

where:

t = Time period for which summation was made, sec

a = Correction factor for brake applications other than stops

The correction factor a takes care of brake applications at points other than scheduled stops. It will vary with traffic conditions, but it is felt that a satisfactory estimate could be made. Equation (4) gives the total energy conversion rate for the brakes on the vehicle. To obtain energy conversion rate for any one brake, the following equation must be used, wherein R is the percentage of braking done by that brake:

$$hp_b = hp_{tb} \times R/100 \quad (5)$$

In many normal stops vehicle kinetic energy is converted so rapidly that only a negligible amount of heat is dissipated. If brake drum temperature is not to reach an excessively high level during such a stop, the brake drum must have sufficient volume to absorb all heat generated in the most severe stop. Weight, and consequently, minimum drum thickness can be computed by equating maximum kinetic energy conversion to heat storage capacity of the drum as follows:

$$\frac{KE}{778} = c\ W\ (\Delta T) \quad (6)$$

where:

$$KE = \tfrac{1}{2} \left(\frac{gvw}{32.2}\right) \left(\frac{V}{1.467}\right)^2 \left(\frac{R}{100}\right)$$

V = Highest speed from which a stop is to be made, mph

R = Percentage of braking done by a given brake

c = Specific heat of drum material, Btu per lb-F

W = Weight of drum, lb

ΔT = Allowable temperature rise, F

Making use of the following:

$W = D\ t\ w\ \delta$ (approximately)

D = Inside diameter of drum, in.

t = Average drum thickness, in.

w = Drum width, in.

δ = Density of drum material, lb per cu in.

Equation (6) can be solved for t. If D and w are known from other considerations, t can be determined from equation (7).

$$t = \frac{KE}{778\ c\ D\ w\ \delta\ (\Delta T)} \quad (7)$$

Equation (7) gives minimum drum thickness for adequate heat storage capacity.

Requirements for continuous operation of brakes, however, are entirely different. The heat storage capacity of the drum becomes insignificant, and is of no importance if braking occurs for many minutes, either continuously or intermittently. Heat dissipation rate of the brake system becomes the important factor.

As brake temperature increases, dissipation rate increases and temperature will reach a point where dissipation rate equals heat input or energy conversion rate. The problem is to get a brake which will reach this condition below the critical temperature; namely, one with a critical heat dissipation rate greater than energy conversion rate.

In absorbing energy, a brake generates heat at the lining-drum interface. This energy leaves this surface by the three paths shown in Fig. 7. A small amount of energy, estimated to be of the order of 5% of the total, is conducted through the lining to the brake shoe. Heat is convected and radiated from that portion of the drum inner surface not covered by the linings. The percentage dissipated in this manner will be large, perhaps as high as 30% of the total heat dissipation, for open-type brakes and will be quite small for closed-type brakes. The remaining heat is conducted through the drum to the outer drum surface and to the wheel and bearings. From these surfaces heat is dissipated largely by convection. A portion of the heat is radiated from these surfaces, but where the surfaces are hot enough to radiate a considerable portion of the energy, the volume of material which could receive this radiation is small. This means that these surfaces soon reach a temperature such that only a small amount of energy is radiated to them. Some brake designs put the brake out where it is not covered by the wheel; in this case radiation is greatly improved along with improved convection. The amount of energy convected from a surface depends on the flow of air—both in quantity and condition of flow (turbulent or laminar), outer surface area, temperature difference between convecting surface and air, and condition of the convecting surface.

Although the manner in which heat is dissipated from brakes can be fairly well described, a quantitative determination of the amount of heat dissipated from a brake under given condition as yet cannot be made with any accuracy. Consideration of variables affecting heat dissipation reveals that a testing program for heat dissipation ratings would have to be set up very carefully. A partial list of these variables is given here to provide some idea of the complexity of the problem:

a. Drum diameter.
b. Drum thickness.
c. Clearance between the drum outer diameter and inside of wheel.
d. Thermal conductivity of drum material.
e. Drum design.
   (1) Open or closed design.
   (2) Shape of outer surface.
   (3) Drum length.
   (4) Drum web thickness.
f. Lining length.
g. Lining width.
h. Location on vehicle.
i. Wheel type.
j. Drum temperature.
k. Vehicle speed.
l. Scale or dirt crust on drum.

While this list of variables is impressive, it seems certain that experimental factors can be determined for some variables to allow close approximation of the heat dissipation capacity for a given vehicle without necessitating a complete test for every new design. One point makes this estimate less critical than might first appear. If predicted heat dissipation rate is high by a small amount, the temperature will be slightly higher than design temperature. This results in a somewhat shorter lining life which, while not desirable, is not critical.

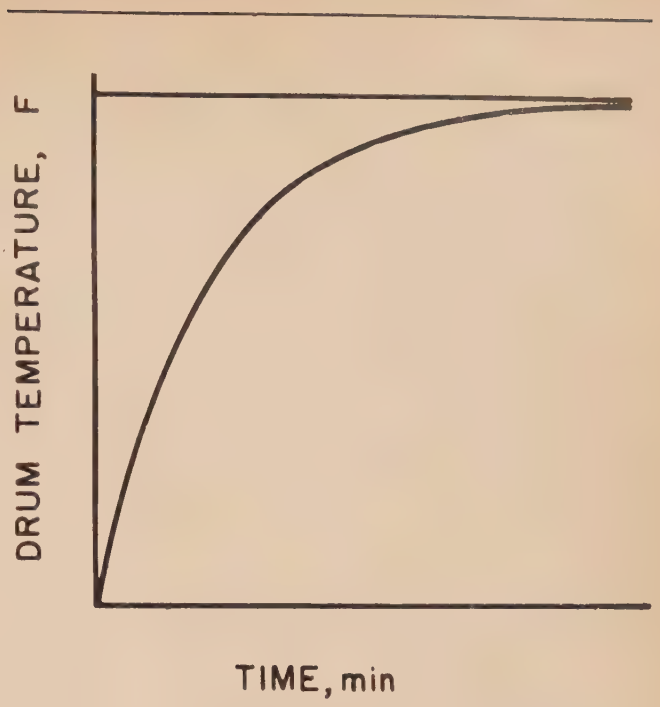

Fig. 9—Plot of drum temperature versus time

Tests to determine dissipation capacity of a given brake design with a particular wheel in its location on the vehicle have to be made either on a vehicle or in a wind tunnel with a mock-up of a portion of the vehicle. Tests can be made with a vehicle by towing on level ground or by descending hills. Separation of rolling resistance, air resistance, chassis friction, and engine motoring power should be done on the basis of the SAE Truck Ability Prediction Procedure, since that can be used later in arriving at the necessary dissipation capacity.

A heat dissipation capacity graph such as Fig. 8 can be determined for a particular brake in a vehicle by the following test procedure. The brake is mounted on the desired vehicle with a thermocouple installed as described later. The vehicle is then driven down a hill with as uniform a grade as possible with at least three different loads. Loads should be chosen so that the product of gvw and per cent grade will cover the expected operating range. A minimum of three runs should be made with each load at speeds covering the maximum range of operation. Drum temperature should be read at intervals during the descent and plotted versus time as in Fig. 9. If the curve does not approach an asymptote, the brake did not reach equilibrium conditions and the data cannot be

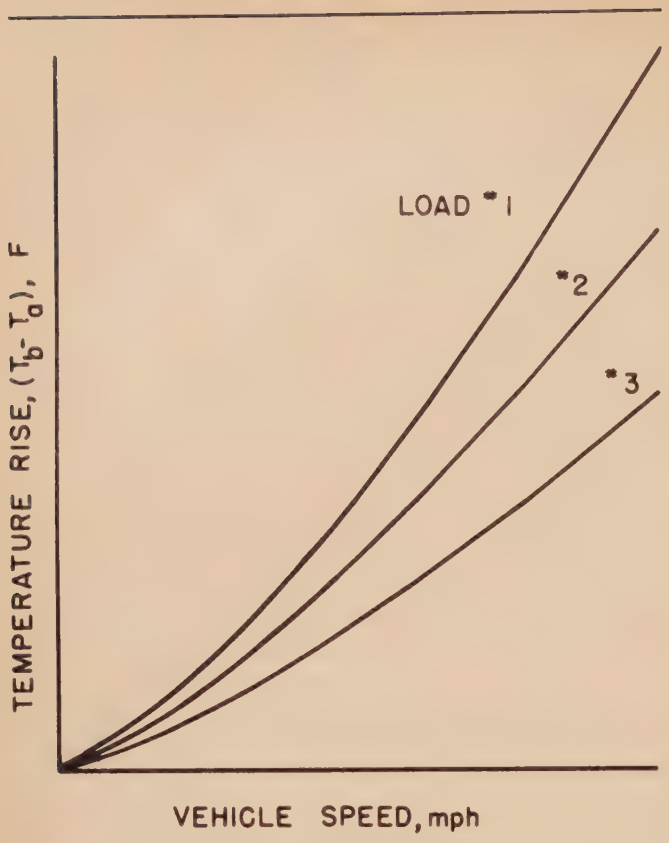

Fig. 10—Constant grade curves of temperature rise versus vehicle speed for several different loading conditions

used. A plot of temperature against speed for three loads should result in Fig. 10.

Since constant temperature curves are desired in Fig. 8, a point on one of the constant grade curves in Fig. 10 at the desired temperature is taken and speed of descent read from the graph. From the speed, grade, and gross vehicle weight the energy conversion rate can be determined by equation (3). Since equilibrium has been attained, heat dissipation rate will equal the energy conversion rate for the brake. This value is recorded as the heat dissipation rate of the brake-wheel combination for a particular location on the vehicle and test speed and temperature. This is a point on Fig. 8. The remaining points for the given temperature are determined in the same manner. As many temperatures may be taken as needed.

This procedure gives energy dissipation curves for a particular brake-wheel combination and location. To simplify testing, an attempt should be made to determine whether experimental factors for the effect of axle location, wheel type, and clearance between drum and wheel exist. If tests show little variation in these factors, then those factors need not be separately determined for each brake. It is even conceivable that the difference between dissipation rates of different drum and brake designs, will be such that use of a correction factor will give sufficiently accurate results.

In order to correlate the tests, a standard test procedure and method of temperature measurement must be established. The test procedure would govern vehicle type and its conditions of operation. Temperature measured should bear a relation to the drum surface temperature as this is the critical point in the brake. In order to establish a practicable means for determining a reference temperature without the use of slip rings, it is recommended that a thermocouple be mounted in a bronze shoe which bears on the brake drum with a light spring force. Friction of the rider in which the thermocouple is mounted will result in a certain error, but this can be determined by calibration. Running the lining tests in a similar manner also will reduce the effect of any error. The actual temperature is not as important as the relation of thermocouple readings to wear characteristics of the lining material. By establishing a standard location and contact pressure for the thermocouple shoe, thermocouple reading should bear a definite relationship to drum surface temperature. The relation between this reading and lining characteristics can be established on the dynamometer. A difficulty will arise in interpreting the results of these tests in terms of rapid wear and destructive wear since wear rate curves for various materials cannot be expected to have the same shape or critical temperature. This problem will have to be solved after sufficient standardized tests have been made to cover most of the possibilities for variation in the curves.

The procedure for brake selection using the values of energy conversion rate and heat dissipation rate is based on the necessary equality of these two values. The designer must decide the critical condition of operation for the vehicle on the basis of his experience or by a standard (if one can be developed). He will also set the distribution of braking effort. The energy conversion rate for each brake can then be determined for this condition of operation by equation (3) for truck operation or by equation (5) for transit-type operation. A brake having the necessary heat dissipation rate at the conditions of operation is then located from heat dissipation rate charts. If two or more brakes will accomplish this, the designer must choose between them on other bases such as cost, availability, or compliance with legal codes. The brake selected must be checked as to torque capacity with a reasonable energization, and also should be checked for average lining pressure due to maximum energization of the brakes.

By the previous procedure the brakes have been chosen on the basis of descending a particular grade at a chosen speed (probably the maximum speed at a particular gear ratio), and with some load (probably the rated load). The brake temperature should be very nearly the desired value when the descent is made under these conditions. In use, the truck will descend other grades with other loads, and if operation is to be satisfactory the speed must be such that the temperature will be equal to or less than the desired operating temperature.

The correct speed of descent down any grade with any load can be determined from the relationships previously presented. A graphical solution for this speed is shown in Fig. 11. The energy conversion rate curve can be determined and superimposed on the heat dissipation rate curve. The intersection of these two curves will be the correct speed of descent. While this speed could be maintained without overheating the brakes, it would probably be better to specify the gear ratio to be used. This ratio would probably be chosen to limit speed to a value below the correct speed, but the choice should be made by the designer, as slightly increased wear rate resulting from overspeed on certain grades might be acceptable.

The energy conversion rate and drum temperature variation with time to bring a 50,000-lb truck to an extreme emergency stop (19.5 ft per sec per sec), from 60 mph is shown in Fig. 12. Since a brake is basically a constant torque device, deceleration is constant and speed will decrease linearly with time. Energy conversion rate is the product of torque and wheel speed and also decreases linearly with time. Peak of the power conversion curve is 5000 hp, and decreases linearly to zero, giving an average energy conversion rate of 2500 hp during the stop. Comparison of maximum energy conversion rate with engine horsepower reveals that brakes will be called upon to absorb energy at a rate more than 30 times engine horsepower at the peak and 15 times on the average during such a stop.

Approximate temperature curves for inner and outer drum surfaces based on values obtained by the Schmidt method, a graphical method for solving heat flow problems, are also shown in Fig. 12. Two conditions are apparent which vitally affect brakes. One is the extremely high peak temperature which results in high thermal stresses since some parts of the drum are still quite cool. The other condition is the relatively late and slow temperature rise at the outer surface. These conditions will affect heat checking of drums and heat fade, and the heat dissipation during braking.

Brake ratings in present use are based either on lining area or on drum swept-area (area of the drum with which the lining comes in contact). Ratings based on gross vehicle weight per square inch of lining area are the most commonly used and have even been incorporated into legal regulations. Ratings based on energy input per sq in. of drum swept-area are used by many designers as one of the criteria of a good design. These ratings have been considered by one of the SAE committees, but none has been accepted yet.

The most commonly accepted method of rating vehicle brakes today is the ratio of gross vehicle weight to total lining area of the vehicle brakes. Not only is this value used by the manufacturers and operators, but a number of legal bodies have accepted this method of rating. Use of this figure stems from the ready availability of the values on which the rating is based and from the fact that most brake troubles are concerned with the lining. Both gvw and total lining area are given in most summaries of vehicle specifications. Furthermore, the method will give as good results as any rating method which has yet been proposed if certain limitations are met.

The first limitation has to do with the relation of lining area on each axle to the braking effort developed there. If the ratio is constant for all brakes on the vehicle, the total values of weight and lining area will have the same ratio as the value for any axle. Braking effort is the decelerating effort developed at the road. In the case of air brakes, braking effort will be proportional to the area of the air cylinder times the slack adjuster length (AL factor), if tire size,

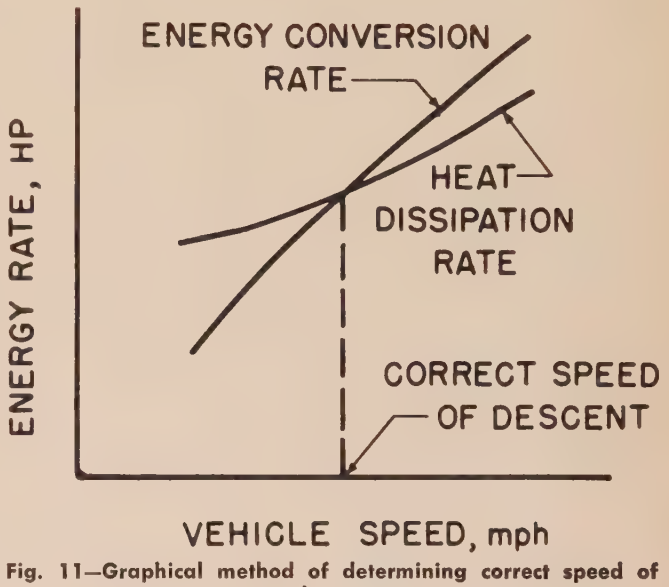

Fig. 11—Graphical method of determining correct speed of descent

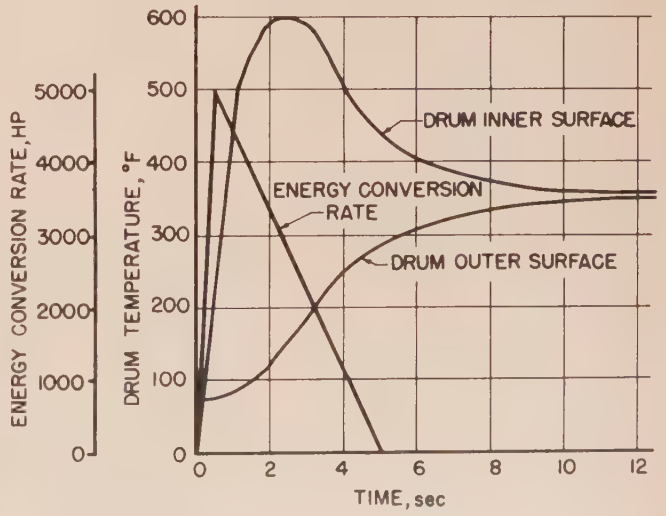

Fig. 12—Energy conversion rate and drum temperatures versus time

drum size, cam size, and lining coefficient are constant throughout the vehicle. For brake systems generally it will be proportional to the energization factor if tire size, drum size, and lining coefficient are constant throughout the vehicle. It should be emphasized that braking effort distribution is set by the designer and does not vary with varying weight distribution either from loading or from weight shift with deceleration. Possibly some standard should be developed to guide the designer in his selection of braking effort distribution as related to weight distribution and weight shift due to deceleration, and taking into account various road surface conditions met in operation.

The second limitation has to do with the extent of lining per shoe and number of shoes per brake. If lining length per shoe is held to approximately 120 deg, good use is made of all the lining. If appreciably more than this length of lining is used per shoe, the load rating given to this brake will be too large since, due to varying pressure distribution and wear distribution on the lining, and the fact that added lining area does only a fraction of the work done by the lining in the center of the shoe. With an arbitrarily set lining length and only two shoes per drum, lining length and area are roughly proportional to brake swept-area, as discussed later. Where more than two shoes are used, the rating will be somewhat in error since drum temperature—and therefore wear rate—will be higher for a given rating. This results from the energy input rate being increased while heat dissipation capacity for a given temperature remains the same or perhaps even decreases if open-type brakes have been used.

There are other limitations with regard to the manner in which brakes are used. The use of a single value for all trucks in a certain type of operation assumes that all drivers will use the brakes in much the same manner—stopping from the same speeds at the same deceleration, descending the same hill at the same speed in the same gear ratio and with the same proportion of rated gvw, or making the same number of stops per hour from the same speed for the case of transit operation. Obviously, these conditions are not met, but the approximation seems to be good for most vehicles.

It is seen then, that this method can give reasonable results, but it has one great shortcoming if lining length is not limited. Whenever a brake designer finds he cannot provide sufficient lining area by the method he knows to be best, he may add lining area by increasing the angle of lining, or by placing large drums on the front wheels, where they cannot be used fully due to steering difficulties which would arise. This practice would most likely result where the designer is faced with a legal requirement for a certain ratio. This practice is not the general rule, fortunately, but there is nothing in the present use of the method to prevent this type of abuse.

A second method of rating brakes on the basis of lining area is one which takes the average unit pressure on the lining, as exerted by the actuating mechanism, as the basis for design. This method has not had common acceptance because the values on which it is based are not readily available to many people. The method has the advantage of giving a value which has a direct effect on wear rate, but it has most of the shortcomings of the first method.

There are no official rating methods at present based on drum swept-area, but a number of authorities in the field have developed relationships which relate the gross vehicle weight to drum swept-area. This expression is related to work and energy, but some terms are missing. The missing terms have to do with magnitude and rate of energy input. The absence of these terms implies that vehicles will be operated at similar critical energy input rates. A number of vehicle-category constants have been set up in an attempt to include all types of energy inputs that might occur. Energy input per sq in. of drum surface is an index of drum heat loading and is related to the temperature at the drum inner surface. This is related both to lining material wear and to fade, characteristics which often accompany a temperature rise. There are a number of methods of applying this concept in developing a rating method.

Energy input per unit drum swept-area is also related to the brake's heat dissipation ability in that outside area is roughly in the same proportion to swept-area for most brake designs. This is fairly rough and several variables are missing which do not allow for improved designs. A more exact method of handling the brake's heat dissipation ability is given in an earlier section.

Perhaps the most direct method takes vehicle kinetic energy divided by the drum swept-area. The question immediately arises as to how the kinetic energy is determined; at what speed or what speed change? Inability to establish a standard for determining kinetic energy has held up the development of this rating system. The condition of operation can be assumed, as has been done implicitly with the gvw per unit lining area or the gvw per unit drum swept-area, but then this method has an advantage only in that the designer cannot gain advantage by increasing lining area. A further refinement of this method takes another variable out of the rating by using axle weight—for some assumed weight distribution and deceleration—as a basis for finding kinetic energy. This is correct only if braking effort is made proportional to the same weight distribution.

Another rating method which contains the variables of energy and drum swept-area is the so-called Work Formula. This formula divides the product of gvw and tire rolling radius by drum diameter squared times lining width. This can be considered in a number of ways, but the most general interpretation would be the gvw per unit drum swept-area times ratio of rolling radius and drum diameter. Gvw per unit drum swept-area is a measure of kinetic energy input for a given set of conditions of initial velocity,

velocity change, or rate of descent and gear ratio on hills. Ratio of tire rolling radius and drum diameter is a measure of lining pressure necessary to provide a given deceleration. It is readily seen that an increase in rolling radius or a decrease in drum diameter will call for an increase in lining pressure to give the same braking force at the road. The results of this method are much the same as the one using kinetic energy per unit drum swept-area, but do not make apparent the assumed conditions of operation.

Analysis of present rating methods has shown that most present rating methods will give satisfactory results for certain given conditions. Methods based on lining area can be abused but a limitation on lining length will eliminate this abuse. Another method of correcting much of the abuse is to use the projected length of the lining on the drum diameter. All methods can be abused if distribution of braking effort is not the same as distribution of braking area—lining area or drum swept-area, as the case may be. Use of more than two shoes will show an increased rating when lining area methods are used. True improvement is less than might be expected since drum temperature will depend mostly on drum configuration, and so the brakes will operate at a higher temperature with the increased input rate. In open-type brakes addition of another shoe may even cause an increase in wear rate at the original rating since some inside heat dissipating area is covered and the brake will operate at a higher temperature. Experiments reported by R. K. Super ("Brake Designs and Methods of Rating Brakes for Commercial Automotive Vehicles," SAE Transactions, Vol. 54, May 1946, pp. 205–213), demonstrated that the effect on wear rate of increased lining pressure as the lining length was reduced, was balanced by decrease in temperature due to better cooling of the drum inner surface until the lining was made shorter than 90 deg.

One shortcoming of all present rating methods is the failure to seperate energy conversion rate and heat dissipation rate. In general, this has resulted in all brakes being given the same rating regardless of their heat dissipation rate. A correction factor based on heat dissipation rate of the brake in a standardized test would serve to eliminate this inequity.

The provision of adequate brakes on vehicles has become a point of concern with various public bodies. It is easy to see why this concern has developed when reports of the results of brake failures are read. It is not so easy to determine a policy to recommend to these bodies.

It should be pointed out that no design code can be exact in the prediction of operational temperatures or lining life since the designer has no control over eventual vehicle operator and there are no legal limitations on truck speed on a downgrade or on the operator's use of the accelerator in bus operation. There is a speed limitation on the bus imposed by traffic conditions, but with higher engine power the driver may "push" traffic, resulting in more frequent brake applications. Therefore a good design cannot insure satisfactory operation. Operation and maintenance of brakes play as important a part in successful braking as does design.

Since operation and maintenance are the critical values in avoiding brake failure, it is recommended that a brake temperature indicator be provided to inform the driver of brake-drum temperature, and that an emergency warning device be provided to warn him when the brake application device—air cyclinder or electromagnet—has reached a point where insufficient actuator travel remains. Whenever this signal is given, the driver should adopt an emergency procedure until the brakes can be adjusted. The remaining travel could be checked at critical grades by authorities as an additional check against failure. By using a brake-drum temperature indicator, the driver will be informed of the operating condition of the brakes and can avoid destructive temperatures as he now avoids boiling engine coolant on ascent. These two devices will enable operators to avoid many drastic brake failures.

The more exact procedure for brake selection recommended in the paper may not be necessary if testing proves the validity of the use of correction factors as suggested. This method should be of help to the designer in critical cases. A brake rating procedure based on this method could be developed if a need still exists, but it will be difficult to develop it into a legal code.

The present rating method of gvw per square inch of lining area would form a good basis for rating brakes if certain limitations and corrections are made. Limits of lining length should be specified and a correction used if the length is greater than specified. A correction should be made if lining area distribution is not the same as braking effort distribution. A correction factor should be allowed for designs which result in improved heat dissipation rates. With these limitations and corrections and an adequate breakdown of service conditions with constants determined by experience, this system of rating will give reasonably good results.

DISCUSSION

Better Understanding of Brakes And Brake Rating Necessary

—Ralph K. Super
Rockwell Spring and Axle Co.

THIS subject has been under active discussion in the SAE, for approximately 10 years. It appears that whenever automotive brakes must be evaluated for anticipated performance, a basic concept for rating brakes becomes highly desirable. In previous presentations of papers on brake rating, there has not been the emphasis on fundamentals that is apparent in this paper. A restatement of basic concepts of energy conversion and dissipation is particularly valuable in that they should be clearly established and quite universally accepted before any satisfactory effort is

warranted toward the accumulation of necessary empirical constants or operational factors. This compilation of data may have any number of approaches, some of which are suggested by the author. The factors are also subject to progressive changes as vehicle design and operational conditions go through a normal evolution. This paper lends encouragement and hope that an academic concept of fundamentals for brake rating can be evolved which will be generally acceptable in the industry.

That phase of brake rating which relates to the selection of empirical constants and the use of this information in actual brake design will always be subject to variables of design compromises. The author is constructive in suggesting procedures for the accumulation of data which could be used effectively in reducing "brake rating" to design practice.

It appears inevitable that an acceptable method of rating brakes will become a part of automotive engineering. This should have a degree of uniformity based on fundamentals. The present status of brake performance on *every* type of vehicle reflects a need for greater effort at this time than existed 10 years ago. The requirement for developing a better understanding of brakes and brake rating, particularly within the industry, was never more urgent.

Brake Rating System Subject To Many Complex Variables

—W. P. Panny
Chrysler Corp.

AS the author points out, brake rating systems are used by both designers and the various legal agencies. Lately, the activity of these legal agencies is becoming more intense because of highway safety programs that have been prompted by current accidents due to brake failures.

It should be said that a brake rating system should be viewed carefully, since it can result in a vehicle rating system that will govern the operations of the vehicle in service. This warrants careful attention on two counts.

1. First, relating vehicle operation to brake capacity is a very complex problem because of the number and degree of variables.

Rating the brake as a mechanical device alone is a relatively simple matter. This can be done under set laboratory conditions, and results can be used to compare one brake to another with a high degree of accuracy.

However, when relating the brake to its performance in a vehicle, difficult variables appear, such as loading, weight distribution, weight transfer, body shapes, grades encountered, and methods of brake application. With such variables the system could result in a very complex structure, and its use to legal authorities would be questionable.

Whether the system could be simplified and still maintain accuracy is a big question.

2. Second, the friction brake is an effective instantaneous energy absorption device, with limited effectiveness in continuous operation. If, through a brake rating system, it is expected to have the brake maintain vehicle operation on a grade comparable to that on the level, brake must absorb several times the horsepower required for level road operation. Friction brakes are being improved. However, on the horizon there are other devices such as exhaust and hydroretarders that will be effective during the continuous deceleration period. With a brake rating system such as one based on lining area, no consideration is available for the beneficial effects of such retarders.

Present SAE Recommended Practice Needs Amending for Bonded Brake Segments

—S. G. Tilden
Permafuse Corp.

AS a manufacturer of adhesives used in bonding brake lining to brake shoes, we look with fear and trepidation on the author's suggestion that rate of heat dissipation can be increased by operating the brakes at elevated temperatures!

However, since resins used as binders in most organic brake linings are kin to those used in the manufacture of adhesives, and since the braking heat is generated at the surface of the brake lining and must penetrate through the brake lining to reach the interior bonded surfaces, the lining itself will usually start to decompose under severe braking before the bond line gets hot enough to damage the adhesive. This phenomenon makes possible the use of present bonding adhesives in the overall construction of the brake shoe assembly.

This is not true of ceramic or cerametalic friction materials, which will withstand very high operating temperatures and not decompose before presently known adhesives fail. In other words, at this time we should not consider bonding these latter friction materials with presently available bonding adhesives.

We do have compounds under study which show much promise in increasing the hot strength of bonding adhesives, but any of these will certainly disintegrate before the ceramics do, and the first point of failure in the brake lining-brake shoe system should never be in the means of attachment.

I was glad to hear the author expound upon the fallacy of the advantages of overly long brake lining segments. The SAE Brake Committee set up standards of brake segment length which were approved by the Technical Committee and are now published (as revised in 1952), in the SAE Handbook as a "Recommended Practice." For brake diameters up to 12 in., long and short segment lengths recommended are 120 and 93 deg respectively; for brake diameters 13 to 17 in., the lengths recommended are 120 and 98 deg.

However, these lengths are for riveted segments and include 12 deg of unusable chamfer (6 deg at each end), in the up-to-12-in.-diameter brakes, and 10 deg of unusable chamfer (5 deg at each end), in the 13- to 17-in.-diameter brakes.

If segments are bonded to the shoes and the chamfer eliminated in favor of providing a sharp vertical wiping edge at each end of the segment, recommended lengths should be reduced by 12 deg in the up-to-12-in. sizes, resulting in segment lengths of 108 deg for the long and 81 deg for the short segments, and by 10 deg in the 13- to 17-in. sizes, resulting in segment lengths of 110 deg for the long and 88 deg for the short segment.

This is not generally being done at the present time and results in areas of contact at the toe and heel ends of the shoe not intended when the standards were promulgated.

The present SAE Recommended Practice should be amended at an early date to provide for an appropriate reduction in segment length if segments are bonded and the chamfer eliminated.

Heat Dissipation Ability Desirable as Brake Rating

—C. D. Christie
Eaton Mfg. Co.

WE should agree on some way of determining the answer to "how big is a brake?"

A rating based on gvw per square inch of brake lining or per square inch of drum swept-area should not be used. It would limit future brake design. In spite of limiting conditions, there would surely be some way of obtaining a good rating and still have an unsatisfactory design.

We believe a rating based on the brake's heat dissipating ability seems to offer great possibilities. Then brakes for each axle would be chosen to have a dissipation rate in proportion to axle loading. We are, of course, assuming that the brake chosen for a certain axle will have a maximum torque capacity such that it can provide the desired deceleration under rated load.

We could then equip vehicles with brakes which would meet most service requirements. We would have available as extra equipment brakes with a high dissipation rate for use in mountainous country.

If this arrangement meets with legislative approval and is written into motor vehicle requirements, higher rating brakes should be required in mountainous states than in the plains states.

With steady improvement in materials it should be possible to increase maximum allowable operating temperature. Therefore, if a rating is established, the maximum temperature considered for a particular rating should be stated.

Action must be taken to standardize the retarding effect per

axle per pound of operating air pressure and per inch of operating vacuum in order to balance tractor and trailer brakes and to have each axle in the train provide its share of braking effort. There is no use putting proper brakes on a tractor or a trailer if its companion vehicle is not properly equipped. This should apply to light as well as severe brake application.

Regulating Measures Should Be Based on Performance

—F. R. Nail
Mack Mfg. Co.

I HAVE read this paper very carefully and critically, not with the intent of finding fault or debatable points of view, but rather, to learn if he has been able to develop a method of rating brakes free of assumptions, independent of operation, and incorruptible by human manipulations. In spite of such lofty hopes, absence of such a Utopian solution in no way detracted from its value.

While this presentation does not provide the perfect answer, it certainly covers an adequate range of brake design factors and how each may be used by the designer as an index or guide. A very informative, and to me the most appreciated part of this paper, has been devoted to pointing out the weaknesses of any index to design when attempting to use it as an inflexible basic rating. I congratulate the author on a very thorough and detailed analysis of this problem.

Engineers will recognize the validity of the various approaches to determine the requirements of adequate braking. The actual detail design resulting from such studies and computations naturally will vary considerably from one vehicle to another. What is not so obvious, however, is the fact that basic relationships will also vary. For instance, one type vehicle may use a design standard of a specified number of pounds of gvw per square inch of brake lining. Another type vehicle may vary as much as 50% from that figure. Yet, both brake designs can be adequate and safe for the intended service application.

This statement is not contrary to what the author points out. In his discussion of energy conversion rate and heat dissipation rate, he states: "The designer must decide what will be the critical condition of operation for the vehicle on the basis of his experience or by a standard, if one can be developed."

Further comment on qualifying conditions affecting the establishment of fixed ratings based on methods outlined would be aimless repetition as they have been very ably covered. As a complementary phase of this subject, I would like to submit observations on the possible uses and abuses of a brake rating method.

Ratings based on design factors are not an expression of the capacity of brakes to accomplish a designated job. They are an index, and as such, should be, and are, used by designers as references. The designer does not accept them as a standard of ability, for they are based on one or more assumed conditions. Variation in any of the assumed conditions will change the index value. It is true that the numerical value can be restored to the original scale through the use of a correction factor. This does not produce a scale that is any better related to actual performance ability because the correction factor is also developed from assumed conditions.

At the time such assumptions are made, operating phenomena and physical relationship of associated parts are the variables. We certainly are not advocating the premise that there will be no advancement in chemical and physical properties of components such as lining and brake drum material. If, for instance, a substantial improvement in heat absorption capacity and conductivity of a brake drum alloy were introduced, the complete scale of ratings would be obsolete.

The search for a method of rating brakes, I believe, is inspired by the desire to give the layman a yardstick with which he can determine, before trial, whether or not a brake will be safe and durable. Such a yardstick would also simplify the task of legislative bodies, whose responsibility it is to protect the public against the operation of unsafe and misapplied vehicles. The desire is commendable.

In my opinion, the results will not serve the best interests of either safety or progress. Such action presupposes that a good design will insure satisfactory operation, which the author points out is not realistic. Operation and maintenance are the critical values in avoiding brake failures.

Regulatory measures should not be based on design factors which require technical training for proper interpretation and which are subject to change as technical knowledge of the profession advances. They should be based on the end result—performance. What is desired by everyone concerned is insurance that the vehicle will stop under the command and control of the driver at a maximum deceleration consistent with safety, and that the vehicle can be stopped or controlled as frequently or as continuously as traffic conditions dictate.

Deceleration consistent with safety, and methods of testing the deceleration ability of motor vehicles, have been established through cooperative action between industry associations, technical societies, and the Bureau of Public Roads. The same cooperative effort could establish a second performance test to insure the necessary heat dissipation characteristics by requiring a specified number of emergency stops before "brake fade" reduces brake effectiveness below a predetermined value.

Admittedly, testing of vehicles could be a burden to law enforcement agencies. But inspections will also be a burden. Assuming the "perfect yardstick" were available, protection of the public would still necessitate inspections if misapplications of vehicles are to be prevented. Spot brake tests would not have to be more numerous.

In this connection, it is significant to note that more and more fleet operators are conducting their own inspection tests. Again it is performance, and not ratings, that assures them they have the desired safety.

With brake ability established by performance tests, methods of rating brakes can be left to the designer. It is his job to interpret the rating index to provide the braking system which will produce safe control.

Performance is the issue—safety the goal.

Suggests Investigating Increased Radiant Heat Dissipation

—John Thomas
International Harvester Co.

AS the paper points out, it would be convenient to find some simple way to compare expected brake performance which would, at the same time, reflect the actual capability of the brakes involved. Inherent in any rating method, however, is the danger that progress will be impeded, because improvements in designs and materials might not reflect an increase in rated capacity. The author's proposed rating method assumes that brake capacity is entirely dependent on heat dissipating characteristics of the brake assembly. It also supposes that some maximum temperature beyond which the brake assembly would become ineffective, or be destroyed. Improvements in lining and brake drum materials which would tolerate higher temperatures, but which might not increase heat dissipating capacity, might not reflect an improvement in brake performance if the author's rating method is used. If improved material should prove more expensive and brakes are rated on heat dissipating capacity only, then there will be no incentive to make the improvements. In fact, economics would mitigate against such improvements. Situations such as this are bound to arise when any value except measured actual performance is used to draw up specifications or set ratings.

There are also some practical problems associated with the proposals contained in the paper. All motor vehicle brake tests are plagued by the wide spread of experimental results. This is especially so when road tests are made, probably because of the large number of variables involved. Any rating method based on road tests will, therefore, be subject to rather wide limits of measured performance. These limits could very well be wide enough to cause considerable overlapping in performance for dissimilar brake arrangements. A great deal of background testing would have to be done in order to set a base line so that valid comparisons could be made for various brake arrangements.

The paper has some interesting implications which reflect on

the specific brake problems which have been plaguing all motor vehicle manufacturers as well as operators. It is evident that heat absorbing capacity is necessary for attainment of successive stops without overheating brakes, while heat dissipating characteristics are most important for continuous brake operation on grades. In considering these problems, one is impressed by the fact that we really know very little about how brakes dissipate heat. There are the conventional three ways for heat to be dissipated: conduction, convection, and radiation, but how to proportion heat dissipation among these three is a question which can only be answered by some extensive testing. The answer could have a profound effect on brake design. Because the rate of emission of radiant energy varies as the fourth power of the absolute temperature, it would seem to be the most important of the three dissipating mechanisms. Certainly a great deal could be done to brake drums and the parts surrounding brake assemblies to step up energy radiation.

Some interesting ways are suggested in which some of these avenues might be explored. From a test standpoint, however, the number of variables introduced by road testing becomes somewhat awesome, and it might be impossible to correlate data between successive tests. In spite of these anticipated difficulties, it would seem worthwhile to run some of these tests in an effort at least to establish some trends, even though specific values might be lost in the mass of data.

From a brake design standpoint, it appears that continuous control of the change of potential to kinetic energy during downhill operation of a heavily laden motor vehicle is the major problem associated with motor truck operation. This is really a heat dissipation problem. Improvements in design or development of devices to promote rapid dissipation of heat during prolonged braking cycles are worthwhile design goals.

Reports Author's Equation Agrees with Practice

—R. A. Goepfrich
Bendix Products Div.

THERE are certain terms missing in most brake rating methods in use today. They usually make certain assumptions because of the lack of easily obtainable data. Furthermore, resultant brake rating figures are usually empirical and, from experience, are assigned values which include missing terms and correction factors.

We use a brake rating method whereby we calculate unit work loading on the lining. This is derived simply by dividing kinetic energy which must be absorbed by the brake by lining area and time required to stop. The result is in terms of foot pounds of kinetic energy per square inch of lining per second.

We have been able to establish some rather definite values for work loadings compatible with good general brake performance for various vehicles and for various types of service. For passenger cars and light trucks under 10,000 lb gvw, the figure is 1000 based on a stop from 70 mph at 22.5 ft per sec per sec deceleration. For trucks heavier than the above, it is 1000 based on a stop from 40 mph at 19 ft per sec per sec deceleration, and for propeller shaft brakes, 1500 based on a deceleration of 10 ft per sec per sec from 20 mph.

Note that this method is based on lining area rather than drum swept-area. This is because in most brake installations today, the lining is the more critical element. However, as the author points out, lining lengths must be judiciously selected to prevent erroneous results.

The paper indicates that a high percentage of the total heat is convected and radiated by that portion of the drum rubbing surface not covered by the linings, especially in open-type brakes. This fact is seemingly gaining more recognition, particularly in some of the spot-type disc brakes. Evidently new values for lining work loadings will have to be established for the rating method previously described. These values would be influenced further by improvements in lining and drum materials.

The author describes a method for determining heat dissipation capacity of brakes on a vehicle. We believe it would be possible to determine heat dissipating rates of the basic brake and drum combination, for various loads and speeds, on the dynamometer, and to determine correction factors on the vehicle for various types of operation. Thus, we might be able to arrive at the proper physical characteristics in the design stage by calculation rather than by experience and comparison.

In checking equation (7) for determining drum thickness required to properly dissipate heat, we applied it to an existing typical 16,000-lb truck. At a speed of 40 mph, which we use in our present method of rating, we find that drum thickness comes to 0.161 in., or less than $\frac{3}{16}$ in. However, at 70 mph, the calculated thickness is 0.494 in. Actually, the drum thickness on the truck is $\frac{1}{2}$ in. Our calculated unit work loading on this brake was about 800 ft-lb per sq in. of lining per sec. This illustrates that the value of 1000 ft-lb, which we have chosen as being compatible with good brake performance, is about right and is well on the safe side.

While existing methods of rating brakes are handy and have served the industry well, we think there is merit in the author's approach, and we hope that some people will experiment with it. We certainly expect to.

Ultimate Use of Vehicle Has Important Bearing on Brake Design Sufficiency

—L. T. Flynn
General Motors Corp.

MY remarks are made as a supplement to the paper, since it is felt that there are a number of pertinent items which have a direct bearing on brake design and, consequently, on brake function. Note that some items touched upon will be in the form of questions.

As an example, operators of large, over-the-road vehicles equipped with automatic transmissions have reported that as much as twice the brake life is being secured with automatic-transmission-equipped vehicles than with vehicles equipped with mechanical transmissions.

This, then, would lead us to believe that a different consideration might be given to brake design where the vehicle is equipped with an automatic-type transmission which provides braking through gears and, consequently, a lesser requirement from the brakes proper.

Truck manufacturers have a difficult problem in that when a vehicle is built, they generally have no idea as to the final use of the vehicle—whether it is to be used as a tractor or a straight truck—and, therefore, the brakes as supplied, while satisfactory in one instance, could be unsatisfactory in another.

For instance, suppose the normal vehicle, as built, has a gvw rating of 28,000 lb. This same vehicle, when used as a tractor, could be called upon to become a part of a train which has a 50,000-lb gvw rating. Brakes on the tractor, especially at the front axle, will require a lesser rating than if used in a straight truck. While provisions are now being made to adjust for this difference, nevertheless, it has to be incorporated in the basic design.

The author incorporates a formula in his paper which can be used to determine the amount of iron needed in brake drums under certain conditions. The formula has to do with commonly used alloyed irons. An entirely new approach might be in order when bimetallic drums are used. A number of these designs are available and, in some instances, excellent results have been achieved.

One additional point is that with the constant pressure to reduce vehicle weights, everyone eyes a heavy brake drum (needed in many designs), with a little distaste.

Certainly the braking system on a 6-wheel vehicle using 6-wheel brakes can be lightened considerably, but since many components are interchangeable with normal 4-wheel units, it is common practice to install this same material on the 6-wheel vehicle, where, actually, some of the weight might not be required.

Another item of interest is that some time ago we used cast 6-spoke wheels and obtained excellent life from brake drums. In the process of lightening our vehicles, a 5-spoke wheel was substituted, the method of attaching drum to wheel was modified, and attaching area was reduced. Two things occurred:

1. Brake squealing, under certain conditions, was objectionable.
2. Shortened drum life was experienced, due, we believe, to reduction in the attaching area of the drum to the wheel.

After we modified and added about half the weight savings back into the wheel, we arrived back at the original brake condition, reestablished drum life, and quieted the brake. I cite this instance only to point out that all brake design data cannot be written into a formula.

Before closing, I want to touch lightly on a subject which is becoming increasingly important to operators of tractor-trailer units. It has been our finding in the last several years that drivers of tractor-trailer vehicles are becoming more and more alert to improper relationship of brakes between tractors and trailers.

Whether this matter is receiving more attention at this time due to somewhat higher traffic speeds with the improved power plants and gearing, or to new highways, has not been established, but we do know that drivers are reporting more and more frequently that tractor brakes are more efficient than those on the trailer, thus giving the driver a hard time under certain conditions requiring a quick stop.

We can assume only that some more attention must be given to the problem of timing brake application between tractor and trailer and that some easy means of maintaining such adjustment must be provided.

Describes Analysis in Arriving at Selection of Brake System Components

—Edmond Soroky
Wagner Electric Corp.

IT is especially gratifying to learn of an "academic" approach to the problem of brake rating. Having served on a subcommittee which has studied this same problem but has not yet been able to agree on a method worthy of recommendation to the Society, the writer is especially pleased to find the topic discussed by a party with no particular axe to grind.

The author has clearly shown that brake system components must be such as to satisfy one of two heat balance equations: (1) for a single stop, heat generated equals heat stored, and (2) for fairly continuous braking, energy conversion rate equals heat dissipation rate. He has indicated that either total heat generation or rate can be calculated from fairly simple equations based on work demanded of the brakes, but that heat dissipation rate would be very difficult to foretell and would require very extensive experimentation. Even if the establishment of a safe limiting value of dissipation rate were feasible, selection of speeds, grades, and stopping frequencies to be used in finding the heat generation rate would still be a matter requiring the designer's judgment. An evaluation of the requirements of a braking system by heat balance must therefore remain largely empirical. The author apparently recognizes the empirical nature of the brake rating problem in the latter part of his paper where he discusses varous existing methods, and concludes that the "gvw per sq in. of lining area would form a good basis for rating brakes if certain limitations and corrections are made."

The writer agrees that the various existing methods of calculating and expressing the adequacy of a braking system can be very satisfactory when they are judiciously applied, or can be quite the reverse if a designer is negligent or wilfully artful in their application. Every method involves the use of an "allowable" unit loading value—be it vehicle weight, average energy in stopping, pressure on lining, or other quantity—the magnitude of which must be selected to suit the vehicle and service to which it is to be subjected. This selection then becomes the keystone of the application of the method and the point at which the brake designer's judgment (based on experience), has no substitute.

Because of the interdependence of vehicle weight, speed, brake torque, braking time, deceleration, drum swept-area, and drum heat dissipation capacity, all existing (as well as proposed), rating methods seem to have a degree of equivalence if used along with the "checks," as the author calls them, of torque capacity, lining pressure, and braking effort distribution. The writer can visualize no single formula or equation which will take these latter factors into account along with the others. Legal requirements of gvw per square inch of lining (or drum), area, when used along with requirements of braking distances or decelerations, if these are properly measured, would go a long way toward assuring minimum standards of braking system adequacy. But the brake engineer should aim for better than minimum acceptable performance.

At the writer's company, brake system components are selected by making analyses as follows:

1. A "design" deceleration value is decided upon, for rated load, dry pavement conditions, from data which have been collected in our road testing department and which show physically attainable values for vehicles of various classes.

2. A desirable braking distribution is selected. This is generally taken to allow 5 to 10% for load transfer, which we realize to be a compromise between the shift which will occur at low and at high decelerations. In later steps, the load assigned to each brake pair is based on this assumed distribution.

3. Brakes are then selected, on the basis of torque capacity as established in dynamometer and road tests, to be capable of imparting the selected deceleration from step (1) to the established weight of step (2). This selection is made with a view toward resulting unit lining loadings of suitable value for the anticipated service to which the vehicle will be put. Since we do not use linings longer than about 120 deg per shoe, we feel that lining areas are as useful as drum swept-areas in making predictions of brake performance. The selection also takes into account the requirement of adequate ventilation for cooling drums. It has been found that the ratio of brake to rim diameter should be limited to 80% in general (or 75% for semi-drop-center truck rims).

4. Instantaneous "power loading" is computed for the selected brakes. This is computed from: (a) brake load per square inch of lining from step (3), (b) an assumed normal top speed, and (c) an assumed average deceleration. This loading, in horsepower per square inch of lining, actually corresponds to the author's "energy conversion rate" divided by the brake's lining area. Comparison of computed values with those of previous designs, which have given satisfactory performance in similar service, gives a fair indication of the ability of these brakes to properly dissipate the generated heat. That is, we believe that ample heat dissipation rates are automatically attained by limiting the power loading to reasonable values, as dictated by experience.

We have established 2 hp per sq in. as a desirable maximum value of instantaneous power loading, calculated at 50 mph and 10 ft per sec per sec deceleration for average truck service or at 70 mph and 15 ft per sec per sec deceleration for passenger cars. Average power during a stop would, of course, be only half as great.

5. When unlike brakes are used on front and rear axles, it is desirable to have them proportioned so that both axles require lining replacement at about the same time. Wear rate may be estimated as proportional to power loading, although drum and lining temperatures play an important role. Generally speaking, better ventilation at the front brakes provides lower temperatures and slower wear compared to rear brakes. With due allowance for this difference, wear life may be found by dividing usable lining thickness by the unit power loading.

We find most truck manufacturers desirous of providing their trucks with adequate modern braking systems. The past 10 years have seen an almost complete changeover, in hydraulic brake systems, to brakes with two equally energized shoes, wherein the lining is more efficiently utilized than in older designs. In a few cases, truck sales departments have insisted that drum diameters and lining widths, especially on front axles, be larger than the engineers have felt necessary. These demands are responsible for the criticism the author directs at the weight-area method—that the ratio of lining area to braking effort should be nearly the same for all axles. The emphasis on total lining area, on the part of both legislative bodies and truck salesmen, can easily lead to glazed linings and very ineffective front brakes, destroying the proper balance of braking forces. While the desirability of increased lining areas cannot be denied, an accompanying unbalance of power loading between front and rear brakes must be avoided.

Too frequently front brake effectiveness is limited by inability of the suspension system to withstand brake torque reactions. Merely increasing front lining areas without increasing front brake effectiveness is not in keeping with the best interests of the eventual vehicle user.

Some Aspects

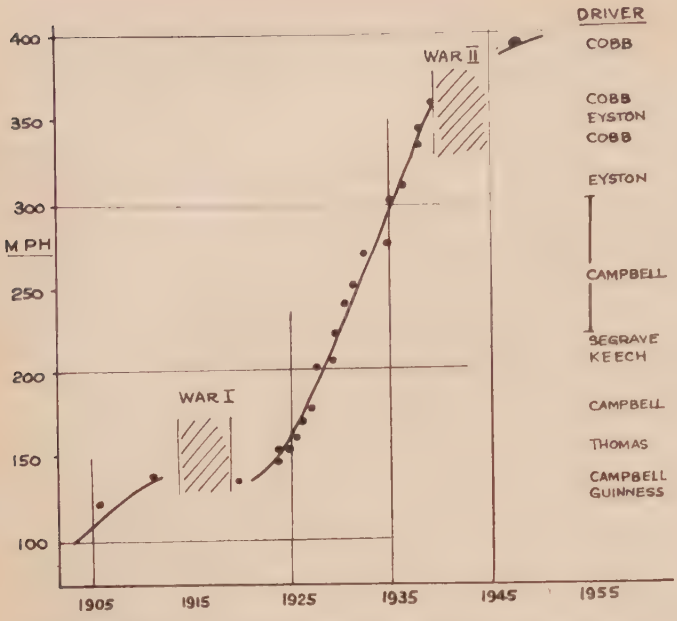

Fig. 1—Graph of world's land speed record against date

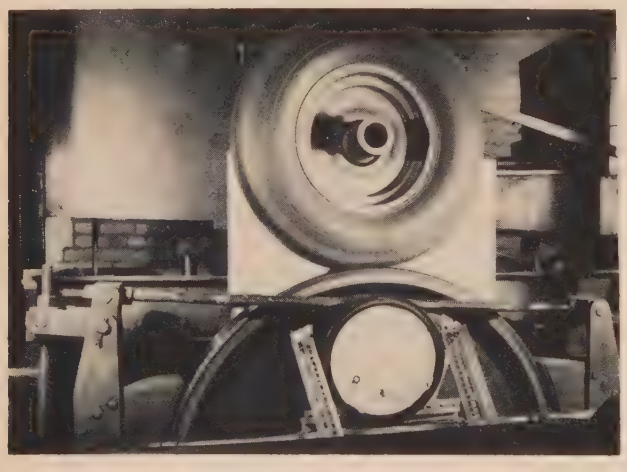

Fig. 2—Photo of high-speed machine from drum side

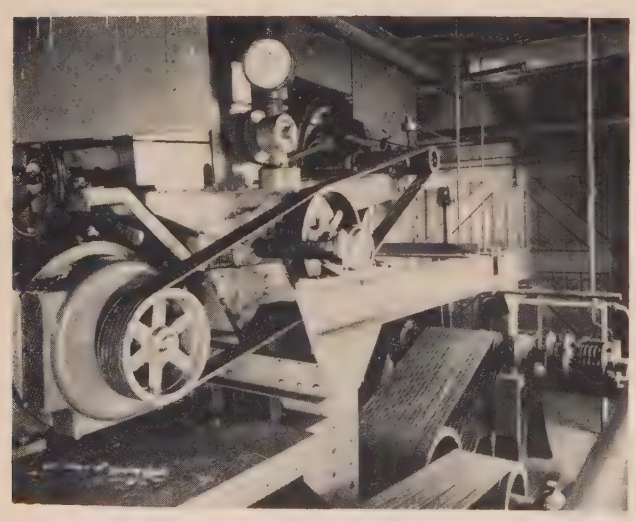

Fig. 3—Photo of high-speed machine showing beam, spin-up drive, and torsion dynamometer connections

TWENTY-FIVE years ago, there was a clear difference between normal and high-speed tires. The ordinary car of the early 1930's was going fast at 60 mph, racing-car speeds were around the 100 mph mark, and in fact, the world's land speed record stood at only 231 mph in 1930.

The corresponding speeds for the cars of today are practically doubled. Many models of stock cars in the United States and about 20 in the United Kingdom are now capable of 100 mph with safety on the road, some sports cars even going up to 140 mph. Racing maximum speeds are over 180 mph, and the record speed for an automobile over one mile, under the somewhat special conditions on Utah Salt Flats, has been pushed up to the 403 mph mark (Fig. 1).

This does not mean that we just fit the racing tires of 25 years ago to today's fast cars as the regular tires. We have to provide tread life, silence, and comfort, and, above all, reliability which the customers are entitled to expect.

This change is one index of the great strides which the automotive industry has made in two decades. Those of us concerned with the supply of tire equipment recognize clearly that the vehicle makers are the dominant partners who set the pace on performance, and we must at least keep pace. We may perhaps derive some comfort from the fact that the perform-

# of High-Speed Tire Testing

E. F. Powell, Dunlop Rubber Co., Ltd.

This paper was presented at the SAE Passenger-Car, Body, and Materials Meeting, Detroit, March 8, 1956.

ance of no car, truck, or aircraft has yet been limited by the inadequacy of its tires.

It is proposed, therefore, to talk about some of the methods used in proving passenger-car tires built to stand speeds in the region of 100 to 150 mph and to include some experiences in the very high-speed field.

A brief reference is made also to the way in which the special test problems of aircraft tires are tackled, as a good illustration of the general theme, which is to show that a large proportion of high-speed tire testing can be carried out on indoor machines to supplement actual proving on the vehicle.

Structural Tests

The primary consideration for assessing a high-speed tire must be safety. The risks attending failure are obviously greater at high speed on public roads, and the driving skill required to detect a failing tire and to take corrective action is considerable.

The road system in the United Kingdom is not favourable to evaluation by actually running vehicles at speed on the roads. In contrast with the United States and some parts of Europe, we have very few stretches where 100 mph can be held with safety for more than a few minutes. Our roads are a legacy from a more leisurely past, and too little has been spent for many years on the type of two-way straight arterial highways which are among the national assets of Germany, France, and America. Yet with our great dependence on export trade and the ever-growing British vacation habit of Continental touring in fast cars, we must provide tires to stand up to such usage.

This is why indoor testing on drums forms a major part of our high-speed proving technique, and experience has now given us a good correlation so that the results are reliable.

Drum Testing—Most of the machine testing is done on a smooth steel drum, which an electrical drive with very flexible speed range. Any chosen speed can be held steady within 1%, and a torsion dynamometer is included in the drive for measurements of tire-power consumption. Other special features of the high-speed machine are devices for very rapid loading and unloading of the tire onto the drum, and a spin-up drive mounted on the beam so that prerotation to fairly close matching speeds can be effected before contacting the tire. The purpose of this is to avoid the variable acceleration time and reduce the tread damage due to slippage at very high speeds. Figs. 2 and 3 show the machine.

The test schedules are chosen to suit the likely use on the specific cars for which the tires will be the

THE main emphasis is on the testing of tires for use on high-performance cars, and both indoor and road testing techniques are described. The importance of insuring freedom from "traction wave" or "tread ripple" is stressed.

As the difficulties of road evaluation are greater with high-speed tires, machine testing-methods have been extensively developed and correlated with service performance. These preliminary tests give a good indication of the tire's capabilities before road tests are planned.

The subsequent evaluation of tread life, casing strength, and handling characteristics is illustrated on suitable stretches of Continental roads.

Proving methods for world's land speed record tires and high-speed aircraft tires are included as examples of cases in which almost complete reliance must be placed on indoor testing.

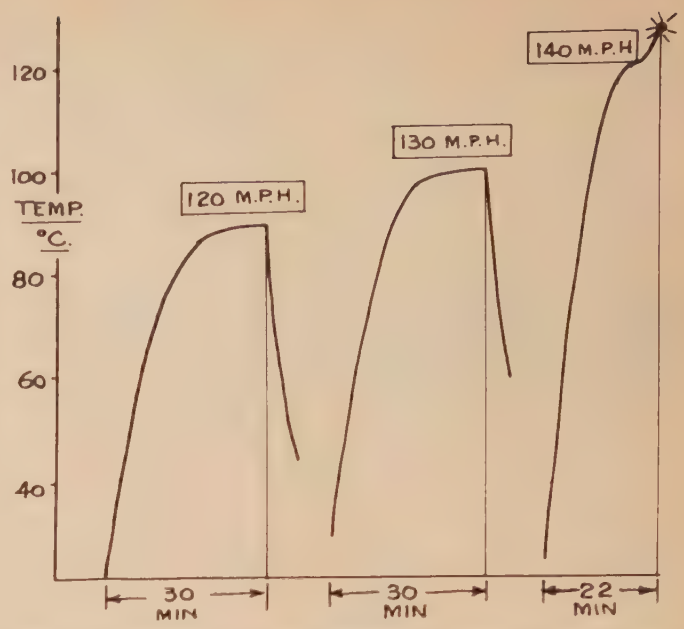

Fig. 5—Maximum tire temperatures in steady 30-min runs on drum

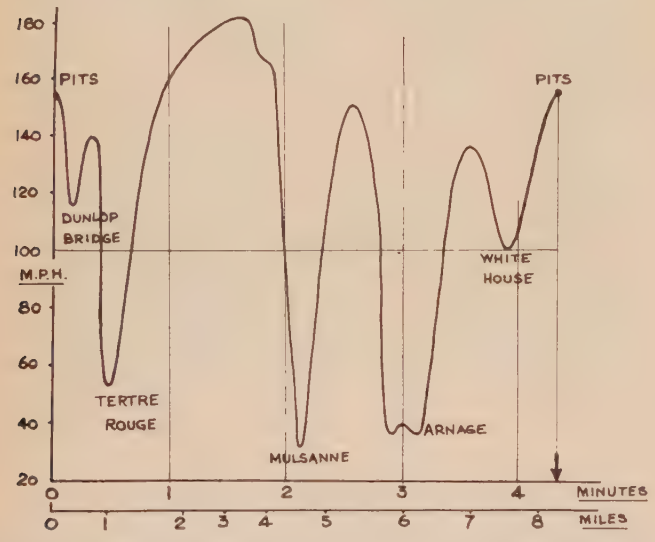

Fig. 4—Graph of speeds in one lap of 1954 Le Mans race to show short duration of peak speeds. Lap time was 4 min, 17 sec at 115 mph

normal equipment. The peak speeds are not held for long periods even in racing work, the extreme examples in European racing probably being the Le Mans Grand Prix race of 24-hr duration or the Mille Miglia in Italy which is on normal roads. Actual figures from 1954 Le Mans are illustrated in Fig. 4.

Tests are usually arranged in 30-min or 60-min runs at a series of stepped speeds, increasing by 10 mph or 5 mph near the limit of endurance. Stable temperatures are attained in about 10 min and some typical data are given in Fig. 5. Overheating is the major cause of failure at high speed, so that temperature measurement made on each run gives an indication of the safety margin.

Standing Waves or Tread Ripple—This interesting phenomenon is encountered in high-speed running due to the oscillatory recovery of the tire after deflection. It is sometimes called "traction wave" and is easy to observe as the waves are stationary in space, though the tire material is moving through them. Photographs of severe examples are shown to clarify this, the duration of the exposure being several tire revolutions as can be seen in Figs. 6 and 7.

The effects of rippling are a sharp rise of rolling resistance (which, in fact, shows before ripple is visible), and consequent tire heating, and due to the severe stressing, the tire life is very limited if the severity of the ripple is more than about $\frac{1}{20}$ in. Any given tire design has a definite speed ceiling from this cause, and it is essential to establish that this is higher than the maximum speed in use. Techniques for accurate measurement and comparison have therefore been developed.

A camera mounted in exact alignment with the contact area is set to include the face of the tachometer, and a standard scale for subsequent enlargements of the prints. Then by a rapidly accelerating run of about 30-sec duration, with a photograph taken every 2 sec, a series is available which includes conditions of severe ripple without great risk of tire failure.

The type of data obtained is shown in Fig. 8 where amplitude is defined as the height of the first peak on

Fig. 6—Photo of severe tire ripple (¾ view)

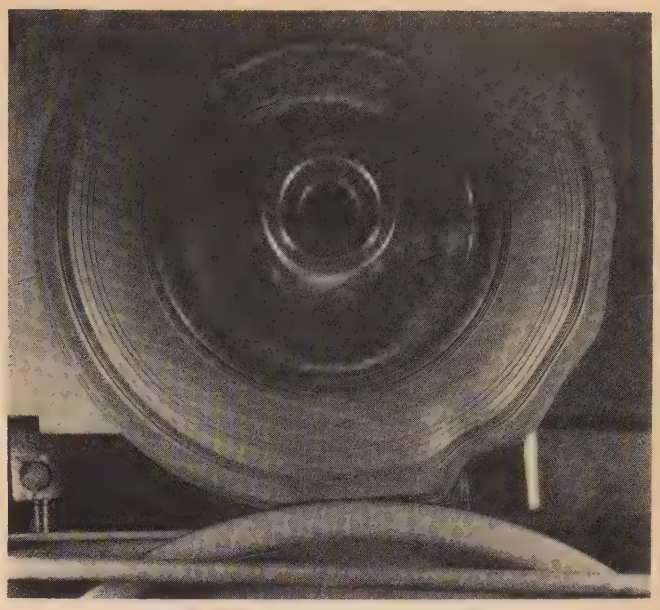

Fig. 7—Photo of ripple from end-on position to show amplitude

the tread outside the true tread circle. The main conclusions which are drawn are:

1. The definite speed at which ripple starts which is controllable by tire design.

2. The sharp, almost linear rise, as speed increases. It is not possible to drive through to a higher speed where the tire again runs normally.

3. The marked effect of higher inflation pressures. This will be referred to later when discussing road testing. One psi increases lifts-ripple speed by roughly 2 mph.

4. The negligible effect of tire loading.

These data were all obtained on the 48-in. drum of the machine shown in Figs. 2 and 3, where it is easy to control the exact conditions, but it will be seen that we were begging a most important question, that is, how far are these results applicable in road running? A project was therefore planned to make a direct comparison.

It is not proposed to burden you with all the theory of the cause of tread ripple. An excellent paper is on record by Joy, Hartley, and Turner [1] which deals with this, and it is sufficient to say that it is possible to calculate theoretically the expected difference due to the effect of drum curvature and flat road. For a 48-in. drum, the speed appears to be about 15 mph higher for a given severity on the road than on a drum, which exaggerates the effect. It may be said that experiments have been made on the machine shown in Figs. 9 and 10 by running a pair of tires one on another to imitate the flat road. They were not successful in producing the right kind of wave shape, due to the elimination of tread shuffle.

Using the same tire for drum and road work, an

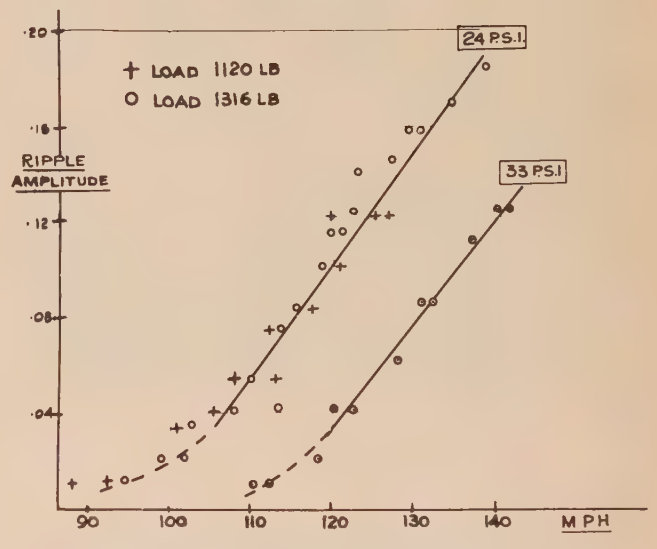

Fig. 8—Graphs of ripple amplitude against speed

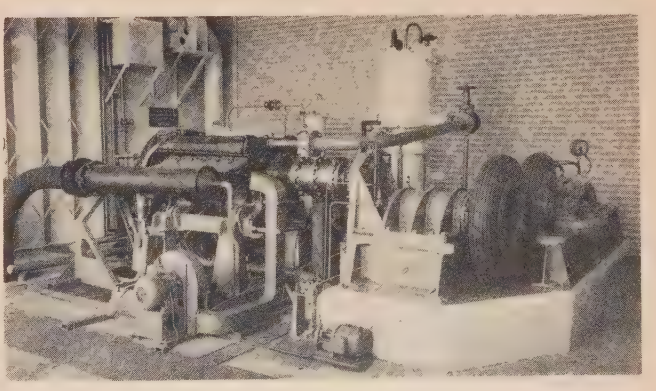

Fig. 9—General view of high-speed aero tire tester

[1] SAE Transactions, Vol. 64, 1956, pp. 319–333: "Tires for High-Performance Cars," by J. P. Joy, D. C. Hartley, and D. M. Turner.

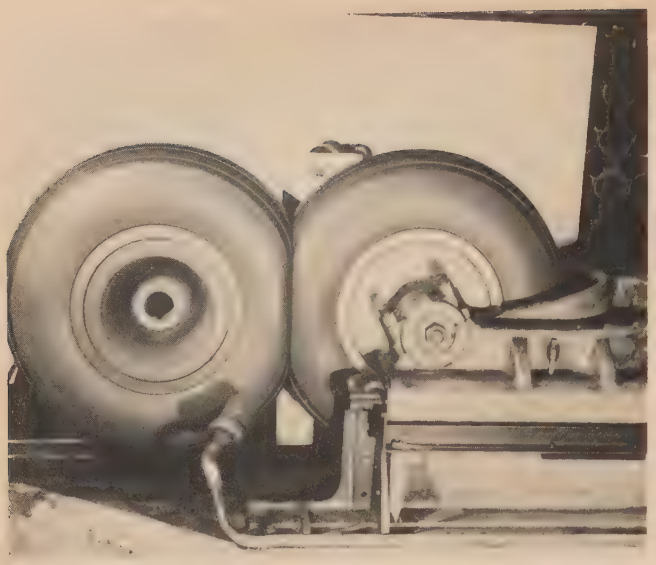

Fig. 10—Arrangement of double tire loading on aero machine

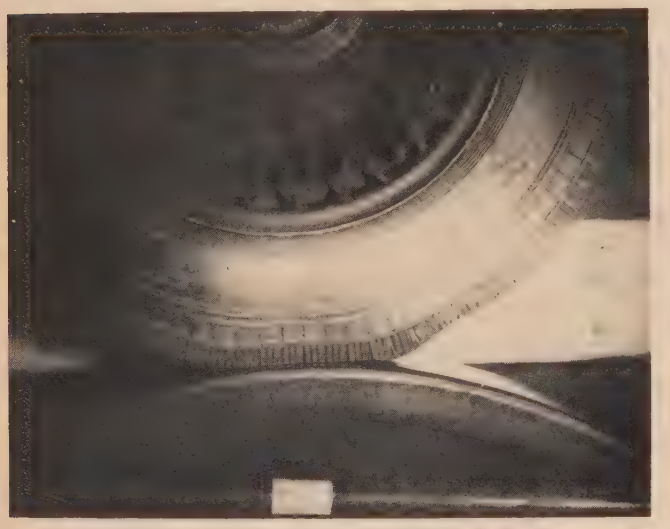

Fig. 11—Flash photo of tire rippling on drum

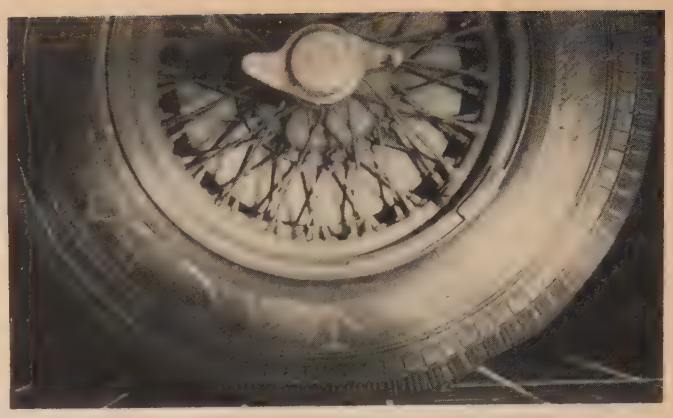

Fig. 12—Flash photo of tire of Fig. 11 rippling on road

Table 1—Jabbeke Road-Test Results

Tires	Pressure Rise from Cold, 19 F	Pressure Rise from Cold, 26 R	Time Run, min	Speed, mph	Maximum Tire Temperatures, C: Left front	Right front	Left rear	Right rear
Competitor 1, 6 Ply	24	32	10 6	100	[101]	[107]	97	[100]
Competitor 2, 6 Ply	25	33	9 2	102	[132]	[145]	129	119
Dunlop, 6 Ply (O.E. for Car)	25	36	48 3 Return trip	106	100	93	99	104
Dunlop, 4 Ply	25	33	19 7	106	103	111	104	106
Dunlop, 6 Ply Experimental	26	35	20 3	108	101	127	123	125

Note: 1. In temperature columns, numbers in boxes indicate failed tires, causing finish of run. Failures indicate tread stripping.
2. Speed was at full throttle. Maximum speed is dependent on tire fit.

Aston Martin car was run on the banked circuit at the Motor Industries Research Association Proving Ground, with the experimental tire on the left front wheel. The camera was set up alongside the track which was marked out with a grid of white lines for measurement and as a guide for the driver.

The firing of the flash tube, which illuminated the position where the tire would be exactly opposite the camera, was timed by the cutting of a crossbeam of light by the front of the car. A calculated and adjustable delay of about 20 milliseconds had to be introduced in the electronics to allow for car length, and the short exposure necessary to give a sharp picture at 150 fps involved large lens apertures and a shallow depth focus. We had, therefore, to ask the driver to toe the line to within 1 or 2 in. as he came past 3 ft from the camera, and it must be admitted that not all the pictures were usable. The set-up rather called to mind the Ancient Mariner in Coleridge's poem, who "with his glittering eye, stoppeth one in three."

However, sufficient pictures were obtained for a comparison of shapes and amplitudes to be made. Figs. 11 and 12 show a matching pair of closely similar ripples on the same tire. It is quite fortuitous that the flash happens to have stopped the tire on both occasions at nearly the same point. The drum speed was 85 mph and the road speed was 100 mph, thus confirming roughly the theoretical difference for curvature.

With these and casing fatigue tests at more moderate speeds, we proceed with road trials with some preliminary idea of the tire's capabilities.

Road Tests at High Speed—A special stretch of the highway in Belgium between Ostend and Ghent has been found most useful for this work. It is a dual-way concrete road, level, and straight with practically no side turnings for 40 miles, and even though it carries normal traffic, it can be used for runs of 20 to 25 min at over 100 mph (Fig. 13).

Ballast is added to the car in addition to the crew of driver and observers to bring up the load to the full rating, and pressures are set cold for this load. All tires and wheels are balanced statically when fitted to within 4 in.-oz.

Preliminary warmup runs between 60 and 90 mph were made so that no high-speed run is attempted without a half-hour break-in period.

Speeds are known to about 1% accuracy from previous calibration of the speedometer and rolling radius of various tires and from a time check on the run.

Experience has shown that tires attain a stable temperature in about 10 min of continuous running, and with allowances for cooling rates, the temperatures are measured as quickly as possible at the end of each run.

Table 1 gives extracts from tests made in September, 1955, on 8.20/15 tires on a Rolls-Bentley car in which five of the 20 tires failed due to thread stripping during the run. Some interesting conclusions can be drawn from these figures.

1. High temperature was not the sole cause of failure. Inadequacy of rubber compounds is indicated in these tires which failed at moderate temperatures.

2. Pressure rise is vitally important. You will recall from the drum test curves of Fig. 8 how increasing pressure gave a margin of security by pushing up the ripple speed. Deflection is also reduced, and this is one function of the warmup runs which lift the pressure to within 1 or 2 psi of the final maximum. Inadequate pressure rise can be explained by tire growth which allows the internal volume to increase.

We have found that an increase of 5 psi in inflation pressure is equivalent to about 5 mph increase in failure speed. Limitation of tire growth is therefore an important factor in high-speed tire design. A gradual buildup to high speed would also seem to be a safe rule for using high-speed cars, so that the full pressure rise is available for the peak speed.

3. The reduction of maximum car speed with some tires, coupled with the failures strongly suggest that ripple was occurring on these tires.

4. Our judgment on these results would be that any of the three last types of tires in the table would be adequate for this car.

The figures quoted from this most recent work are typical of similar proving carried out on the tire equipment for such cars as Austin-Healey, M.G., Jaguar XK 140, Triumph TR 2 and 3, and Sunbeam Talbot.

These cars are suitable for the work in the next phase of tire testing which is concerned with the endurance and handling characteristics on the road.

Endurance and Handling Tests

The evaluation of tread life, casing endurance, and, to use Maurice Olley's phrase, road manners, follows the lines of orthodox road testing, except that a suitable road and car must be chosen to keep up an average speed of 60 mph, that is, a maximum of 90 to 95 mph.

Some work can be done on circuits or closed race

Fig. 13—View of Jabbeke road for 100-mph tests

Fig. 14—Typical road conditions on French Circuit

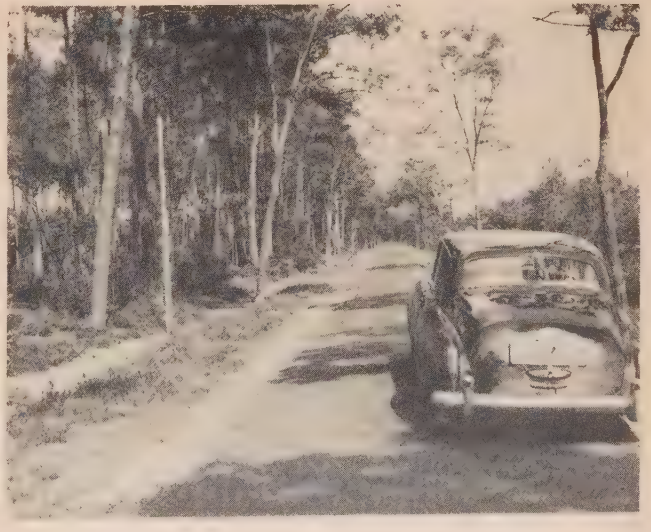

Fig. 15—Typical road conditions on French Circuit

Fig. 16—Typical road conditions on French Circuit

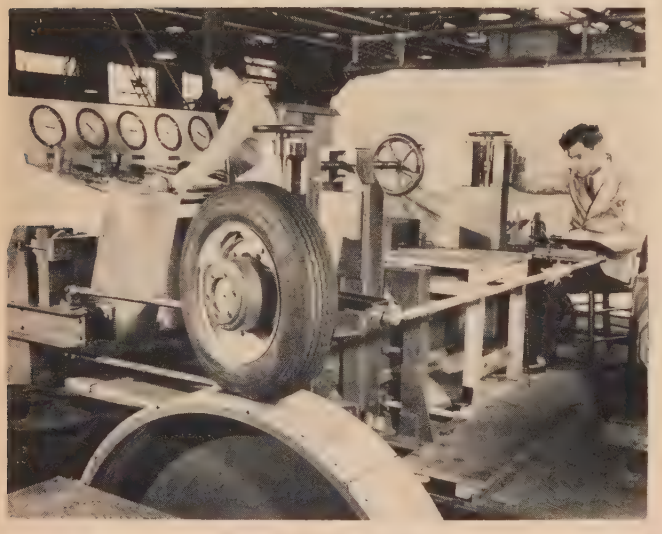

Fig. 17—One-ton cornering force machine with tire in position, showing control panel

Fig. 18—Detail of drum surface of Fig. 17

tracks, converted from disused airfields, but we find that, though the emphasis on fast cornering involved is useful in evaluating tire-handling properties, the abnormally fast and irregular wear is a disadvantage.

Road service conditions under full load are needed to evaluate casing, tread life, and bead adequacy, and we find the roads in Southern France are best for this purpose. Figs. 14-16 show conditions somewhere between Bordeaux and Biarritz.

Drive reactions on comfort, squeal, and cornering are valuable in such tests, as they form a check on our indoor tests on machines, in which tire fundamentals such as cornering power, self-aligning torque, and camber thrust are measured at quite moderate speeds.

Our experience on these indoor tests may be of interest. We have found that all speeds up to 50 mph on the drum machine shown in Fig. 17, give closely similar results on cornering behavior, and we noted with great interest the excellent paper by J. J. Robson [2] on this subject. In our more limited speed-range, we would agree with his findings in general, and we feel that testing at more moderate speeds is quite sound. It is interesting that at 100 mph the driver on the Jabbeke road was able to comment that the 4-ply tires of lower bias angle gave more feel than the 6-ply. They had also shown a higher self-aligning torque on this machine.

The surface of the drum is knurled, not plain (Fig. 18). We find this gives a high and consistent friction, without much tread damage, and is much less susceptible to change due to films of oil or rubber than is a plain steel drum.

The rate of development of lateral forces after steering, which is governed by distance travelled rather than time, is an important factor in normal speed-driving, but we think less so at very high speed. As S. A. Lippmann showed,[3] the tire takes up its steering distortion and generates 99% of its full lateral forces after rolling only some two or three revolutions after a sudden change of heading.

Our own measurements made on a flat board pulled slowly under the tire suggest that only about one revolution is necessary to give 90% of the ultimate value of cornering force, and thus confirm the data of Lippmann very closely. It follows that, since steering movements at high road-speed are slower in relation to distance travelled on the road, this adjustment rate as Lippmann called it, is not a major factor in high-speed steering behavior.

High-Speed Testing of Aircraft Tires

Although rather a digression from tires for purely earth-bound vehicles, the story of high-speed tire-testing would be incomplete without some brief refer-

[2] SAE Transactions, Vol. 64, 1956, pp. 335–337: "High-Speed Cornering Forces," by J. J. Robson.

[3] "Car Stability and Transient Tire Forces," by S. A. Lippmann. Paper presented at SAE National Passenger-Car, Body, and Materials Meeting, Detroit, March 4, 1954.

ence to tests used for development and proving of aircraft tires.

The test rig, primarily for airplane tires, is interesting since the design is unique. The drive is taken direct from a Rolls Royce Merlin aircraft engine of 1700 hp and one of a pair of identical test tires is mounted indirectly with gears to run at crankshaft speed. This design gives great flexibility of control, without the difficulty of very heavy electrical power demands for short periods, and allows for very rapid acceleration (Figs. 9 and 10).

The machine is a valuable alternative to the Adamson inertia type of tester.

Bump loads of up to 18,000 lb can be applied by pneumatic jack for periods of 1 or more sec to imitate landings, and any schedule for aircraft tires up to 300 mph can be duplicated.

One typical take-off test for fighter tires is an acceleration run at 60-fps pressure under 10,000-lb load up to 225 mph.

The double-tire method of testing avoids the difficult problem of designing light and safe steel drums for this range of speed and high loads. We find it essential in some of our tests to use the actual light-alloy aircraft wheels to mount the tires, rather than more durable steel wheels which are too heavy to permit the rapid acceleration needed for a 20-sec take-off test. A heavy steel wheel would be quite unsuitable for a landing test involving touchdown on a stationary tire, as the excess inertia would damage the tire.

Temperature rise in the tires is very fast, peaks of 80 to 100 C being reached in 30 sec. Even with artificial water-cooling between runs, no more than 2- or 3-cycles per hr can be achieved at high speed.

A particularly difficult problem arises in proving tires for aircraft which are to operate on airfields at high altitudes (Mexico City is an example). Here the lowered atmospheric pressure reduces engine power to such an extent that abnormally long runs of 1½ to 2 miles are needed to take-off. It is essential to put a tire through many cycles of such conditions to ensure good service.

At the maximum speed of 3000 rpm, corresponding to surface speeds of about 320 mph on the larger landing wheel tires, it is possible to burst some aircraft tires designed for much lower speeds by purely centrifugal stresses without load. This provides a useful overspeed check on designed casing strength, and the aircraft wheel construction at the same time. The purpose of the torpedo netting which covers the spindles is apparent in tests of this nature.

Speed-Record Tires

Returning now to land tires, and the world's land speed record, the test work involved in trying out tires intended for the ultimate in speed, which is now 400 mph, is a good example of a field in which complete reliance must be placed on the indoor drum testing as no road tests are possible until the record attempt is made. It is also certain that a tire failure would have serious consequences to vehicle and driver, although the course is straight and level.

A considerable amount of preliminary work is therefore involved before a world record attempt which is always a special undertaking.

Provision of tires for the intermediate speed range of 200 mph for motocycle and light-car records has been useful in providing experience for the designers and testers in the very high-speed range.

The same machine seen in Figs. 2 and 3 has served for all the attempts since 1936, but has reached about the limit of its speed capacity. We have run tires under a load of one ton at 420 mph for periods of 1 min for the 1947 events by John Cobb.

The very thin tread is the maximum that can be used to withstand the centrifugal stress, which at the top speed, amounts to 3200g.

Accurate measurement of loaded rolling radius and centrifugal growth are an important part of the test, as the driver relies on this data to judge his speed.

Conclusion

The emphasis in this paper has been on the testing of tires for the powerful, but safe, cars which are on the market in quantity, rather than on race tires and special record breaking, although the testing methods have much in common. The economic problems of tire making for racing purposes are probably as difficult as those of the production of the racing cars themselves, and the manufacture of tires for racing is often a matter of policy rather than commercial reward.

It is pleasing that the stock cars of modern design are now capable of such performances that some of this experience can be built into tires for quantity production.

Summarizing our views on the relative merits of indoor testing compared with evaluation on the road, we would classify high-speed tire-testing into three groups:

1. Proving of production tires for high-performance cars as a preliminary to actual road testing, where conditions of use are accurately known. The correlation is usually accurate and reliable, and road testing can be planned more efficiently after preliminary indoor tests. However, the whole of the testing could be carried out on the road in such cases, if desired.

2. For competition and race work, where road tests are expensive and difficult. Here drum testing is valuable in saving of time and expense.

3. For special record events and proving of aircraft tires, the full indoor evaluation is practically essential before even preliminary trials are made on the vehicle, as the consequences of failure are too

serious. The indoor testing methods must be such that a good margin of safety is provided.

Acknowledgments

Acknowledgments are due to the Dunlop Rubber Co. for permission to publish the information covered in this paper. It will be appreciated that the field covered has necessitated drawing upon the experience and knowledge of many specialist colleagues, among whom are H. R. Fletcher (aero), D. Badger (racing) V. E. Gough (tire research) and J. I. S. Williams (tire testing), whose cooperation is recorded with thanks.

DISCUSSION

Discusses Importance of Safety At High Speeds

—W. F. Perkins
B. F. Goodrich Co.

THE methods used in evaluating tires at high speed have been somewhat neglected in the reports which have been made as the result of high-speed tire-design problems, although they have a considerable popular appeal since any mention of high-speed tire testing usually causes the hearer to visualize the driving of vehicles at break-neck speeds across country somewhat similarly to some of the cross-country speed runs which were popular in earlier years. As a matter of fact, however, such testing must be done on carefully controlled scientific basis, or it becomes too generalized to be effective.

Mr. Powell's paper points out the interrelationship of indoor and outdoor testing procedures, with the indoor machines doing the early stages of tire development and the outdoor road tests being used to make final evaluations. This is the custom generally followed in the United States as well as in Great Britain and on the Continent. In considering his statements as the results of such testing, we can only agree that the first requirement to be met in the development of high-speed tires must be that of safety, and that in the final choice between various tire characteristics which are controllable, this must be the paramount item for consideration. The importance of suppression of the standing wave, formed at high speed, to tire safety has been pointed out, together with the fact that the destructive effects of this wave are evident in the standard construction passenger-car tire in the 95 to 100 mile speed range. There has been unanimous agreement on this, and we are completely in accord with the basic findings. We must point out, however, that in this country this presents a much more acute problem on tire safety, in that for the past four years we have been faced with the development of a tire not for special purpose vehicles or even for a premium-price tire market (where some control exists through sales marketing for selected application) but for the family car, which today in almost all versions has a top speed of above 100 mph. In addition, we now have some roads, and are getting more of them, where such speeds are possible.

Mr. Powell has pointed out that the ripple or standing wave has a definite and somewhat controllable speed at which it starts, that there is a sharp increase in wave amplitude as speed increases, and that higher inflation decreases ripple. We cannot confirm the exact correlation of change in critical speed to tire-pressure rise which Mr. Powell has shown, but we do confirm a relationship in which higher starting pressures, and a rapid rise to running pressure make possible safer operation at full throttle speeds where unsafe operation could result from lower starting pressures. In general, our findings confirm those reported by Mr. Powell that:

1. Failure at high speed almost invariably comes from tread stripping or portions of the tread tearing from the design which is probably a very forunate circumstance for many motorists as it provides an unescapable warning of, "cease and desist."

2. A rapid pressure rise from starting pressure is necessary for safe operation and in this connection, the usually wet air found in most service-station air-lines is a distinct advantage, as it has been our experience that tires which could be warmed up to at least one-third higher than starting pressures before starting a high-speed run, would be relatively safe at 100 mile or higher speeds. It follows, that tires started at higher pressures than normal need less warmup time to reach safe operating pressures. Currently, most really high-speed driving is limited to our Western areas, and in those areas, by trial and error, the actual service air-pressures used have been raised by the general public to a figure close to the needed end-of-warmup values. This is another instance illustrating that many times the practical solution of a problem has been worked out by the public before even the need has been recognized by the technical controlling group. As the possibility of high-speed driving extends to other areas by turnpikes and limited access roads, we must face up to the problem of the occasional high-speed driver, the tourist, usually with a higher than normal seat occupancy and a heavy load of luggage, and others who will attempt to drive at high speed without adequate tire preparation, tire warmup, or even adequate driving skill. This is a serious problem and has yet to be completely solved, particularly since there is a real reluctance to sacrifice some of the other attributes of tire performance which interfere with high-speed performance.

The charts and pictures illustrating Continental roads, and testing conditions are in sharp contrast to some of the conditions used in tire development in this country.

Pneumatic Tire Developed for High-Speed Performance

—W. H. Hulswit
United States Rubber Co.

THE need for a special tire for today's high powered cars in the hands of really fast drivers has been obvious. Our company's high-performance tire, the XP-140, was put into production a little more than a year ago and has been well received by a variety of users who need a tire that will be safe at speeds in excess of 95 mph.

A pneumatic tire is a complex compromise between a large variety of performance factors, durability factors, and cost. To

Fig. A—A 7.60-15 Royal tire running at 100 mph

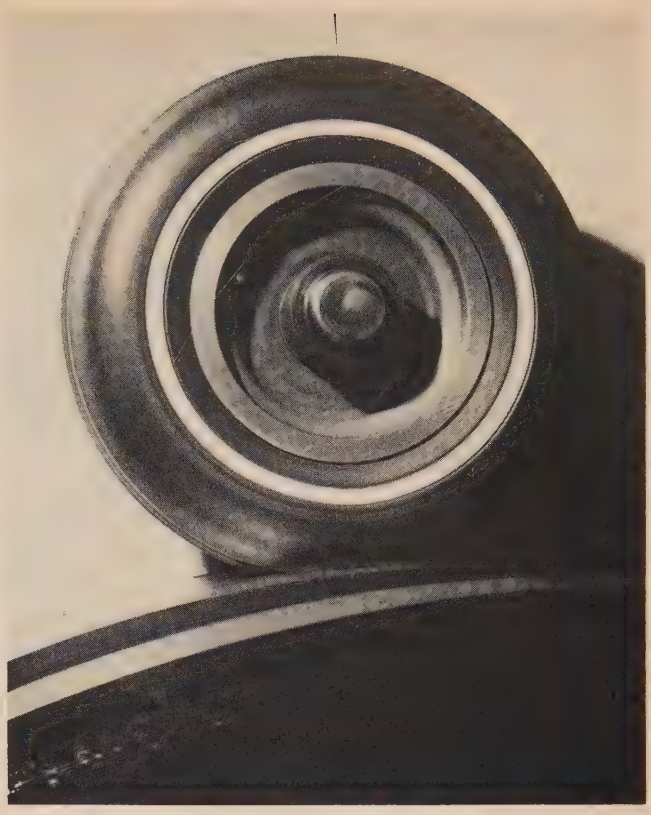

Fig. B—The XP-140 running at 100 mph

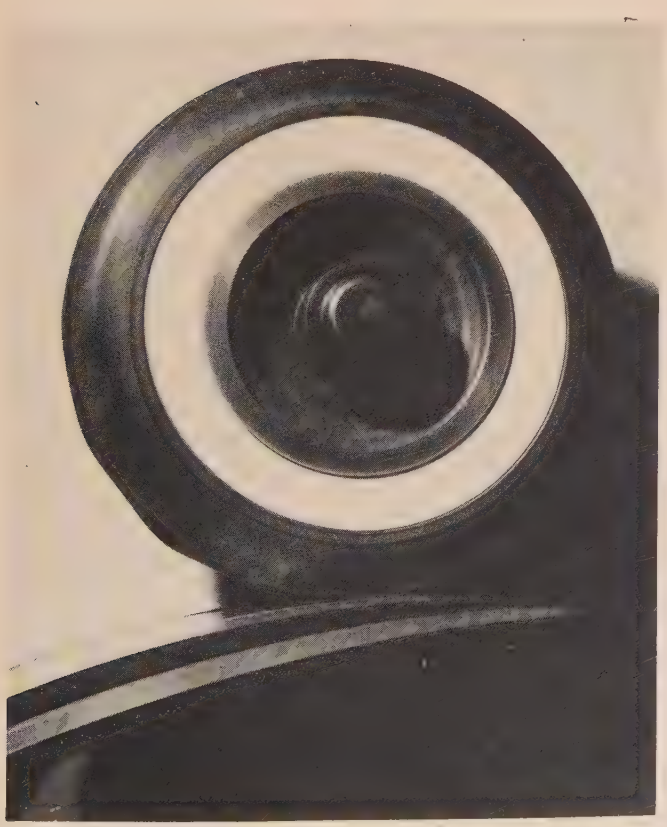

Fig. C—A regular tire running at 120 mph

Fig. D—The XP-140 running at 120 mph

Fig. E—Regular tire running at 140 mph

improve a tire for high speed usage, three major parameters may be adjusted:

1. The tire weight can be reduced, particularly in the crown region.

2. The cord angle can be reduced; that is, made more nearly circumferential.

3. The inflation pressure can be increased.

In developing our XP-140 tire, we have felt that the high-performance tire buyer will still do a major part of his driving at moderate speeds and the general performance should not be sacrificed significantly in gaining high-speed performance. We believe that this has been accomplished in the engineering of this tire. The XP-140 is safe at 130 mph with 24 psi inflation and at 140 mph with 35 psi inflation and it still is an unusually fine handling tire with good durability at normal highway speeds.

The figures illustrate the comparison of our original equipment Royal tire with the XP-140. All these pictures were made with normal (T & R) load and 24 psi inflation.

Fig. A shows a 7.60-15 Royal tire running at 100 mph on our 120-in. dynamometer. You will note the beginning of the standing wave condition in this tire.

Fig. B shows the XP-140 under the same conditions and no standing wave is evident.

Fig. C shows the regular tire at 120 mph. Here the standing wave has become quite pronounced.

Fig. D shows the XP-140 at 120 mph and no standing wave is evident.

In Fig. E the standing wave is very evident in the regular tire extending at least half around. This is the condition in the regular tire at 140 mph.

Fig. F shows the XP-140 at 140 mph and 24-lb inflation. The standing wave is starting to develop. When the inflation pressure is increased to 35 psi, the standing wave in the XP-140 at 140 mph disappears.

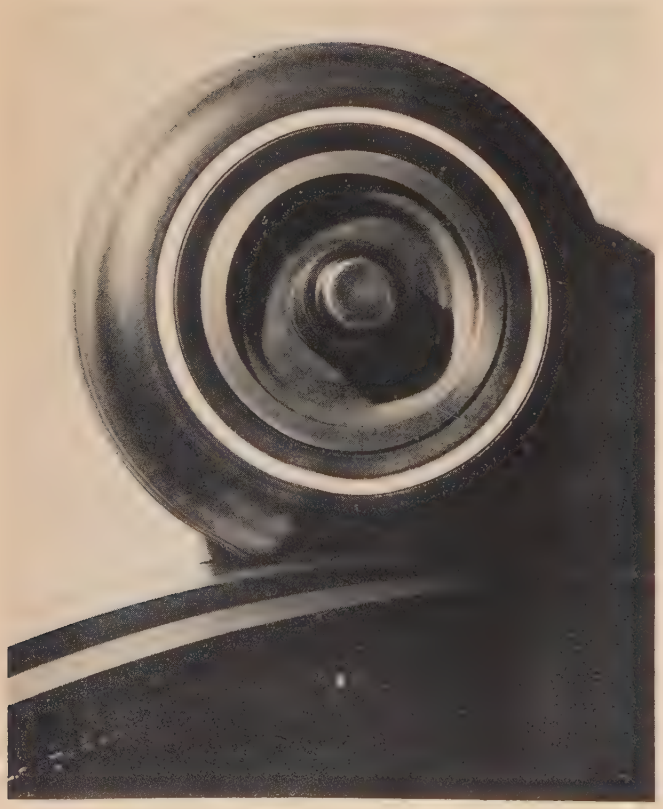

Fig. F—The XP-140 at 140 mph and 24-lb inflation

Effects of High Speeds on Tires Found Through Road Testing Results in Improved Tires

—M. A. Wilson
Goodyear Tire and Rubber Co.

IN examining Mr. Powell's paper, I find no points on which our experience results in strong disagreement. The comprehensive outline of the testing methods described reflects the thoroughness with which his company has studied high-speed tire-performance characteristics, the steps they have taken to establish equipment and procedures needed to evaluate these features, and the development of tires needed to meet their requirements. We know high-speed cars with special tire requirements have been used in Europe for many years so that there is a wide background of experience over a considerable period of time.

There is no question but that in this country also, the very definite trend over recent years toward higher-powered cars with greater speed potentials has necessitated steady improvement in high-speed tire performance in order to keep pace, even though our speed laws have not been noticeably raised.

Our experience here in developing and testing tires having suitable high-speed performance somewhat parallels that of Dunlop Rubber Co., in the respect that laboratory or indoor testing is a necessary and important part of tire-proving technique. The preliminary evaluations obtained in the laboratory can be very influential in determining the construction features which should be given more extensive and time-consuming tests in actual road conditions.

In respect to road testing, although our highway system in some parts of the country is more suitable for high-speed operation than in the United Kingdom, it may not be more ade-

quate than that which exists in certain parts of Europe and, in addition, it generally has more severe limitations on legal speed. As a matter of fact, high-speed testing on the road, which we agree is a most essential part of development and performance evaluation, has become an increasingly difficult problem as speeds of cars have increased. This is because of restrictions on the use of public highways, and the alternative seems to be still greater use of self-contained closed tracks with necessarily banked curves, which unfortunately introduce other undesirable variables.

An interesting comment included in the road-test portion of Mr. Powell's paper was "ballast is added to the car in addition to the crew of driver and observers to bring up the load to the full rating." As has been pointed out before, one of the important factors in high-speed performance is tire overload which on standard-production American cars has had quite a tendency to increase on front axles during the recent years. This overload trend does not seem to be nearly so marked on European cars.

The illustrations given by Mr. Powell on the results of high-speed road tests suggest that "tread stripping," or as we would call it here, tread separation or tread throwing, was the primary cause of the failure. This feature formerly carried similar importance here, but as improvements in compounds and constructions have been made, we find tread chunking is now considered to be the major limiting factor in continued high-speed operation of standard types of tires.

It is our experience that tread chunking is not confined entirely to the results of high-speed operation, but in some cases it is preceded by the tearing or cutting of tread elements resulting from wheel spinning and highly accelerated forward motion of the car, extremely sharp cornering, or rapid deceleration and wheel skidding. This type of driving seems on the increase along with the trend toward higher performance cars.

It may be of interest to give briefly the tire story, during the past several years, in connection with high-speed stock-car endurance runs on the Bonneville Salt Flats. These runs were under the sanction of the American Automobile Association, and involved strictly stock equipment throughout.

Tire pressure was optional, and we found that 40 psi cold, with build-up of 7 to 10 psi depending on day or night temperature, was satisfactory and probably optimum. In these runs, average speed exceeding 111 mph for 1,000 miles, exceeding 108 mph for 24 hours, and more than 101 mph for 7 days have been attained.

The tires accept these conditions of operation without carcass damage or apparent deterioration; the treads do not separate, and tread wear on the salt is slow. Replacement of tires during pit stops were made at tire mileages ranging from 1,000 up to more than 12,000 miles, depending in considerable degree on wheel position. Damage to the tires may be described as a sort of progressive tearing of the tread elements, the rate and extent of which is governed in large degree by the roughness of the 10-mile circular track, and usually resulted in tread chunking. Although the salt surface does not have the temperature and friction characteristics of a paved road, continued high-speed operation over a distance of some miles is typical in certain areas of the United States.

Disregarding the several types of racing tires which are made with special constructions, the tendency in the United States is to expect all standard- or premium-priced tires to give adequate high-speed performance in those cases where cars are operated at speeds well above recognized legal limits, in addition to the properties of good ride, excellent traction, freedom from road noise and cornering squeal, long tread wear, and safety and durability, as desired by the majority of car owners.

Perhaps this is an appropriate time to point out that some of those features which result in the best tire performance in the overall respects are also ones which are in conflict with those necessary for extremely high-speed service over an extended period.

I am quite sure that all major tire companies are well acquainted with these requirements and are producing standard tires intended to be fully adequate to meet them. In addition, companies are producing premium tires having still more high-speed ability with a minimum loss of other features.

Our experience during the past several years has included a very substantial use of nylon in premium-priced tires and more recently in standard original equipment tires used on some makes of cars. It is our opinion that nylon, because of its strength and heat-resisting characteristics, which provide comparative freedom from carcass failure and superior high-speed performance, will become the tire cord of the future for the better grades of passenger-car tires in America.

ORAL DISCUSSION

Reported by W. H. deBruin

Goodyear Tire and Rubber Co.

B. D. Mallory, Goodyear Tire & Rubber Co.: Does the 15-mph differential in the results received from highway versus the laboratory dynamometer exist with tires that were run in contact with each other on the dynamometer rolls?

Mr. Powell: Tires run against each other in this manner do not show the same results as the tires which are run directly against a metal dynamometer roll or on the road. Not enough work has yet been done to establish the difference between these methods of indoor testing.

O. J. Wolfer, Pure Oil Co.: What effect does a driven wheel have on the "standing wave effect?"

Mr. Powell: There are no data on this question, but theoretically the effect of traction is to lower the speed at which the standing wave occurs.

M. P. Hershey, Firestone Tire & Rubber Co.: The overload on the front wheels has more effect than traction has on the rear wheels, with regard to the standing wave, since a front tire usually fails first from high speed.

L. Lipkin, Ford Motor Co.: Is the standing wavelength a function of speed, or tire design, or both?

Mr. Powell: The standing wavelength has no connection with speed. It is dependent entirely upon the circumferential stiffness of the tire structure, which is related to the design of the tire carcass.

Question: How are temperature measurements made on tires?

Mr. Powell: A needle type of thermocouple is used on the road by a team of experienced technicians who are able to make six temperature measurements on each of the four tires before cooling effects of the air temperature take over. With this type of thermocouple, a pointed needle can be quickly inserted to the proper depth of the tire tread immediately after the vehicle comes to rest following a high-speed run. The cooling effects can be calibrated for the first few minutes after the wheel is stopped, which allows the number of readings indicated above.

Fundamental Investigation of Noise Generation by Turbulent Jets

N. D. Sanders and F. C. Laurence,
Lewis Flight Propulsion Laboratory, NACA

This paper was presented at the SAE National Aeronautic Meeting, New York, April 11, 1956.

OUR laboratory is engaged in research on various aspects of the jet noise problem. One phase of this research is an investigation of various devices or modifications to jet-engine nozzles in a search for some method of reducing jet noise. Similar programs are being conducted by airplane and engine manufacturers and by British research organizations. These programs have produced some very promising results particularly with respect to the reduction of noise while the airplane is on the ground. The experiments show also that some reduction in noise is possible while the airplane is in flight. Estimates indicate, however, that a very large reduction in noise will be necessary in order to lower the noise from jet transports to the levels currently produced by reciprocating-engine transports. Experiments so far have failed to attain these large reductions in noise, principally because of a lack of knowledge concerning the fundamental mechanism by which turbulent jets create noise. Therefore, the NACA has undertaken a fundamental theoretical and experimental investigation to determine the nature and the details of the mechanism of noise creation by turbulent jets. This paper is a progress report on the status of this investigation.

Lighthill's Analysis

Lighthill, in England, has made very notable progress in the analysis of the mechanism relating the turbulence in jets to the sound produced.[1,2] Because of the importance of this work, a brief review of some of the most important points in Lighthill's analysis will be given. Lighthill assumes that the turbulence structure of the jet is already known. This turbulence structure must be determined by experimental means. Fig. 1 is a cross-section of an idealized jet. Air discharges from a cylindrical nozzle

[1] Proceedings of Royal Society, Vol. 211, No. 1107, March 20, 1952, pp. 564-587: "On Sound Generated Aerodynamically. I—General Theory," by M. J. Lighthill.

[2] Proceedings of Royal Society, Vol. 222, Feb. 23, 1954, pp. 1-32: "On Sound Generated Aerodynamically. II—Turbulence as Source of Sound," by M. J. Lighthill.

into a surrounding medium which is at low velocity or at rest. Mixing occurs between this high-velocity jet and the surrounding air. This mixing process is extremely turbulent and produces fluctuating velocities. Fluctuating pressures accompany these fluctuating velocities, and sound waves are radiated. Lighthill points out that the moving field of turbulent velocity may be simulated by an equivalent stationary field of quadrupoles. This suggestion is an application of a well-known practice in the fields of acoustics and aeronautical engineering. An example is the case where the flow around an airfoil is simulated by a combination of sources, sinks, and uniform flow. A class of particular solutions to the equations of motion of the fluid is selected, and then the desired solution is obtained by properly summing these particular solutions. Lighthill shows that, in a region remote from boundaries or sources of fluid, a quadrupole is the lowest order of particular solution that can be used because of considerations of momentum and conservation of mass.

Lighthill derives equations relating the strength of quadrupoles to the fluctuating shear within the turbulent region, and these equations give the value of the fluctuating density in sound field. Unfortunately, the quantities appearing in these equations are extremely difficult to evaluate experimentally.

NACA Analysis

Because of the experimental difficulties in applying Lighthill's formula directly to experiment, the NACA has undertaken an analysis that differs somewhat from Lighthill's. At the beginning of the analysis Lighthill's principal suggestion is used; that is, the moving field of fluctuating velocities is simulated by an equivalent field of stationary quadrupoles. At this point, however, the NACA analysis differs from that of Lighthill. Equations are derived that relate the quadrupole strength to the conveniently measurable properties of the turbulent jet. Particularly, quadrupole strength is stated in terms of the components of velocity, which are easily measured by hot-wire anemometers. This analysis was deliberately tailored to fit the types of experiments that might be conducted easily.

Before continuing, it is well to take a closer look at quadrupoles. Fig. 2 shows three very simple types of quadrupoles. In each case they consist of two sources and two sinks arranged in a symmetrical manner so that there is no net flow and there is no net momentum from the combinations. The longitudinal and lateral quadrupoles are quite evident in the figure. The inclined quadrupole and other types of quadrupoles can be shown to be equal to the sum of longitudinal and lateral quadrupoles. The quadrupoles shown here are called concentrated quadrupoles.

Equations relating the quadrupole strength to the pressures, velocities, and the sound power have been derived. Some of these equations are shown in Table

EQUATIONS relating to noise outside a turbulent jet to the turbulent velocities inside the jet have been derived making use of Lighthill's suggestion that the actual flow field can be simulated with a stationary field of quadrupoles.

Hot-wire anemometer techniques were applied to a 3½-in. air jet to measure the turbulent structure of the mixing region. These measurements and the previously derived equations were used to estimate some of the characteristics of the sound field.

The estimated characteristics were found to be in good agreement with the experimentally measured sound field of a full-scale turbojet engine.

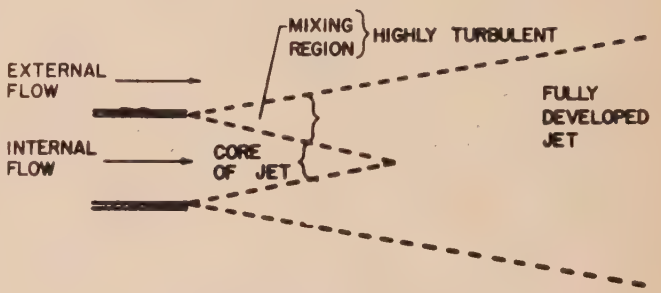

Fig. 1—Cross-section of idealized jet

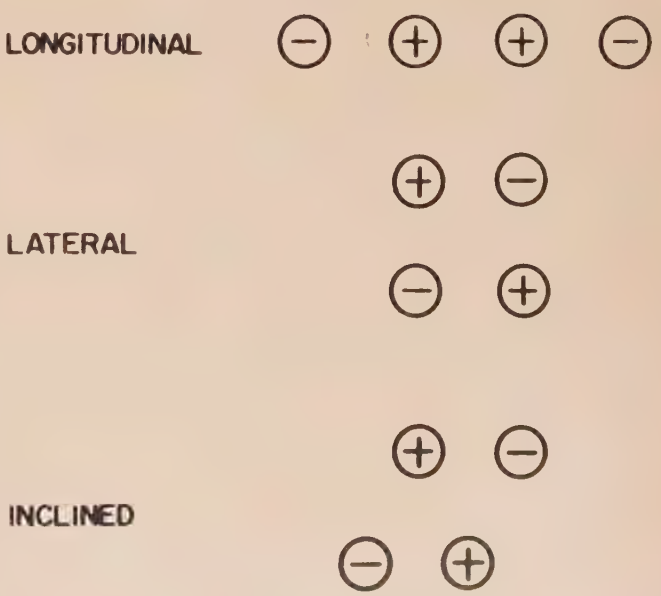

Fig. 2—Types of quadrupoles

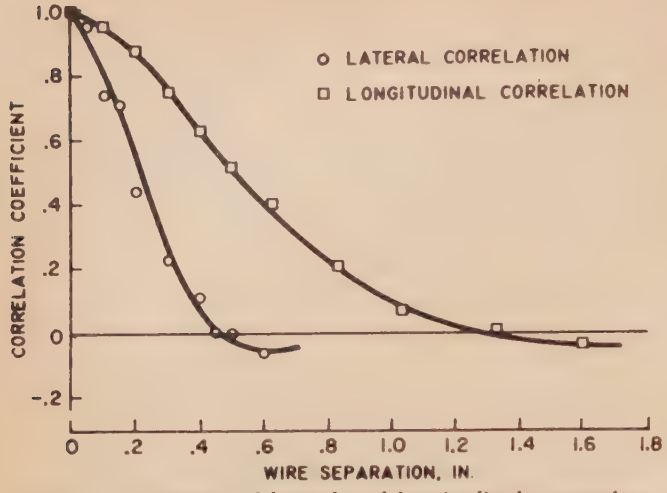

Fig. 3—Comparison of lateral and longitudinal correlation

Table 1—Equations Relating Quadruple Strength to Pressures, Velocities, and Sound Power

Near Field:	
Velocity Potential	$\phi = K_1 \frac{Ae^{i\omega t}}{a^3}$
Velocity Parallel to Mean Flow	$u_x = K_2 \frac{Ae^{i\omega t}}{a^4}$
Pressure	$p = K_3 \frac{i\rho\omega Ae^{i\omega t}}{a^3}$
Far Field:	
Pressure	$p = -K_4 i \left(\rho c\right) \left(\frac{\omega}{c}\right)^3 \frac{A}{r} e^{i\omega \left(t - \frac{r}{c}\right)}$
Sound Intensity	$I = K_5 \left(\rho c\right) \left(\frac{\omega}{c}\right)^6 \frac{A^2}{r^2} e^{i2\omega \left(t - \frac{r}{c}\right)}$
Sound Power	$P_q = K_6 \left(\rho c\right) \left(\frac{\omega}{c}\right)^6 A^2 e^{i2\omega \left(t - \frac{r}{c}\right)}$

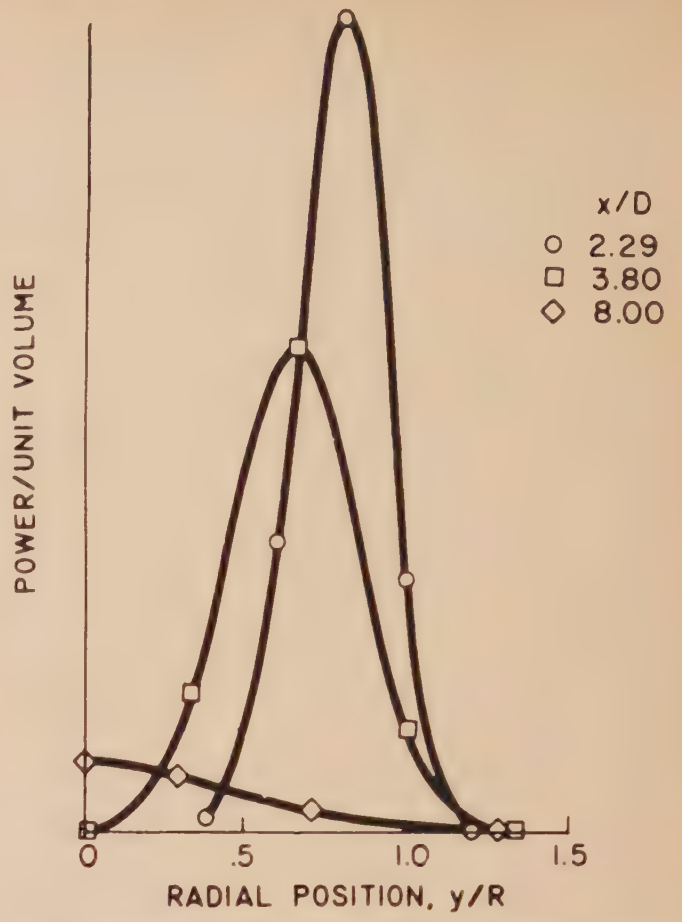

Fig. 4—Estimated radial distribution of sound power, or noise, at three cross-sections of jet

1. The equations are divided into two categories, near field and far field. The terms in the equations are defined as follows:

φ = Velocity potential

$K_1, K_2, K_3, \ldots$ = Directivity factors depending on type and orientation of quadrupoles

A = Quadrupole strength

$i = \sqrt{-1}$

ω = Frequency, radians per unit time

t = Time

a = Distance from center of quadrupole to point at which velocity potentials exist

u_x = Fluctuating component of velocity parallel to mean flow

p = Fluctuating component of pressure

ρ = Density of fluid

c = Speed of sound

r = Distance from center of eddy to point in far field

I = Sound intensity

P_q = Sound power radiated from quadrupole

A turbulent eddy cannot be represented accurately by concentrated quadrupoles for which these formulas were derived. Modifications of these formulas have been derived for distributed quadrupoles, and these distributed quadrupoles can be arranged to adequately represent a turbulent eddy.

The equation relating the horizontal component to velocity u_x to the quadrupole strength and a distance within the eddy a is used in connection with the observed hot-wire anemometer measurements to estimate the quadrupole strength. The quadrupole strength may then be used with the far-field equations to estimate the sound power and sound pressures.

The equations just discussed must be modified in terms of statistically observable characteristics of the flow. The human ear responds to certain statistical properties of the sound, the rms value of the pressures, and the spectrum of the pressures. Vibrating panels in an aircraft structure also analyze sound on a spectral basis and respond to the rms value of the pressures. Consequently, within the turbulent regions, rms values of velocities and the spectrum of velocities will be considered. This procedure greatly simplifies experimental techniques.

The statistical sizes of turbulent eddies are determined by familiar correlation techniques. Two hot-wire anemometers are used, one being held stationary and the other being moved. If u_1 and u_2 are the instantaneous velocities measured by the two instruments, then the correlation coefficient R is:

$$R = \frac{\overline{u_1 u_2}}{\sqrt{\overline{u_1^2}}\sqrt{\overline{u_2^2}}}$$

The value of R is plotted as a function of the wire separation in Fig. 3. The size of the eddy is frequently stated in terms of the scale of turbulence L, which is defined by the relation:

$$L = \int_0^0 R\, dz$$

where z is the wire separation. This integral is the area under the correlation curve.

Correlations have been obtained along three orthogonal lines at selected points within a jet. One correlation is along a line parallel to the mean flow, and the other two are perpendicular to the mean flow. The two lateral correlations extend equal distances, but the longitudinal correlations extend more than twice the laterals. These results indicate that the shape of a turbulent eddy can be approximated by an ellipsoid of revolution with the major axis parallel to the main flow. From this information, the type and orientation of quadrupole can be determined.

Correlations, spectral distributions, and intensities of turbulence have been determined at a large number of points in the field of the 3½-in. diameter air jet over a range of Mach numbers from 0.2 to 0.7.[3] Using this information and the theoretical methods previously described, the distribution of the sound power and pressures in the field produced by the jet has been estimated. Details of this analysis are too involved for a complete discussion at this time.

Results of Analysis

The sound power generated per unit volume of jet has been estimated at many points of the mixing region of the jet. Fig. 4 shows the estimated radial distribution of sound power at three cross-sections of the jet. One of these cross-sections is approximately halfway down the core of the jet, the second section is at the tip of the core, and the third section is farther down the stream. At the station closest to the nozzle, the noise distribution is sharply peaked. The peak of the curve is located less than one radius from the centerline. Proceeding from the nozzle downstream, the intensity of noise per unit volume at the peak of the distribution curve falls and moves toward the centerline of the jet. At the station farthest downstream, the noise power per unit volume has fallen to a quite low value, and the maximum value of the curve lies near the axis of the jet.

The noise power generated per unit length of jet is obtained by integrating the power per unit volume over the cross-section of the jet. The results of this integration are shown in Fig. 5, in which the power per unit length of jet is shown as a function of the position along the jet. Near the nozzle the noise power generated per unit length of jet is approximately constant. At four diameters the noise power has fallen slightly, and at eight diameters the noise power is quite low. Beyond 10 diameters very little noise appears to be generated.

The results shown in Fig. 5 give some clue concerning the variation of sound pressures along the boundary of the jet. The noise power generated per unit of jet length is seen to be a constant; therefore, it is expected that the sound pressures at the boundary of the jet will also be approximately constant as the microphone is moved along the jet boundary. Experimentally measured values of sound pressures at the boundary of a jet from a full-scale turbojet engine are shown in Fig. 6. The sound pressures are constant

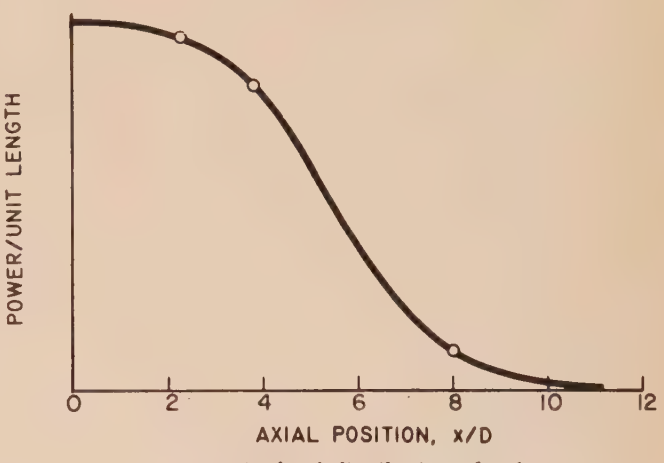

Fig. 5—Longitudinal distribution of noise

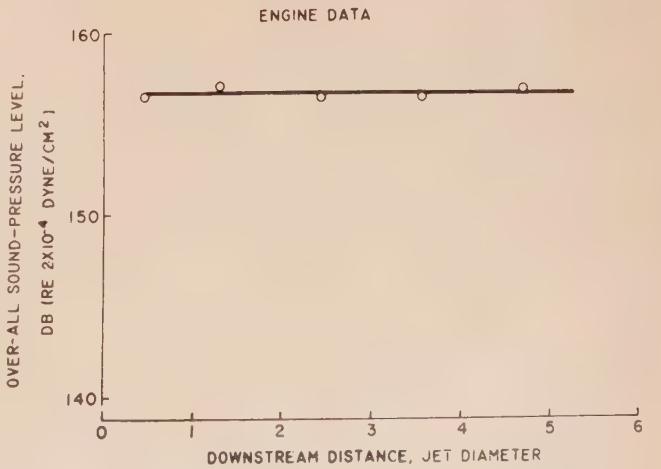

Fig. 6—Overall sound-pressure level along jet boundary

3 NACA TN 3561 (1955), "Intensity, Scale, and Spectra of Turbulence in Mixing Region of Free Subsonic Jet," by J. C. Laurence.

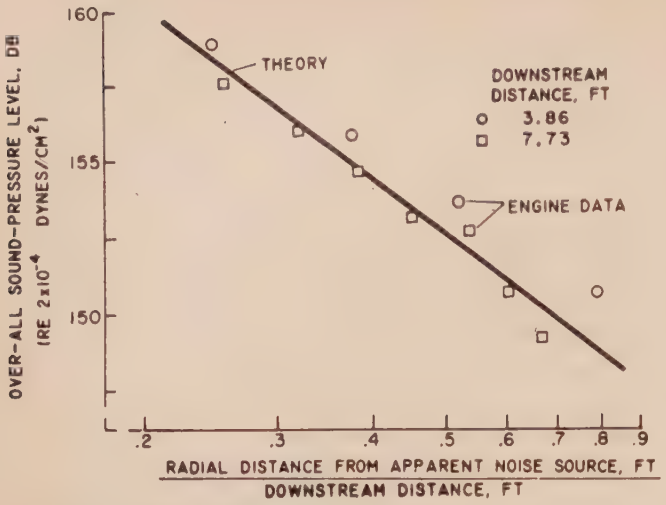

Fig. 7—Overall sound-pressure level as function of dimensionless distance ratio

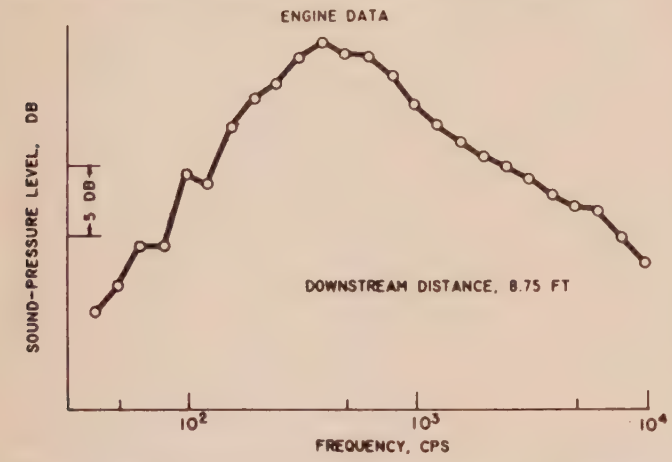

Fig. 8—Experimentally observed noise spectrum at jet boundary

for some distance along the boundary of the jet, and this result very nicely confirms the prediction of the theory.

The variation of sound pressures in the near field along a line perpendicular to the jet axis can also be predicted by the theory in a rather crude manner. An exact analysis of the problem is very difficult; however, a first approximation of the theory indicates that the sound pressures should vary inversely with the distance from the origin of the sound. Fig. 7 shows results of measurements of the sound pressures near the jet of a full-scale turbojet engine. The data are shown in terms of the overall sound-pressure level as a function of radial distance from the apparent noise source. It will be recalled (Fig. 4) that the sound pressures are concentrated in a sharply peaked band. The noise source was assumed to be located at the peak of these curves. The distances used are from this apparent source of the noise. A line with a slope corresponding to the inverse variation of pressure with distance has been drawn on the graph. Although there is not perfect agreement between the experiment and the theory, the data do fall reasonably well along the expected slope. It is believed, therefore, that the experiment and theory are essentially in agreement.

With regard to the frequency spectrum near the boundary of the jet, the theory predicts that a spectrum will be sharply peaked. Fig. 8 shows an experimentally observed spectrum near the boundary of the jet from a full-scale engine, which does show the shape predicted by the theory. The theory has not been developed sufficiently at this time to predict the slopes of the two sides of the spectrum. The theory also predicts that the frequency at the peak of the spectrum will vary inversely with the distance from the lip of the nozzle. Fig. 9 shows the experimentally observed variation of the frequency at the peak of the spectrum with the distance from the nozzle. Although the experimentally determined curve does not have a slope of -1 which corresponds to the expected inverse variation, the observed variation does agree approximately with the expected variation; and again the theory and the experiments are essentially in agreement.

The theory can also be checked by measurements in the far field. The analysis indicates that the total sound power radiated from the jet should vary as the eighth power of the jet velocity and the cross-sectional area of the jet. The same conclusion has been reached earlier by Lighthill. Experimental verification of the prediction is shown in Fig. 10. The total acoustic power has been measured from air jets having diameters of 3, 4, and 5 in. operating at various jet velocities and from two full-scale engines operating at various power settings with and without afterburning. The sound power is plotted as a function of the Lighthill parameter, which includes the eighth power of the jet velocity V^8, the cross-sectional area of the jet S, and the density ρ_0 and speed of sound c_0 in the surrounding medium. A line with a slope corresponding to the predicted variation has been drawn through the data. The data from the air jet and the full-scale engines under most of the operating conditions agree extremely well with the predicted variation. One point corresponding to a high-pressure-ratio engine operating with afterburning does not fall on this curve. Further research is required to investigate this particular condition.

Pressure Correlations

The experimental portion of this fundamental investigation produced some interesting results that are not directly related to the theory. The results are of sufficient importance to structural designers to merit some discussion.

The designer of a structure that will be exposed to the sound field of a jet must know the rms sound pressure, the spectral distribution of the pressure, and the area over which the pressure acts. The im-

portance of the last item has been overlooked by designers.

The area over which the pressures act is determined by a correlation technique involving two microphones. Fig. 11 gives a typical pressure correlation. One microphone was placed near the boundary of the jet from a full-scale turbojet and was held in a fixed position. The other microphone was moved along the boundary, and the correlation between the two microphone signals was calculated by electronic means. The resulting curve represents the loading curve on a surface at the jet boundary.

The significance of the curve can be illustrated by considering the load on three sizes of plates. If a plate 2 in. wide is used, the correlation across the plate is essentially unity and the total load equals the pressure times the area of the plate. If the width of the plate is 8 in., the pressures at the two edges of the plate are 90 deg out of phase, the net load on the plate is proportional to the product of the pressure and the area under the correlation curve between 0 and 8 in. If the plate is 20 in. wide, the load on one portion of the plate acts in opposition to the load on the other portion, and the net load on the plate equals the net area under the curve with proper consideration for signs. Although the net loads may be small, large bending moments might exist.

Experimental research is continuing to determine the loading curves at important locations within the sound field of a full-scale turbojet. The curves are also being determined with selected frequency bands to permit calculations of vibrations in resonant panels.

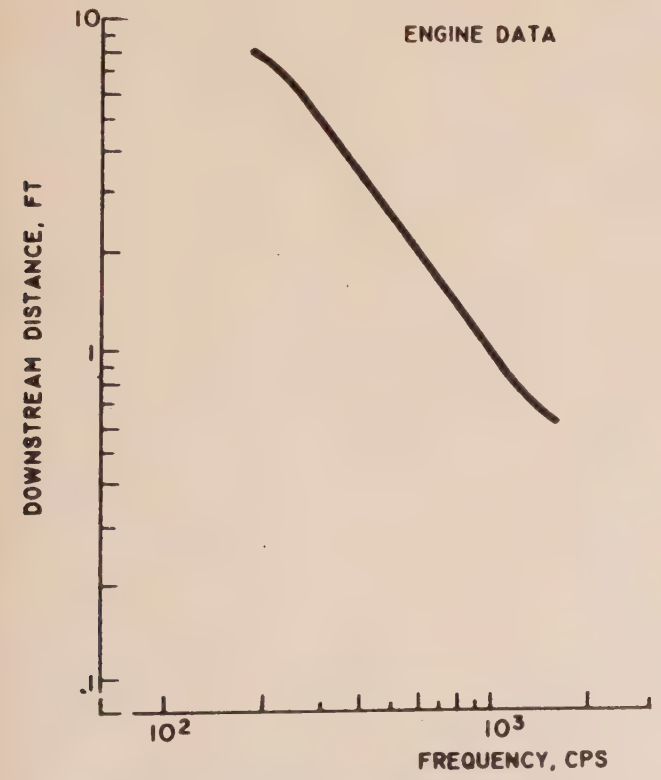

Fig. 9—Downstream distance of maximum sound-pressure level as function of frequency

Conclusions

An analysis has been developed based upon Lighthill's suggestion of the field of stationary quadrupoles. This analysis has been stated in terms of the easily measurable properties of the turbulent jet. Properties of turbulent jets have been measured and estimates have been made of the noise power and characteristics of the noise power radiated. The analysis has successfully predicted many of the important characteristics of the near field and the variation of the total sound power with nozzle size and jet velocities in the far field. The analysis is being continued particularly with respect to the light it might shed upon noise reduction.

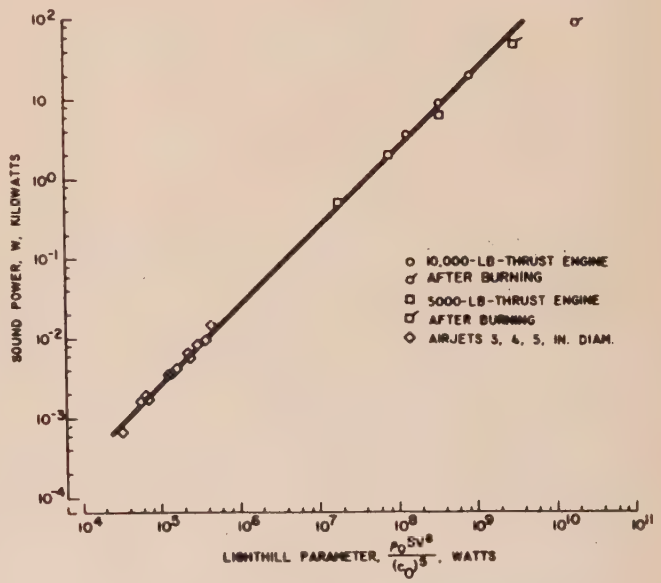

Fig. 10—Sound power as function of Lighthill parameter

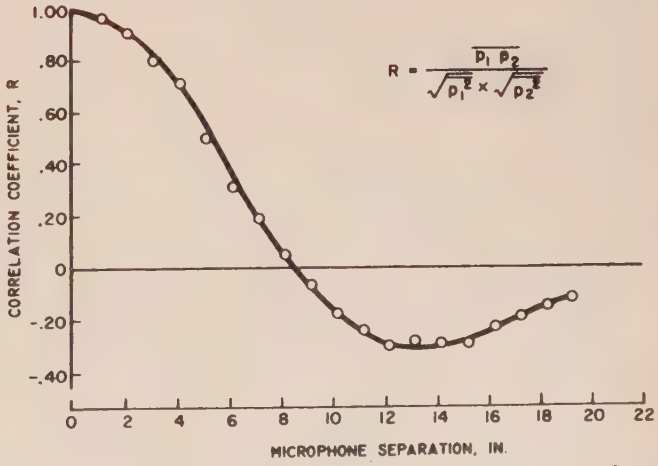

Fig. 11—Correlation of overall sound pressure. Distance from engine along jet boundary, 4 ft

THIS paper is an elementary nonmathematical discussion of the essential thermal processes of gas turbines with emphasis on turbine types suitable for use in vehicles.

A brief history is followed by discussion of the limitations imposed by strength of materials. The effects of pressure ratio, compressor and turbine efficiency, turbine inlet temperature, and atmospheric temperature upon output and thermal efficiency are discussed. The effects of regeneration upon thermal efficiency at full and part load, with particular emphasis upon the latter, are discussed, and the regenerator is stated to be essential to obtain acceptable fuel mileage in vehicle use. The probable performance parameters of a vehicle turbine of about 200 rated hp are explored.

SOME ELEMENTS

IT is not often that an author is chosen for a paper on the grounds that he knows so little about the subject, but this is literally true in the present case. As a matter of self-defense, the background of the paper may well be stated. About two years ago the author prepared a discussion of the essential processes of the gas turbine and of some of the problems of applying it to road vehicles. The discussion was intended for those piston engine engineers who had paid no previous attention to the gas turbine. The discussion was produced in a very small edition and was given very limited circulation. In the process of obtaining reactions to the discussion the author circulated it to various friends and was incautious enough to send a copy to Maurice Olley, who must be blamed for the fact that I was asked to prepare this paper.

The author having, to his regret, no mathematics and only a very limited and slim knowledge of thermodynamics, spent a year educating himself in the matter of computing turbine performance possibilities by means of simple arithmetic. In view of these limitations the author could obviously only discuss turbine performance in very simple terms, and this was the apparent cause of Mr. Olley's approval. To the author's surprise, two internationally known aircraft turbine engineers found the discussion useful, apparently because it dealt with turbine performance without mathematics.

Until three years ago the author was quite unable to compute the specific air consumption of a gas turbine. He was introduced to entropy more than 45 years ago, did not understand it clearly then, does not understand it all now, and does not propose to educate himself in the subject at the age of 65. The simple arithmetic used in the paper is, of course, only possible as a result of the elaborate and precise mathematical analysis of Keenan and Kaye given in their Gas Tables. The author's mathematical ignorance has not blinded him to the fact that the gas turbine, as we know it, is in no small measure the product of the mathematician and the aerodynamicist. This preface is, of course, an alibi but the author does not either expect or hope that it will save him from the scorn of the gas turbine authorities.

Introduction

Recent publicity releases concerning experimental installations of gas turbines in passenger cars have aroused considerable interest in such powerplants. In addition, there is much interest in the probable fuel consumption of such engines.

Much of the interest in gas turbines for road vehicle use is, of course, a reflection of the outstanding success of the turbine in aircraft and particularly so where jet propulsion is involved.

In the aircraft field it has been stated that the propeller gas turbine can equal or better the specific fuel consumption of the gasoline piston aircraft engine. Specific fuel consumptions of 0.4 lb per hp-hr are claimed for the propeller turbine, but it is worthy of note that such figures are only claimed for operating altitudes of the order of 35,000 ft and airspeeds of the order of 400 mph. The propeller turbine is admitted to be inferior to the piston engine in respect to specific fuel consumption at low altitudes and air-

OF GAS TURBINE PERFORMANCE

S. D. Heron, Consulting Engineer

This paper was presented at the SAE National Passenger-Car, Body, and Materials Meeting, Detroit, March 7, 1956.

speeds. While inferior to the piston engine in this respect, it is nevertheless vastly superior to the jet engine under these conditions. While the propeller turbine is inferior to the piston engine at low speeds and low altitudes in respect to specific fuel consumption, it nevertheless has very real advantages (which need not be discussed here), and can be expected to come into increasing use for applications where the extreme airspeed possibilities of the jet are not required. It should be pointed out that present propeller turbine installations are in part jet-propelled, since a minimum of about 10% of the effective thrust power in flight is due to the turbine exhaust.

The discussion below carefully avoids any predictions as to the probability of the gas turbine coming into large-scale use in ground vehicles. The author has received hints that he should emphasize the unsuitability of the gas turbine for road vehicles and other hints that he should point out its marked advantages over the piston engine. The author's limited studies of the gas turbine have in no sense qualified him for the role of a prophet. The turboprop aircraft engine is a good example of the dangers of prophecy. If it had been predicted 10 years ago that turboprops would be flying this year with a specific fuel consumption of 0.40 lb hp-hr, the prophet would have been considered of doubtful sanity. That such a fuel consumption will be obtained in flight this year appears to be extremely probable. This turboprop example is merely included to illustrate the dangers of prophecy and not to hint that the gas turbine may equal or beat the fuel consumption of the piston engine in ground vehicle use. The high-speed, high-altitude airplane is peculiarly suited to the characteristics of a gas turbine. The characteristics of an aircraft turbine are such as to make it largely unsuitable for vehicle use, but this does not condemn the turbine, since it is apparent that the vehicle turbine, if successful, will be basically different from the aircraft type.

The recent large orders for Douglas and Boeing jet-propelled airliners will doubtless stimulate public interest in turbine-engined ground vehicles and provide the daily press with further opportunities for references to "pet-propelled automobiles." The fact that these airliners are to be jet-propelled no more proves that the gas turbine is the ideal engine for a ground vehicle than does the successful use of the gasoline engine in passenger cars prove that it is the ideal engine for the main propulsion units of 6000-hp freight locomotives. Each type of powerplant has its place, and the diesel engine, despite its success in locomotives, is hardly ideal for the main propulsion units of a 200,000-hp ship.

The author has been responsible for all computation involved in the text and illustrations (others have, however, checked the data). In view of the author's limitations it is obvious that some phases of turbine performance are either incompletely treated or neglected altogether. Some of the author's associates have kindly offered to compute performance data which are beyond his understanding but he has preferred to limit the contents of the paper to his own knowledge (or what may seem to be knowledge).

The paper, in the main, is limited to discussion of the broader aspects of turbine performance in road vehicle applications and does not discuss mechanical

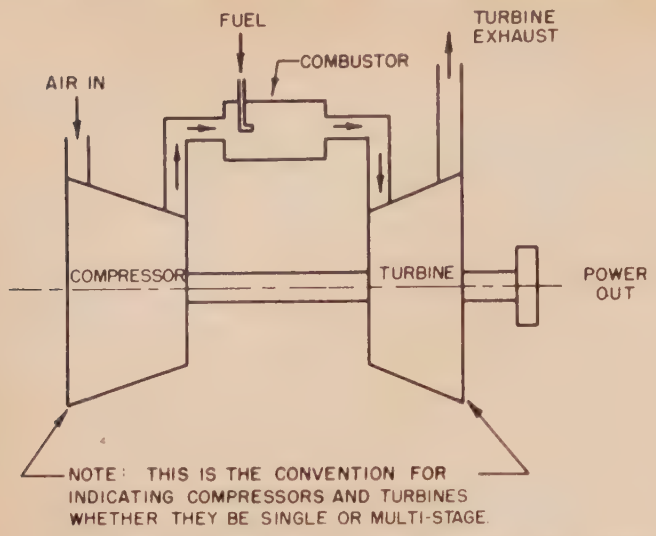

Fig. 1—Diagrammatic layout of simple open-cycle gas turbine

detail more than is necessary to outline turbine characteristics. The author has no design or development experience with gas turbines. He has had some experience with aircraft turbine materials since 1928 and with aircraft turbine fuels since 1945.

While prophecy concerning the future of the turbine in road vehicles has been avoided, certain predictions have been made as to the most likely type of turbine for vehicle applications.

Present Position of the Gas Turbine

The idea of the gas turbine is quite old, but its reduction to a successful power unit is almost entirely the product of two modern developments, namely, efficient compressors and materials with sufficient strength at high temperatures. The first commercially successful gas turbine can be said to be the Buchi turbosupercharger (patented about 1905) and first used with diesel engines about 1911. In this case the turbine inlet temperature was and is low and compressor efficiency not very important, since it is (or has been) a question of getting something for nothing as the energy in the gases is otherwise wasted.

Probably the first commercially successful gas turbine developing shaft power on its own rather than as an adjunct of a piston engine was the Brown Boveri industrial turbine [1,2] of about 1938. This unit had a thermal efficiency of about 18% at full load (about 0.75-lb oil per shaft-hp-hr) which was achieved with the low turbine inlet temperature of about 1000 F. The turbine drove an a-c generator and, of course, idled at synchronous speed (3000 rpm). It is of interest that fuel gallonage used at no load was 42% of the volume used at full load (4000 KW). Brown Boveri also, somewhat earlier, produced another successful gas turbine in the form of the Velox boiler. In this device a compressor supplies air under pressure to the pressurized firebox of the boiler in which oil is burnt. The products of combustion are cooled by the boiler and then pass to a gas turbine which drives the compressor. The Velox boiler is of academic interest, since it is the only known open-cycle gas turbine in which the products of combustion pass through the turbine and which can completely burn all the air.

While the Buchi supercharger, the Velox boiler, and the Brown Boveri industrial turbines were the first commercially successful gas turbines they all avoided the major difficulty of the gas turbine as we know it today by operating at relatively low turbine inlet temperature. The first successful gas turbine operating at high inlet temperature was the turbosupercharger used to supercharge gasoline piston aircraft engines. This was not commercially successful until about 1946 but was a military success by 1940. This again is an old idea first produced in France by Rateau about 1915. Its success was almost entirely due to a long and arduous struggle carried on by the General Electric Co. and the U.S. Army Air Force. A gasoline engine of about 6/1 compression ratio commonly has an exhaust temperature of about 1900 F. Even with considerable cooling of the gases between the exhaust valve and the turbine, inlet temperature could be as high as 1800 F, and this led to endless difficulties with the turbine blades and wheel. In the early units the turbine wheel was enclosed and the exhaust gases led to atmosphere through an exhaust pipe. The big step which saved the exhaust turbosupercharger from death was that of installing the turbine wheel in the open beyond the cowling and with the nozzle box flush with the cowling. This life-saving idea, developed about 1923, was due to the late E. T. Jones of the U.S. Army Air Force. While Jones' idea saved the turbocompressor from almost immediate death, the turbine wheel and blades still gave much trouble. Jones, by reducing wheel and blade temperature, had made the materials much stronger.[3] This was, however, not enough (although enough to keep the device alive) and it was not until about 1942 that blade material of adequate strength became available in the form of precision-cast (lost-wax process) blades of the dental alloy vitallium (stellite 21).

A turbine connected to the exhaust of a gasoline engine is still not an easy problem although reliable and durable installations are in service. The Wright turbo-compound engine, for example, uses aircooling of the turbine wheel and blades.

When, in 1935, Whittle in England and von Ohain

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1 *Engineering*, March 3, 1939, pp. 247–250: "The Combustion Gas Turbine," by A. Meyer.

2 *Engineering*, Jan. 5, 1940, pp. 1–4: "Load Tests of a 4000 KW Combustion-Turbine Set," by A. Stodola.

3 The writer knows the circumstances intimately since he was associated with Jones at the time of the invention which was never patented and never adequately credited to Jones.

4 "Development of Aircraft Engines and Aviation Fuels," by R. Schlaifer and S. D. Heron. Pub. by Harvard Business School, Boston, 1950. In this Schlaifer gives the history of aircraft turbine development in England, Germany, and the United States. His history is probably the most convenient for the reader who is interested in the broad aspects of the subject.

and Wagner in Germany almost simultaneously started their three developments of jet-propulsion aircraft gas turbines,[4] their problem was easier metallurgically than that of the aircraft turbosupercharger, since jet propulsion for aircraft was feasible at much lower turbine inlet temperatures than those involved in the gasoline engine turbosupercharger.[5] Part of this advantage was lost because the turbine wheel and blades had to be enclosed so that the turbine exhaust could be ejected by the jet pipe. Likewise, aircooling of the turbine wheel and blades (paralleling such cooling in the turbocompressor) was not initially employed although it may have been considered.

Beyond the fact that the turbocompressor was known to be a practical working mechanism, it does not seem to have contributed much to the early work of Wagner, von Ohain, and Whittle. It is unlikely that the turbocompressor in itself would have led to the development of gas turbines for aircraft use, although it unquestionably led to the Velox boiler and thence to the Brown Boveri industrial gas turbine. The gas turbine for vehicle use is unquestionably a descendant of the jet-propulsion aircraft gas turbine.

The Basic Problem of the Gas Turbine

The gas turbine and the piston engine have one and only one basic similarity in that both produce power by the expansion of heated gas.

Essentially a gas turbine produces power by compressing a gas which is then heated and after heating is expanded through a turbine. The gas is usually atmospheric air but need not be and may be a monoatomic gas such as argon or neon if a closed cycle is used. The gas, if air, is usually drawn into the compressor from the atmosphere and is usually heated by burning fuel in it. The products of combusion then expand through the turbine and are ejected to the atmosphere. Such a turbine is said to be open cycle and is shown in Fig. 1. There is another form of open-cycle turbine in which the products of combustion do not pass through the turbine. In this turbine (Mordell patent) shown in Fig. 2, the cycle is similar to that in Fig. 1 except that the air after compression is heated by passing through an externally-fired heat exchanger, or air boiler. Part of the exhaust air from the turbine is used for combustion of the fuel in the air boiler. There are a considerable number of variations of the open-cycle turbine and some of these are discussed below. In open-cycle turbines the power output is normally varied by changing the turbine inlet temperature. While this is, of course, practiced with aircraft, the most efficient method of reducing power output is produced by increasing aircraft altitude at constant turbine inlet temperature.

The closed-cycle turbine is basically different from the open-cycle type, and its fundamentals are shown in Fig. 3. In the closed-cycle type the gas is compressed, heated in an air boiler, expanded through a turbine, and then cooled in a heat exchanger which is usually considered to require an ample supply of cold water. After cooling, the gas is again compressed. The gas used may be air or may be inert to avoid oxidation such as nitrogen. If inert it may be one of the heavy monoatomic gases such as argon which can be shown to give higher thermal efficiency than the diatomic gases such as nitrogen and air. Power out-

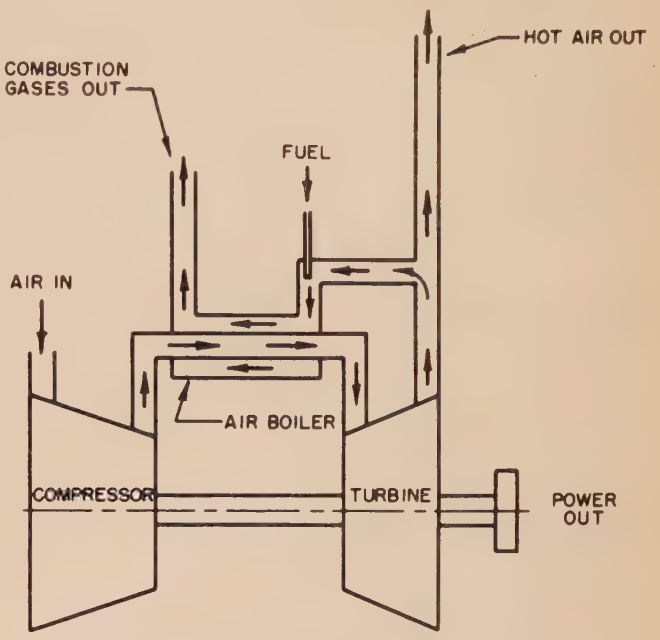

Fig. 2—Open-cycle turbine with air boiler

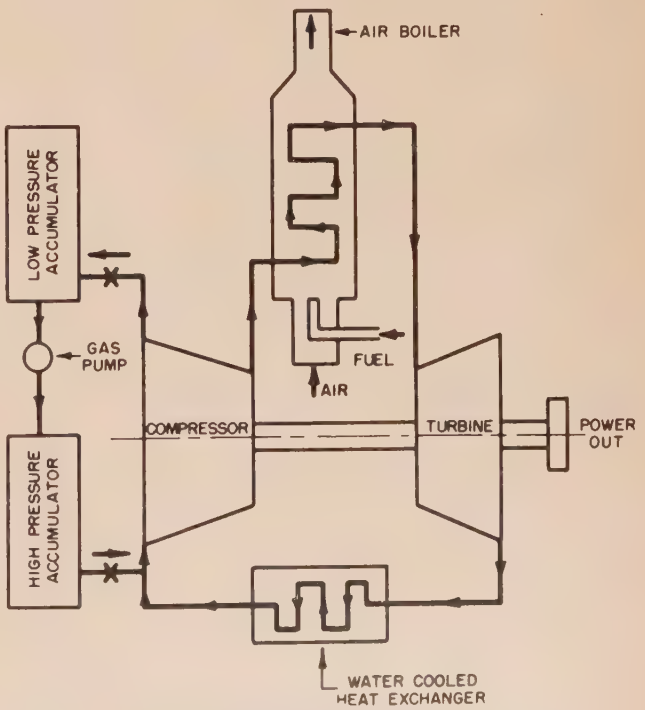

Fig. 3—Closed-cycle turbine

5 Jet propulsion is feasible at a turbine inlet temperature of 1200 F but may not be attractive. A temperature of 1500 F is presently general but up to 1700 F is used.

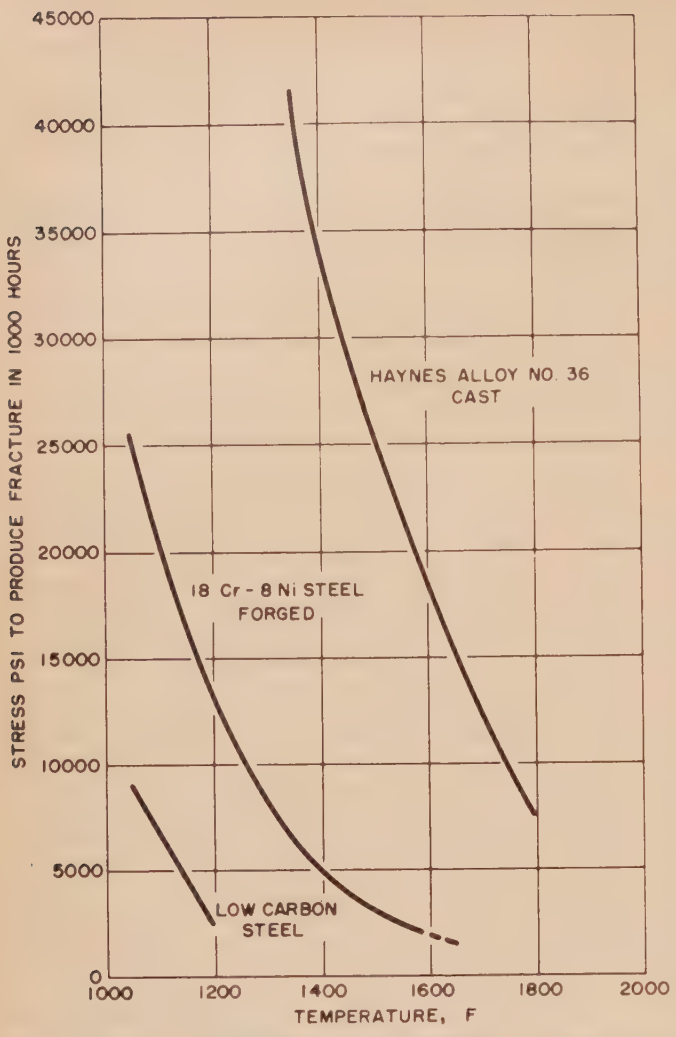

Fig. 4—Effects of temperature upon 1000-hr hot strength of actual and potential turbine materials

put is varied in the closed-cycle turbine by varying the density of the gas in the working circuit while maintaining substantially constant turbine inlet temperature. Variation of density is obtained by pumping gas in or out of the working circuit, the gas being bled from or charged into high and low pressure receivers. The turbine inlet pressure may be 300 psi or more, compressor inlet pressure 70 psi or more, and due to these pressures the turbines and compressors are much smaller than open-cycle machines of equal power output. Largely due to the fact that constant turbine inlet temperature is maintained, the part-load thermal efficiency is much better than that of the open-cycle type, except in the case of the aircraft turbine where power output is varied by change of altitude. Due to the facts that gas is recirculated and can be inert, the working circuit can be kept clean and in a high state of efficiency. Fig. 3 shows the simplest form of closed-type turbine, and it is built in much more complex types. The closed-cycle turbine in its very complex forms can apparently equal the diesel engine in thermal efficiency at both full and part load. Due to complexity and the need for quite cold cooling water, the closed-cycle turbine as a mobile power plant seems only to be suitable for marine use and will not be further considered.

Turbines of the types shown in Figs. 1, 2, and 3, in which the compressor and turbine or turbines are rigidly connected to a single shaft from which power is taken are known as "single shaft turbines." In jet engines, where the compressor or compressors and the turbine or turbines are rigidly connected and all operate at the same speed, the term "single shaft engine" is also used.

A gas turbine, at the present maximum permissible turbine inlet temperature, uses six or more times as much air as a gasoline engine of equal power output. In a gasoline engine the power output during the expansion or explosion stroke (that is, from ignition to the opening of the exhaust valve) is approximately proportional to the weight of air used multiplied by the temperature drop. This may be defined as the gross power and from this must be subtracted the power absorbed during the other strokes by work of compression and by mechanical friction. In a gasoline engine the maximum temperature may be as high as 4000 F and the temperature drop during expansion may be more than 2000 F. In a gasoline engine the gross power output will be approximately proportional to the weight of air used per stroke multiplied by the temperature drop multiplied by the number of power strikes per unit time.

In a gas turbine the gross power output, that is, the power produced by the turbine wheel itself, will be proportional to the weight of air used per unit time multiplied by the temperature drop. In the gas turbine the possible temperature drop is far lower than in the piston engine for a number of reasons. In the piston engine the maximum cycle temperature occurs only intermittently and for only a small percentage of the total cycle time and in a container with cooled walls. In a gasoline engine the maximum cycle temperature is sufficient to melt any of the materials including the spark-plug ceramic exposed to the gas. In the gas turbine, on the contrary, the maximum cycle temperature [6] is attained continuously, and many of the working parts of the turbine have to operate at a temperature nearly equal to the maximum cycle temperature. Some of the working parts, and particularly the turbine blades, or buckets, are highly stressed, and since the strength of metals drops off rapidly with increase of temperature, the maximum permissible temperature is obviously limited. In discussing the hot strength of metals and alloys the normal tensile figures have little practical significance since the materials are plastic and are continually stretching under load. Even though some materials may not stretch, they nevertheless suffer cumulative damage with time which eventually produces rupture. As a

[6] See below for discussion of the combustion process in a gas turbine.

result, the strength is a function of the length of time that the load is imposed; thus at 1500 F one of the very best turbine materials (cast Haynes alloy No. 36 which is used mostly for buckets) will fracture in 10 hr at a stress of 36,000 psi, in 100 hr at 29,000 psi and in 1000 hr at 25,500 psi. An example of the effects of temperature upon the 1,000-hr strength of actual and potential turbine materials is shown by Fig. 4.[7,8] Haynes alloy No. 36 contains about 55% cobalt, 14% tungsten, 19% chromium, and 10% nickel. The 18 chromium 8 nickel steel (hereafter 18 and 8) is not usually used for gas turbines, but is an excellent example of a material with relatively good high temperature properties. Low carbon steel is also shown in Fig. 4 for reference. In Fig. 4 it will be noted that each 150 F increase of temperature very approximately halves the strength of 18 and 8 and of Haynes alloy No. 36. By this rule, if 18 and 8 had a strength of 24,000 psi at 1050 F, it would have 12,000 psi at 1200 F, 6,000 psi at 1350 F, 3,000 psi at 1500 F and 1,500 psi at 1650 F. It will be noted that this is not far off the values shown in Fig. 4. It may be noted that in a long-life turbine operated at 1500 F maximum turbine inlet temperature, the permissible stresses will be about 50% of those shown in Fig. 4 for this temperature. Fig. 4 shows that metals lose strength at an exceedingly rapid rate with increase of temperature. This loss of strength is paralleled by an even more rapid loss of resistance to attack by oxygen and other constituents of the hot gases. There is another significant temperature problem with gas turbine materials known as thermal shock which is the tendency to crack as a result of sudden and rapid temperature changes. This is particularly liable to happen in parts of varying thickness which do not heat or cool uniformly. A turbine bucket is a good example of such a part. The road vehicle turbine with frequent and rapid load (temperature) changes is likely to be prone to suffer from thermal shock.

Fig. 4 indicates the major reason for the present limitations upon the operating temperatures of gas turbines. This figure serves to show why the maximum turbine inlet temperature is less than the temperature *drop* in the piston (gasoline) engine cycle. A gas turbine with a temperature drop of 850 F through the turbine is exceptional, and this will only be attained by an aircraft propeller turbine with the high turbine inlet temperature of 1650 F and the exceptionally high pressure ratio of 12/1. A turbine unit with an extremely efficient regenerator and only a moderate pressure ratio may, however, have a total temperature drop of about 1000 F from the turbine inlet temperature to the discharge from the regenerator. Less than half this temperature drop will occur across the turbine, however.

The above should be sufficient to show why the gas turbine must of necessity use a great deal more air than a gasoline engine of equal power output.

Components of Gas Turbine—In this discussion only the open-cycle turbine in which the products of combustion pass through the turbine will be considered. Such a simple open-cycle turbine consists essentially of three components, a compressor or compressors, a combustion section which may consist of one or many combustion chambers, and a turbine or turbines. To these essential components others may be added, such as an intercooler between the stages of a multistage compressor which can reduce the work done in compressing the air, and a heat exchanger or regenerator which transfers heat from the turbine exhaust to the air after discharge from the compressor but prior to the combustion of fuel.

Multistage compression with intercooling is not currently of interest for road vehicles. Multistage compression without intercooling is employed on the new Fiat experimental passenger car turbine and is used for all current locomotive turbines. Fig. 5 shows diagrammatically an open-cycle turbine with heat exchanger or regenerator.

Gas turbine compressors are currently of two types: the centrifugal and the axial. The centrifugal type resembles a centrifugal water pump. Such compressors are mostly single stage but 2-stage units are in successful commercial service use in civil air transports (Rolls-Royce "Dart" engine in Vickers "Viscount" airliner). The axial compressor type is now predominant in aircraft turbines. It is always multistage and as many as 16 stages may be used. An interesting axial compressor used in the Pratt &

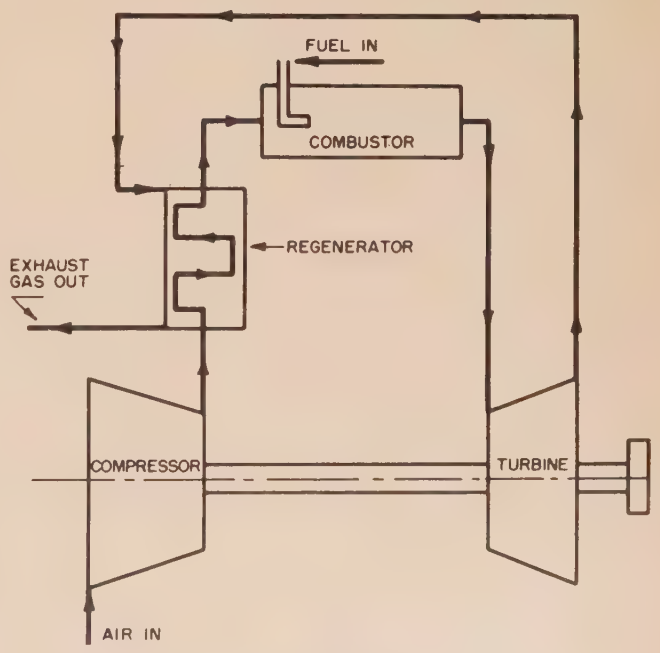

Fig. 5—Open-cycle gas turbine with regenerator or heat-exchanger

7 "Haynes Alloys for High Temperature Service," Pub. by Haynes Stellite Co., Kokomo, Ind., 1950.
8 "Stainless Steel Handbook." Pub. by Allegheny Ludlum Steel Corp., Pittsburgh, 1951.

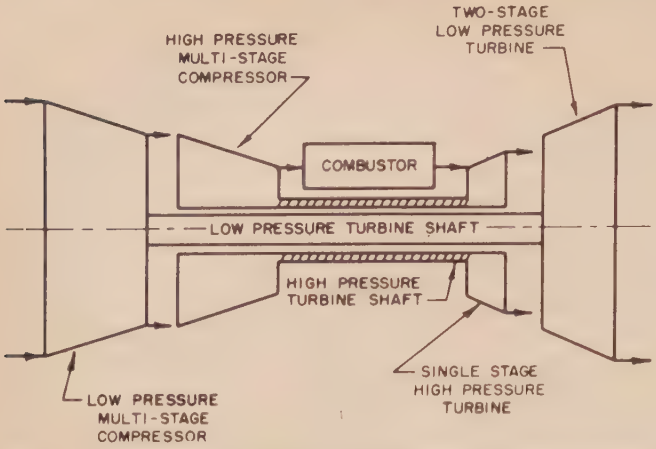

Fig. 6—Twin-spool axial compressors driven by independent turbines

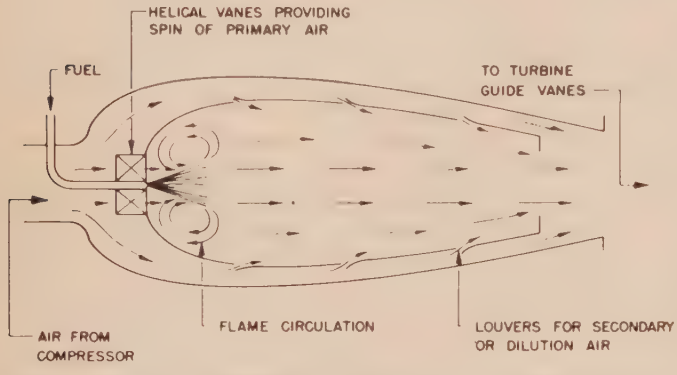

Fig. 7—Can type combustor in which fuel is sprayed into air as fine mist and burned in this form

Whitney J-57 jet engine is shown in Fig. 6, and is known as the "twin spool" type. In this type, two separate compressors, each involving many stages, are used and are independently driven by separate turbines. This arrangement gives high compressor efficiency and has other important advantages, but this is no place to discuss them.

Turbines are of two types, namely axial and radial. The axial type is predominant at present and exclusively used in the aircraft field and other fields involving high power output. Aircraft turbines presently use from one to four stages, and 10 or more stages have been used in industrial turbines. The axial type with a large number of stages closely resembles the reaction type (Parsons) steam turbine used in large electric generating stations. The radial type closely resembles a centrifugal compressor but with the gas flowing inwardly rather than outwardly. It possesses the virtue of simplicity (and probably low *relative* cost of construction). This type is essentially similar to the Francis water turbine used in most very large hydroelectric plants. The radial type has to date been used only in relatively small gas turbines and almost entirely in single-stage form. It is being increasingly used in turbosuperchargers for diesel engines. It appears to have many advantages for small turbines,

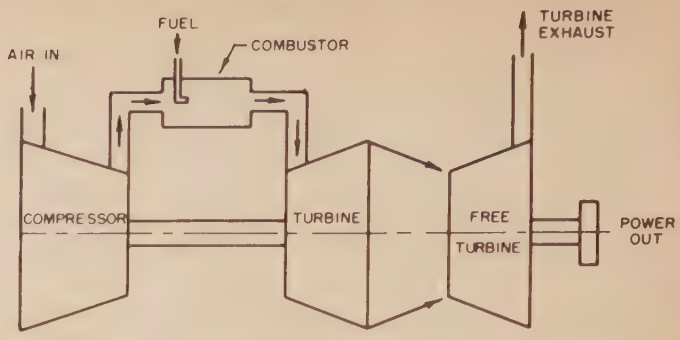

Fig. 8—Simple open-cycle gas turbine with free, or power, turbine

and it can be readily applied to multistage vehicle turbines where the shafts of the compressor and power turbines need not be coaxial.

The combustion section of a gas turbine involves the exceedingly difficult problem of burning part of the air with fuel at approximately stoichiometric (chemically correct) fuel/air ratio (hereafter F/A ratio), and then diluting the combustion products with the remainder of the air without quenching the reaction and thus wasting fuel by incomplete combustion. Since a gas turbine rarely burns more than 25% of the available oxygen, dilution without quenching, and uniform dilution so that the final mixture does not have hot streaks represents a difficult problem. Uniform dilution is not always desirable, and in some cases a core of uniformly diluted hot gas surrounded by an annulus of relatively cold gas is used to reduce temperature of the discharge end of the combustor and other parts ahead of the first stage turbine wheel. There are two types of combustor, one in which the fuel is sprayed into the air as a fine mist and burned in this form (see Fig. 7). In the other, known as the vaporizing type, the fuel is pumped or metered into a hot tube through which part of the combustion air is flowing. The hot tube and the hot air flowing through it evaporate the fuel, and a rich fuel-air mixture is discharged into the combustion zone where it is mixed with additional air and burned. Subsequent to combustion of the rich fuel-air mixture in a suitably metered air supply, the excess or dilution air is added and mixed with the combustion products.

In current applications of gas turbines to road vehicles the most advanced applications consist of a single-stage centrifugal compressor driven by an axial turbine with a second axial turbine, in series with the first one, but having no mechanical connection with it, developing the power that is supplied to the wheels. This is shown diagrammatically in Fig. 8 and represents the basic arrangement used in the Boeing, Rover, and General Motors road vehicle turbines. Fig. 9 [9] shows the actual arrangement of the General

[9] "Automotive Power—Gas Turbines or Piston Engines," by M. M. Roensch. Paper presented before American Petroleum Institute, Detroit, February, 1954.

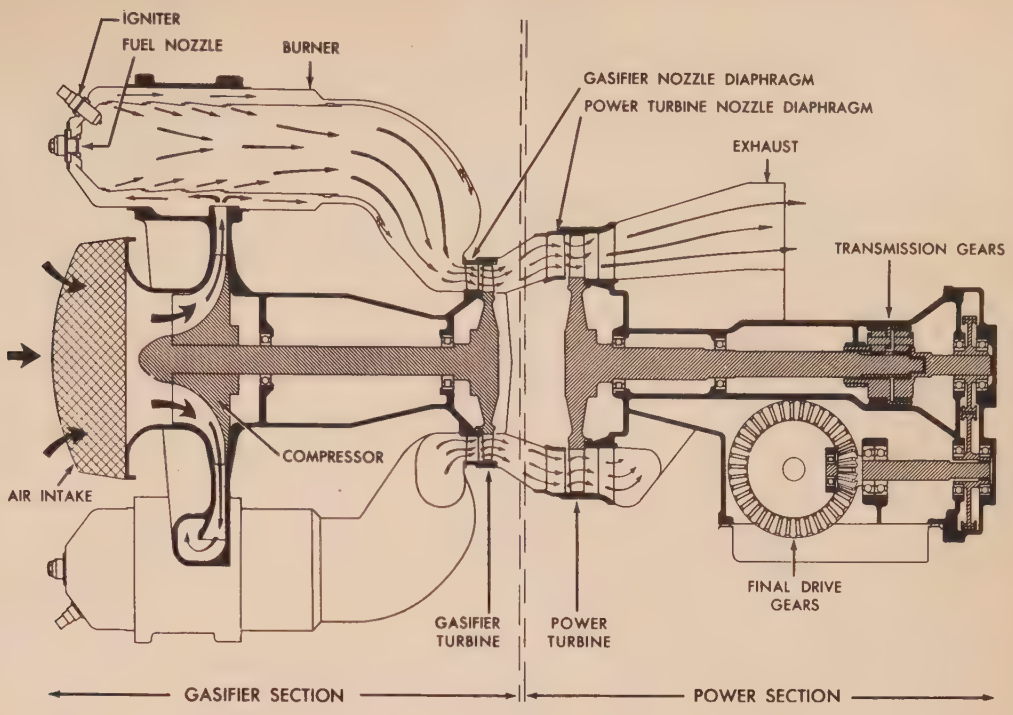

Fig. 9—General Motors experimental gas turbine for ground vehicles

Motors experimental turbine. Fig. 10 shows another variation which is identical with that shown in Fig. 9 except that a regenerator is added. The arrangement shown in Figs. 8, 9, and 10 is of the 2-shaft type usually known as a "free turbine." Fig. 10 appears to be essentially the arrangement used in the 1954 Chrysler road vehicle turbine. In the arrangement shown in Figs. 8, 9, and 10, the speed of the second turbine, the power or free turbine, bears no necessary relation to the speed of the first turbine. In fact, with the compressor turbine running at constant speed the power turbine can deliver nearly constant power output over a considerable range of rpm and thus is functioning as a torque converter. The maximum degree of torque conversion with the power turbine stalled appears to be about 2½/1 (zero output from the power turbine), but the conversion appears to be quite efficient up to a torque ratio of about 2/1 and with the output shaft down to 50% of rated rpm. Most experimental turbine installations in road vehicles use reduction and reverse gears only between the turbine and the rear wheels, thus relying entirely on the power turbine for variable torque multiplication. This arrangement may possibly be adequate for a passenger car but would be altogether insufficient for a heavy truck. The Boeing turbine in the Kenworth truck uses a 6-speed transmission between the turbine and the wheels.[10,11] In considering the waste of power by using the free turbine as a torque converter, it should be borne in mind that even under the most favorable conditions, full power at rated speed of the free turbine, the free turbine is only developing a maximum of about one-third of the total turbine power. In other words, the compressor turbine is developing twice as much power as the free turbine. Under conditions where the free turbine is inefficient due to torque conversion, the compressor turbine continues to run at approximately its most efficient speed for a given compressor output.

Before passing to detailed consideration of overall turbine performance it may be well to emphasize the fact that, even in highly efficient gas turbines operating at high turbine inlet temperature and full load,

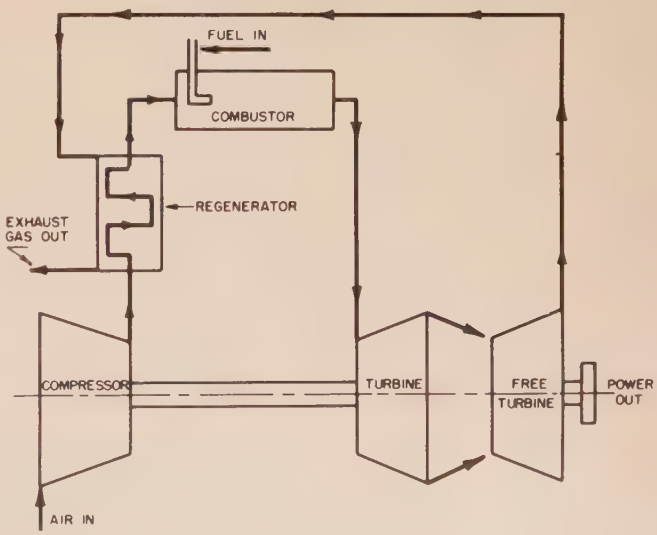

Fig. 10—Open-cycle gas turbine with free turbine and regenerator

[10] SAE *Quarterly* Transactions, Vol. 5, January, 1951, pp. 81–93: "Gas Turbine Propulsion for Ground Vehicles," by W. M. Brown.
[11] SAE *Quarterly* Transactions, Vol. 6, July, 1952, pp. 395–408: "Progress of Gas Turbine Truck Tests," by H. C. Hill.

the compressor absorbs about two-thirds or more of the gross turbine power output leaving only a maximum of about one-third for external work.

Turbine Performance—In considering and computing turbine performance, a few terms must be defined as follows:

(a) Pressure ratio (hereafter P.R.): This is the ratio of initial and final absolute pressures. Thus, in the case of compressor with a P.R. of 4 and an intake pressure of 14.7 psia (zero psi gage at sea level), the compressor discharge pressure is 58.8 psia (44.1 psi gage). In the case of a turbine with an inlet pressure of 57 psia and an outlet pressure of 15 psia the P.R. is 3.80. In gas turbine circles the term *compression ratio* is often used interchangeably with *pressure ratio* where the latter is meant. This use of *compression ratio* for gas turbines where it really applies to the *ratio of pressures* is most confusing to those who are mostly used to piston engines. In a piston engine the term compression ratio essentially, but not entirely, although it is almost completely true, defines the ratio of the density of gases in the cylinder at top dead-center to that at bottom dead-center on the compression stroke. In a gas-turbine compressor the pressure ratio does not define the ratio of the initial and final gas densities since for a given pressure ratio the density ratio will vary with compressor efficiency. Thus, at a P.R. of 4 the density ratio will be 2.70 at 100% compressor efficiency (see below) and 2.36 at 70% efficiency.

(b) Temperature: Turbine and compressor temperatures in the United States are always discussed in terms of the Fahrenheit scale, in other countries the Centigrade scale is mostly used. The British are, however, beginning to discuss turbine performance in terms of degrees Centigrade absolute.

While degrees Fahrenheit (hereafter F) is used for definition, computation is all based upon degrees F absolute. Degrees F absolute equals F plus 459.7. For the purpose of the present discussion F absolute is taken as equal to F plus 460. Degrees F absolute is generally known as degrees Rankine and will be so used hereafter as R. Where degrees Centigrade is used, the absolute scale is degrees Kelvin or K.

(c) Heat content of air: The heat content of air is known as enthalpy and in the United States is often dealt with upon a basis of Btu per lb of air above 400 R (−59.7 F). For this purpose the USN Bureau of Ships chart which is based upon the data of Keenan & Kaye [12] is most convenient and is used here.[13, 14]

For compressor calculations using the pressure ratios of interest for road vehicles, the change of enthalpy may be taken as almost directly proportional to the change of temperature. Due to increase of specific heat of air at high temperatures (turbine inlet temperatures), the temperature change does not fully define the change of energy and the enthalpy tables must be used. In gas turbine computations the temperatures are relatively low, and, consequently, the amount of fuel added to the air is so low (less than 2% for hydrocarbon fuels) that the fuel is usually neglected and the working fluid is considered to be pure air. In gas turbine computations it is possible to completely avoid mathematics and to compute the energy put into or taken out of air in compressing or expanding it over a given pressure ratio entirely by means of the change of temperature.[15] The heat content of air is defined entirely by its temperature and its weight, and it is not necessary to know either pressure or volume. It is necessary to know the pressure if it is desired to add work to the air by compressing it or to extract work from it by expansion.

(d) Compressor and turbine efficiency: Compressor and turbine efficiencies are expressed in percentages which are related to the energy which would be put into the air or taken out of it when compressing it or expanding it adiabatically. Adiabatic compression or expansion involves compressing or expanding without subtraction or addition of heat. If X Btu per lb air is required to compress it to a given pressure ratio at 100% efficiency, then at a compressor efficiency of 70% the energy required will be $\frac{X}{0.70}$ Btu per lb or 142.8% of that required at 100% efficiency. Similarly, if air is expanded through a turbine at a given pressure ratio and gives up Y Btu per lb air at 100% turbine efficiency it will give up only 0.70 Y Btu per lb at 70% turbine efficiency. Thus, compressor work is proportional to the inverse of the efficiency and turbine work directly proportional to efficiency.

Adiabatic efficiency is used throughout this discussion and is quite adequate for the purpose. Compressor authorities usually prefer the more exact "polytropic efficiency" which takes account of friction which does not appear as heat and of heat loss by convection and radiation from the compressor housing. Estimation of compressor efficiency, from the temperature rise for a given pressure ratio compared to the adiabatic temperature rise, can be misleading if the heat loss from the compressor housing is considerable as it may be.

As an example, adiabatic compression of air at a P.R. of 4 from 520 R (60 F) may be considered. This produces a temperature of 771.3 R or a temperature rise of 251.3 F. The enthalpy changes from 28.8 (above 400 R) to 89.4, an increase of 60.6 Btu per lb. If the compressor efficiency is 70%, then the enthalpy change will be $\frac{60.6}{0.70}$ or 86.6 Btu per lb, the temperature will be 877.7 R and the temperature rise will be 357.7 F. The ratio of temperature rise for 70% and 100% compressor efficiencies will be $\frac{357.7}{251.3}$

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[12] "Gas Tables," by J. H. Keenan and J. Kaye. Pub. by John Wiley, 1950. Gas Tables are based upon zero enthalpy at 0 R.

[13] "Gas Turbines and Jet Propulsion," by P. W. Gill, United States Naval Institute, Annapolis, Md. This and footnote 14 are based upon zero enthalpy at 400 R.

[14] "The Theory and Design of Gas Turbines and Jet Engines," by E. T. Vincent. Pub. by McGraw-Hill, New York, 1950.

[15] Such computations are entirely adequate for elementary discussion of this type but are probably not suitable for turbine design purposes.

or 142.2% (this is very slightly different from the ratio of enthalpy change which is 1/0.70 or 142.8%. The difference is due to slight changes in specific heat).

The ratio of initial and final temperatures for a given P.R. will be substantially a constant at any given compressor efficiency and for the purposes of this discussion will be taken to be constant as far as compressor calculations are concerned. Since the ratio of initial and final temperature is constant, the temperature rise will be proportional to the initial temperature. Thus, at a P.R. of 4 and 100% efficiency the temperature rise is 251.3 F and 520 R (60 F) compressor inlet temperature. At 400 R (−60 F) compressor inlet, the temperature rise will be $251.3 \times \frac{400}{520}$ or 193.4 F. Similarly, at 580 R (120 F) compressor inlet the temperature rise will be $251.3 \times \frac{580}{520}$ or 280.0 F. Similarly, the increase of enthalpy during compression may be taken as proportional to compressor inlet temperature. Thus, if compressing to a P.R. of 4 at 100% efficiency involves an enthalpy increase of 60.6 Btu per lb air at 520 R (60 F) compressor inlet, the enthalpy increase at 400 R (−60 F) compressor inlet will be $60.6 \times \frac{400}{520}$ or 46.6 Btu per lb air.

As an example of turbine efficiency, a turbine with a P.R. of 4 and 1650 F (2110 R) inlet temperature may be considered. If the gas expands adiabatically (100% turbine efficiency) the enthalpy will decrease from 440.2 to 270.7, a drop of 169.5 Btu per lb and the temperature drop will be 622.2 F. If the turbine efficiency is 70%, the enthalpy drop will be 118.7 Btu per lb and the temperature drop will be 421.6 F. The ratio of initial and final temperatures is not significant since in this temperature range the specific heat of air varies considerably with temperature (but does not do so significantly at compressor inlet and discharge temperatures).

(e) Computation of net turbine work and thermal efficiency: As an example of computation of net turbine work and thermal efficiency the values for compressor and turbine work at efficiencies of 70% in (d) above may be used. This is purely an example of procedure since a turbine unit with efficiencies of only 70% for both compressor and turbine represents very poor practice as will be seen below. To the data in (d) above is added a computation for a turbine inlet temperature of 1050 F (1510 R). The data are shown in Table 1, and it will be noted that the compressor work is 86.6 Btu per lb air, and that the turbine at 1050 F inlet temperature only puts out 84.0 Btu per lb air, thus the turbine does not put out enough work at this inlet temperature to drive the compressor. At 1650 F turbine inlet the turbine puts out 118.7 Btu per lb air which is 32.1 Btu per lb in excess of the compressor power.

Further computation shows that an inlet temperature of about 1090 F is needed to make turbine work equal compressor work. From Table 1 it will be noted that 161.2 and 324.8 Btu per lb of air need to be added to the air at compressor discharge temperature to bring it to turbine inlet temperatures of 1050 F and 1650 F respectively. It will also be noted that at a turbine inlet temperature of 1050 F the total enthalpy of the turbine exhaust is 77.2 Btu per lb higher than that at the compressor discharge. Similarly, at 1650 F turbine inlet the enthalpy of the turbine exhaust is 206.1 Btu per lb higher than at the compressor discharge. These figures will be discussed further below, but it may be noted that $\frac{77.2}{161.2} =$ 47.9% of the heat that has to be added to the air between the compressor outlet and the turbine inlet at an inlet temperature of 1050 F. Similarly, at 1650 F inlet temperature $\frac{206.1}{324.8} = 63.5\%$ of the heat requirement.

Thus, if the excess heat in the turbine exhaust could be transferred to the compressor discharge, the heat required to be added as fuel would be reduced by about half at 1050 F inlet temperature and by about two-thirds at 1650 F inlet temperature. At 1050 F turbine inlet temperature the thermal efficiency is negative. At 1650 F inlet temperature, the heat added as fuel (without regenerator) is 324.8 Btu per lb air and the net output at the turbine shaft is 321.1 Btu per lb air. The thermal efficiency is $\frac{32.1}{324.8} = 9.9\%$, when based upon the assumption that the fuel combustion efficiency is 100%.

In Table 1 horsepower absorbed by the compressor

Table 1—Compressor and Turbine Performance for a Pressure Ratio of 4, Turbine and Compressor Efficiencies 70%, and Turbine Inlet Temperatures of 1050 F and 1650 F

State of Gas in Unit	Temperature, R	Temperature, F	Total Enthalpy, Btu/lb Air	Change in Enthalpy, Btu/lb Air	Change in Temperature, F	Enthalpy Turbine Exhaust Minus Compressor Outlet	Temperature Difference Turbine Exhaust and Compressor Outlet, F	Hp-Lb Air Per Sec: Put into Compressor	Hp-Lb Air Per Sec: Put out by Turbine	Net Turbine Hp per Lb Air per Sec
Compressor Inlet	520	60	28.8							
Compressor Outlet	877.7	417.7	115.4	+86.6	+ 357.7			122.5		
Turbine Inlet	1510	1050	276.6	+161.2	+ 632.3				118.8	− 3.7
	2110	1650	440.2	+324.8	+1232.3				168.0	+45.5
Turbine Exhaust	1186.7	726.7	192.6	−84.0	− 323.3	77.2	309			
	1678.4	1218.4	321.5	−118.7	− 431.6	206.1	800.7			

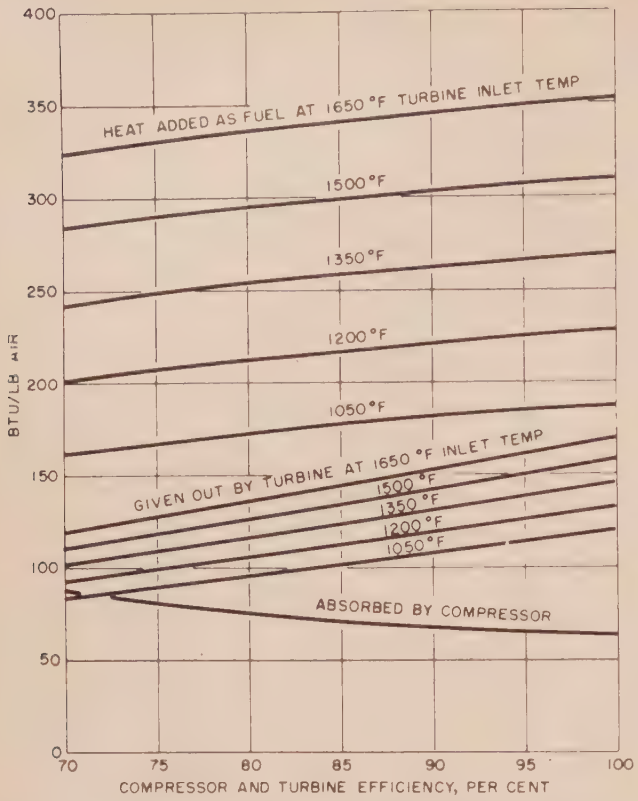

Fig. 11—Heat balance of turbine unit with pressure ratio of 4/1 and compressor intake temperature of 60 F (520 R)

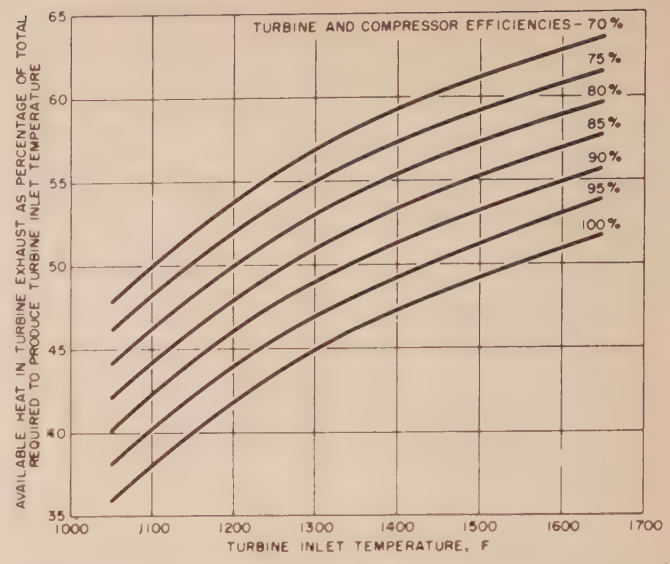

Fig. 12—Heat balance of turbine unit with pressure ratio of 4/1 and compressor intake temperature of 60 F (520 R)

and put out by the turbine is given per lb of air per sec. Gas turbine calculations are usually based upon airflow in lb per sec. A flow of 1 lb air per sec corresponds to 3600 lb per hr and 2545 Btu equals 1 hp-hr. Therefore, hp per lb air per sec will equal

$$\frac{\text{enthalpy change (gain or loss) per lb air per sec} \times 3600}{2545}$$

= 1.415 enthalpy change. Thus, the enthalpy change in compression is 86.6 Btu per lb or 122.5 hp per lb per sec. A rough rule for compressor power [16] is hp per lb air per sec = one-third temperature rise F across compressor. A more exact estimate for a specific heat of 0.24 for air at constant presure is given by $\frac{\text{temp. rise F} \times 0.24 \times 778}{550} = 0.34$ temperature rise F. This last expression will be noted to give a value for the Table 1 data which is within less than 1% of the power value derived from enthalpy. The data in Table 1 are computed upon an often-used basis of simplifying assumptions which are:

First, the pressure ratios of the compressor and the turbine are equal. This will not be true in practice since there will be a slight pressure drop between the compressor and the turbine. Also, there may be back pressure upon the turbine exhaust which will further reduce the pressure ratio across the turbine wheel. Secondly, the mass flow through the turbine is assumed to be equal to that through the compressor. Due to the addition of fuel the mass flow through the turbine at high inlet temperatures is nearly 2% higher than through the compressor. With highly efficient regeneration the percentage increase in mass flow will be much less than 2% as shown below. As a result, the turbine power (at the calculated P.R.) will be up as much as 2% higher than the computed values.

Using the methods discussed above, the heat balance of a turbine unit with a pressure ratio of 4, compressor inlet temperature of 60 F, compressor and turbine efficiencies of 70 to 100%, and turbine inlet temperatures of 1050 to 1650 F is shown in Fig. 11.

Fig. 11 is said to show the heat balance of a turbine unit, but the statement is only even partially true under strictly limited conditions. The statement will be partially true for a turbine unit in which all compressors and turbines run at fixed speed such as may apply in driving a synchronous generator with a single-shaft turbine of the type shown in Fig. 1. In a vehicle turbine such as shown in Fig. 9, it is not possible to vary turbine inlet temperature without simultaneously changing the pressure ratio. In any given turbine of this type the pressure ratio will increase as the turbine inlet temperature increases. In an aircraft turbine of the single-shaft propeller type, it is possible, with a controllable pitch propeller, to operate at constant pressure ratio (constant rpm) with variable turbine inlet temperature but this procedure is wasteful of fuel. Therefore, Fig. 11 and subsequent illustrations showing variable turbine inlet temperature really apply to families of turbines, each turbine being designed for a given pressure ratio at each specified turbine inlet temperature.

[16] First pointed out to the writer in 1951 by Dr. S. Hooker, Bristol Aeroplane Co., Ltd.

Fig. 11 shows that when the compressor and turbine efficiencies are both 70% and the turbine inlet temperature is 1050 F, the compressor takes more power than the turbine puts out. Fig. 11 shows that increasing either turbine or compressor efficiency, or efficiency of both, at a turbine inlet temperature of 1050 F will make the turbine power equal or exceed the compressor power. Fig. 11 in general shows the great importance of compressor and turbine efficiency; it also indicates the very important advantage resulting from increasing turbine inlet temperature. This figure illustrates why some of the early gas turbines, with low efficiency compressors and turbines and using materials with low hot strength, did not develop enough power to overcome their internal losses. In the early gas turbines, compressors with efficiencies well below 70% were probably used, since it is only in the last 20 years that the centrifugal compressors with pressure ratios of 2½/1 or higher per stage and efficiencies of over 70% have been produced.

Fig. 11 shows that if turbine efficiency increases at a given turbine inlet temperature, net power, which is gross turbine power minus compressor power, increases without more fuel being required. If, at a given turbine efficiency and inlet temperature, compressor efficiency is increased, then more net power is obtained but slightly more fuel is required. Thus, at any given turbine inlet temperature, the compressor efficiency determines the heat to be added as fuel.

Fig. 12 may be said to complete the heat balance

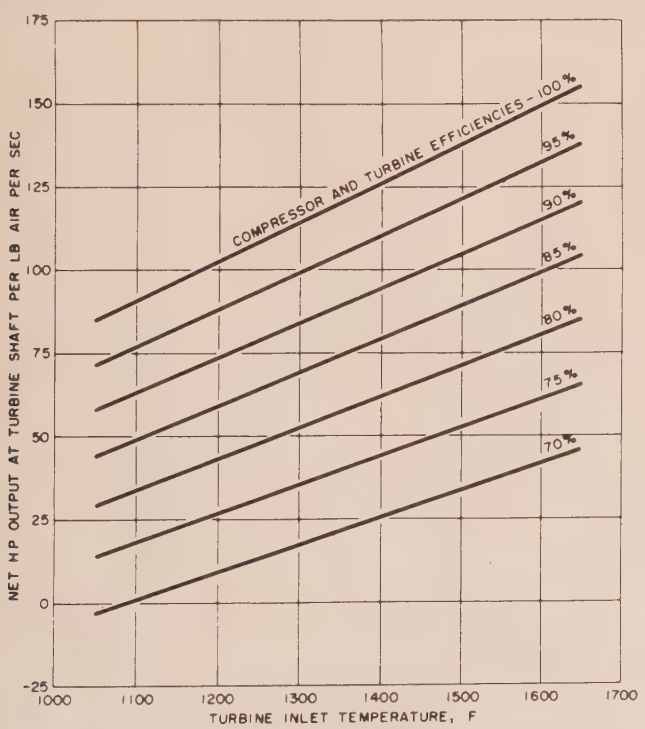

Fig. 13—Effects of compressor and turbine efficiencies and turbine inlet temperature upon net turbine power output at pressure ratio of 4/1 and compressor inlet temperature of 60 F

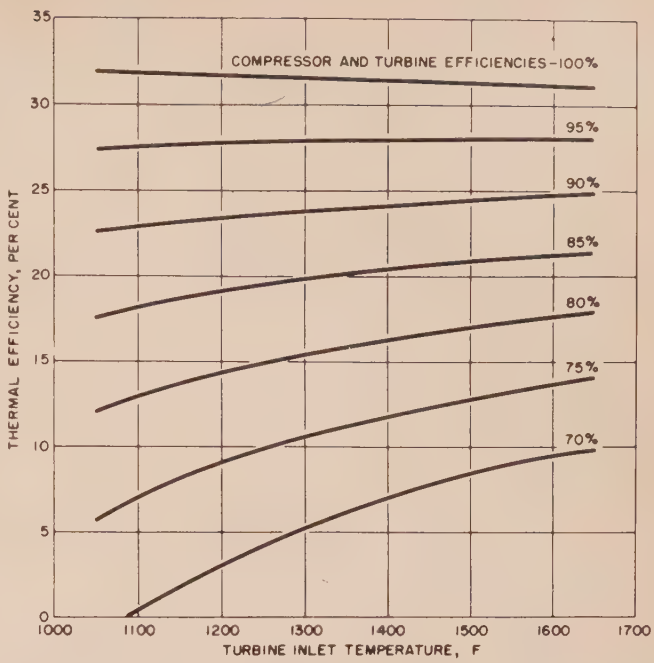

Fig. 14—Effects of compressor and turbine efficiencies and turbine inlet temperature upon thermal efficiency when pressure ratio is 4/1 and compressor inlet temperature is 60 F

of this turbine unit, since it shows the heat in the turbine exhaust which *might* be transferred by a regenerator to the air discharged by the compressor, prior to burning fuel in said air. The data in Fig. 12 are expressed as percentages of the total heat required to raise the compressed air from the compressor discharge temperature to the turbine inlet temperature. In actual quantity (rather than percentage) the heat shown in Fig. 12 is equal to the heat added as fuel (at 100% combustion efficiency) minus the heat drop through the turbine. The data shown in Fig. 12 will be discussed later under the subject of heat exchangers or regenerators.

The data in Fig. 11 tell almost the whole story for this particular pressure ratio and compressor inlet temperature. However, the figure does not present the story conveniently and the data are therefore recomputed upon a basis of hp per lb of air per sec as shown in Fig. 13. This figure, of necessity, presents data at equal compressor and turbine efficiencies and shows net horsepower versus turbine inlet temperature for turbine and compressor efficiencies of 70 to 100% by 5% steps. This figure cannot be used where compressor and turbine efficiencies are not equal. However, if turbine efficiency is 85% and compressor efficiency 75%, the net output will be fairly close to the value shown for equal turbine and compressor efficiencies of 80%. The straight-line relationships which result from plotting on a basis of turbine inlet temperature are very useful in respect to catching errors in the very tedious arithmetic which is involved.

Fig. 14 shows the thermal efficiency (100% fuel

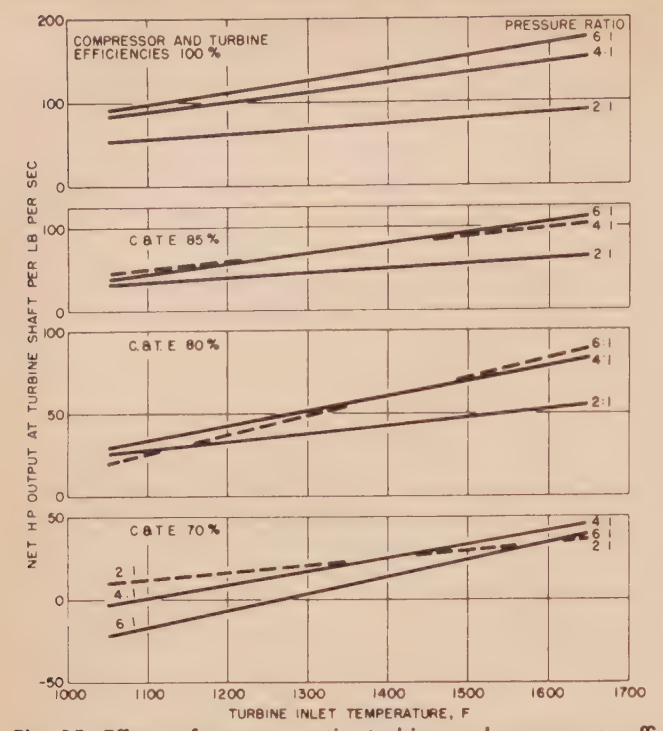

Fig. 15—Effects of pressure ratio, turbine and compressor efficiencies, and turbine inlet temperature upon net turbine power output when compressor inlet temperature is 60 F

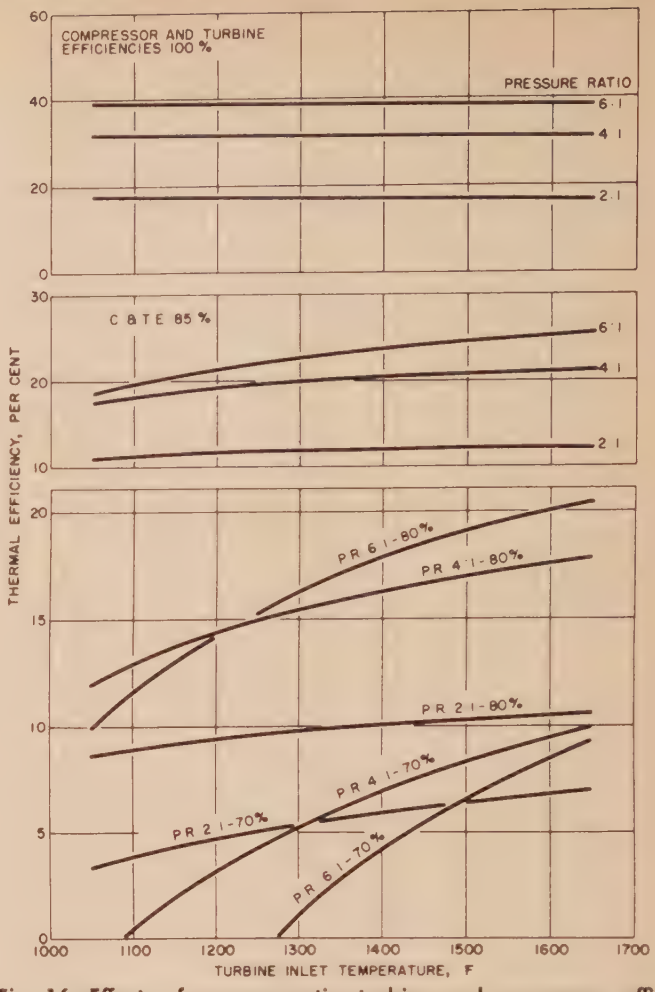

Fig. 16—Effects of pressure ratio, turbine and compressor efficiencies, and turbine inlet temperature upon thermal efficiency where there is no regeneration at compressor inlet temperature of 60 F

combustion efficiency) corresponding to Fig. 13. Analyses similar to those shown in Figs. 13 and 14 but for pressure ratios of 6/1 and 2/1 show that, at high compressor and turbine efficiencies and particularly at high turbine inlet temperatures, 6/1 P.R. is equal to or better than a P.R. of 4/1. At low compressor and turbine efficiencies, and particularly so at low turbine temperatures, 2/1 P.R. is more efficient than a P.R. 4/1.

Figs. 15 and 16 summarize the joint effects of turbine and compressor efficiency, turbine inlet temperature and pressure ratio upon specific power output, and thermal efficiency for pressure ratios of 2/1, 4/1, and 6/1. Figs. 17 and 18 show specific power output and thermal efficiency at low turbine inlet temperatures for pressure ratios of 1.2/1, 1.5/1, and 2/1. The data in Figs. 17 and 18 are included in view of their possible significance at very low load factors. These data are computed only for turbine and compressor efficiencies of 80%.

(f) Effects of regeneration upon thermal efficiency: The author has chosen to use the term "regenerator" for any type of heat exchanger which transfers heat from the turbine exhaust to the air discharged by the compressor. "Recouperator," "regenerator," and "heat exchanger" are commonly used but regenerator has been chosen (possibly incorrectly) in the interest of brevity. The author does not like the term "heat exchanger," since heat exchangers are used in gas turbine units for a variety of purposes other than transfer of heat from the turbine exhaust to the air at the compressor discharge.

Theoretically, regeneration has no effect upon net turbine output. Practically, regeneration will slightly reduce output for a given mass airflow and compressor pressure ratio, since there will be a pressure drop through the regenerator which will reduce the pressure ratio and the gross output of the turbine, but this effect will be neglected here. Also in the case of a rotary regenerator there is *likely* to be leakage of compressed air to the turbine exhaust; also some types of rotary regenerator not only leak but actually transfer air from the compressor discharge to the turbine exhaust. These two effects reduce the mass airflow to the turbine, but are neglected here as is the pressure drop through the regenerator.

The possible effects of regeneration upon thermal efficiency are very large. The effect of regeneration upon thermal efficiency depends upon: pressure ratio, turbine and compressor efficiency, turbine inlet temperature, and per cent regeneration. Per cent regeneration is the percentage of the *available* heat in the turbine exhaust which is actually transferred to the compressor outlet air. Available heat is, of course,

enthalpy of the turbine exhaust minus enthalpy of the compressor discharge air. These variables make the overall effect so complex that it cannot be dealt with here and only a simple example will be presented. It was pointed out many years ago, first by Stodola [2] as far as the author's knowledge goes, that at 100% regeneration the maximum thermal efficiency would be theoretically obtained at a pressure ratio of 1. It is sufficient for the present to point out that effects of regeneration in general are greatest at the lower pressure ratios. It is obvious that the effect of regeneration will be zero at such a high pressure ratio and low turbine inlet temperature that the temperature at compressor discharge is equal to that of the turbine exhaust.

Neglecting the complexities introduced by the numerous variables, it can be categorically stated that the effects of regeneration upon thermal efficiency are so pronounced that it is the obvious step to take. The difficulties involved in accomplishing the obvious will be avoided here and discussed at a later stage.

Fig. 19 shows the effects of regeneration from zero to 100% upon thermal efficiencies at pressure ratios of 2/1 and 4/1. Fig. 19 is computed for the reasonable compressor and turbine efficiencies of 80% and for a compressor inlet temperature of 60 F.

Figs. 20 and 21 show the effects of regeneration for pressure ratios of 2/1, 1.5/1, and 1.2/1, at turbine inlet temperatures of 600 to 1200 F, at turbine and compressor efficiencies of 80% with a compressor inlet of 60 F. Fig. 20, which shows the effects of 100% and 90% regeneration, confirms Stodola in that at 100% regeneration the thermal efficiency is highest at the lowest pressure ratio for all turbine inlet temperatures shown. It is of interest that, when the regeneration is reduced to 90%, the most efficient

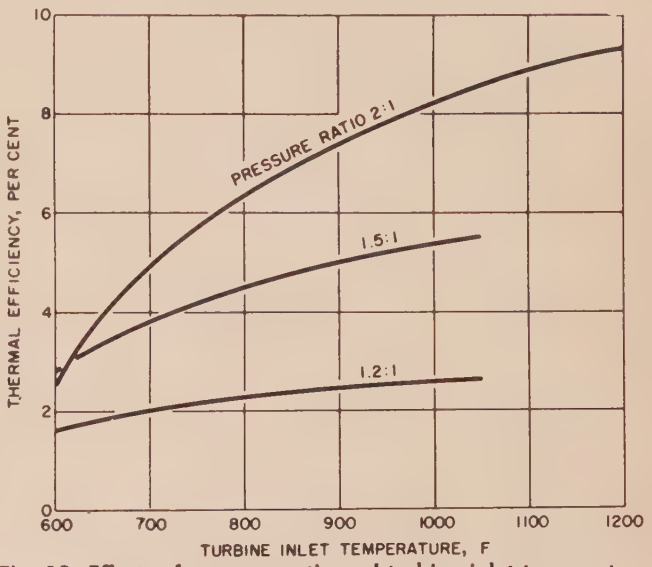

Fig. 18—Effects of pressure ratio and turbine inlet temperature upon thermal efficiency. Zero regeneration, turbine and compressor efficiencies of 80%, and compressor inlet temperature of 60 F

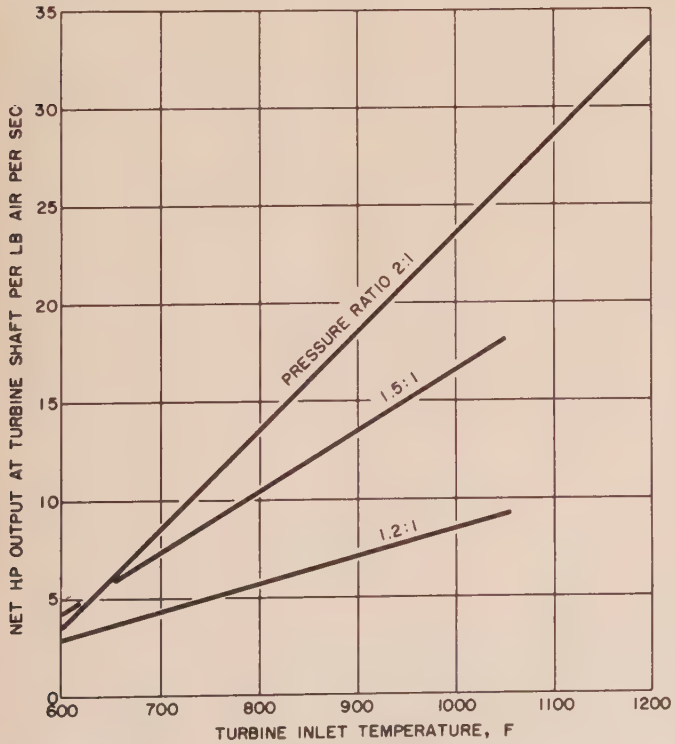

Fig. 17—Effects of pressure ratio and turbine inlet temperature upon net turbine power output. Turbine and compressor efficiencies are 80%, compressor inlet temperature is 60 F

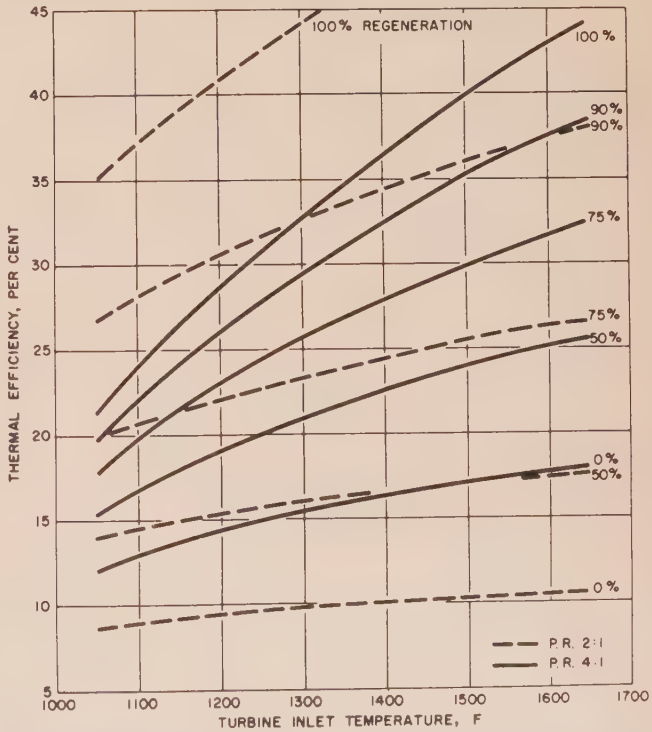

Fig. 19—Effects of regeneration and turbine inlet temperature upon thermal efficiency at pressure ratios of 2/1 and 4/1. Compressor and turbine efficiencies of 80% and compressor inlet temperature of 60 F

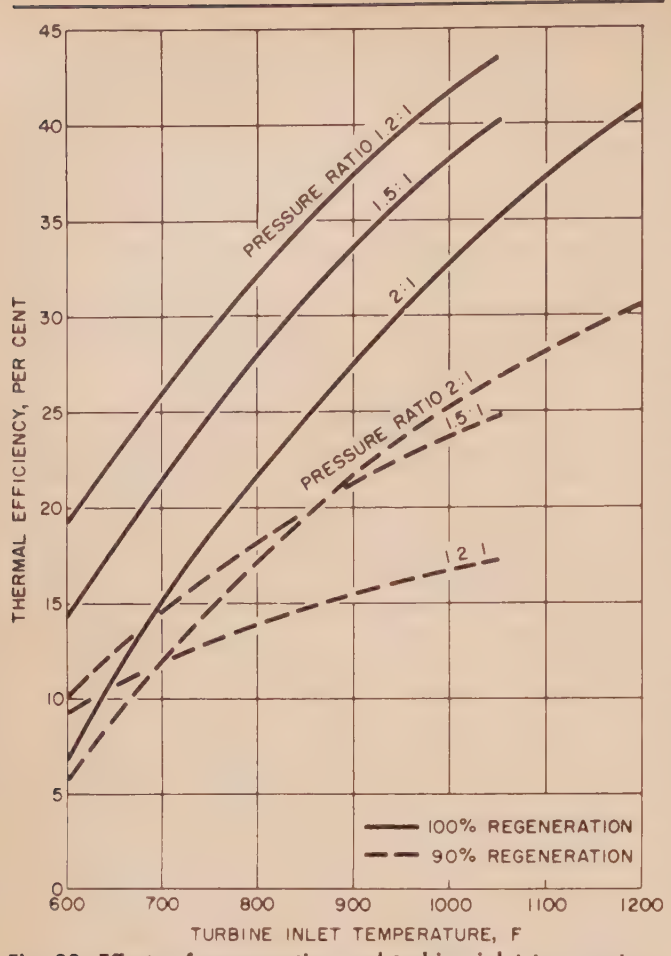

Fig. 20—Effects of regeneration and turbine inlet temperature upon thermal efficiency. Compressor and turbine efficiencies of 80% and compressor inlet temperature of 60 F

pressure ratio is no longer the lowest and depends upon turbine inlet temperature. Fig. 21 shows effects of 50% and 75% regeneration and indicates that, at these amounts of regeneration, a 2/1 pressure ratio is in general the most effective. Figs. 20 and 21 may be compared to Fig. 18 which deals with the same pressure ratios and turbine inlet temperatures but with zero regeneration. Figs. 18, 20, and 21 show in general that regeneration of more than 50% is desirable, although even 50% may double the thermal efficiency. It is noticeable that at 90% regeneration a thermal efficiency of about 10% (about 1.4 lb per hp-hr specific fuel consumption [17]) is obtained at 600 F turbine inlet with pressure ratios of 1.2/1 and 1.5/1.

The author is well aware that discussion of turbine inlet temperatures as low as 600 F may currently seem academic. He is convinced that temperatures of this order will be necessary, if reasonable fuel economy is to be obtained at the very low load factors which will prevail much of the time in some road vehicle applications. Discussion of 90% regeneration at low pressure ratios and low turbine inlet temperatures may also seem academic, but the grapevine indicates that such an efficiency is in the cards even though the efficiency may be much lower at the maximum inlet temperature.

Figs. 22 and 23 further illustrate the advantages of regeneration. In these figures the pressure ratio is 4/1, compressor and turbine efficiencies are 80%, compressor inlet temperature is 60 F, and the turbine inlet temperature is constant at 1500 F and may be considered to represent the "full throttle" condition. At "full throttle" a regenerator efficiency of 75% appears to be not improbable for a road vehicle turbine. Fig. 23 shows that at 75% regenerator efficiency only about 13% of the air will be burnt, thus the actual combustion products (carbon oxides, water, and sulfur oxides) will be at least well diluted even if of excessive volume. In this connection it may be mentioned that if an idle condition of 600 F turbine inlet with 90% regenerator efficiency is assumed, then only about 2% of the air will be burnt and the combustion products will be still further diluted.

The data in Figs. 19 to 23 inclusive, of course, overevaluate the effects of regeneration since they take no account of the losses due to the regenerator which have been briefly mentioned. Presentation of regenerator effects in terms of thermal efficiency is, however, misleading in one respect, since the regenerator can have a marked effect upon fuel consumption at idle where the thermal efficiency is, of course, zero. A rough computation for idling at 600 F turbine inlet with 90% regenerator efficiency shows that the idle fuel consumption will be reduced by about 80% by the regenerator.

An efficient regenerator very largely recovers the losses which occur in the turbine wheel or wheels due to the fact that the energy conversion is less efficient than adiabatic expansion. The academic case of 100% regenerator efficiency may be considered, since in this case all the energy supplied by the fuel is recovered (at 100% fuel combustion efficiency) by the turbine wheel or wheels. In this last case the only energy lost in the cycle is that necessary to compress the air to the compressor discharge pressure. On first thought it might appear that since all the fuel energy is recovered, the turbine efficiency does not matter. This is not true, however, since the power output of the turbine wheel or wheels is proportional to efficiency. Thus, if the turbine efficiency is low, in the case of a 2-stage turbine unit, both stages will be required to be more powerful to produce a given "full throttle" power output. This increased power will require a larger compressor which will absorb

[17] Throughout this discussion fuel efficiency has been expressed in terms of thermal efficiency. The author and most other piston engine engineers are used to evaluating engine efficiency in terms of specific fuel consumption. Thermal efficiency does not produce an immediate impression of relative efficiency. However, expression of performance in terms of specific fuel consumption at zero thermal efficiency has its obvious limitations. Furthermore, *relative* fuel mileage is directly proportional to thermal efficiency, and fuel mileage is the accustomed mode of expression for road performance. For those who wish to consider turbine fuel efficiency in terms of specific fuel consumption there is the very simple conversion factor:

$$\text{Specific fuel, lb bhp-hr} = \frac{13.75}{\text{per cent thermal efficiency}}$$

This factor is correct for a fuel having a lower heating value of 18,500 Btu per lb. Such a heating value is probable for road vehicle turbine fuels.

more power and the compressor power is a total loss in the cycle.

An efficient regenerator has another interesting effect when a free or power turbine, as shown in Fig. 10, is used for torque conversion to such a degree that the efficiency of the turbine wheel falls off markedly. Under these conditions of torque conversion the available power will also fall off markedly in proportion to turbine efficiency. While the power will fall off sharply, the specific fuel consumption will hardly be affected, since most of the wasted energy will be recovered in the regenerator. This is in marked contrast to a hydraulic torque converter where conversion losses reduce power output, increase specific fuel consumption, and often require a heat exchanger to get rid of the energy losses.

A regenerator will, of course, store heat, and this may require special control measures in a vehicle turbine which is subject to rapid load changes. Thus, if a turbine unit has been idling at 600 F turbine inlet and is accelerated to a steady load condition of 1500

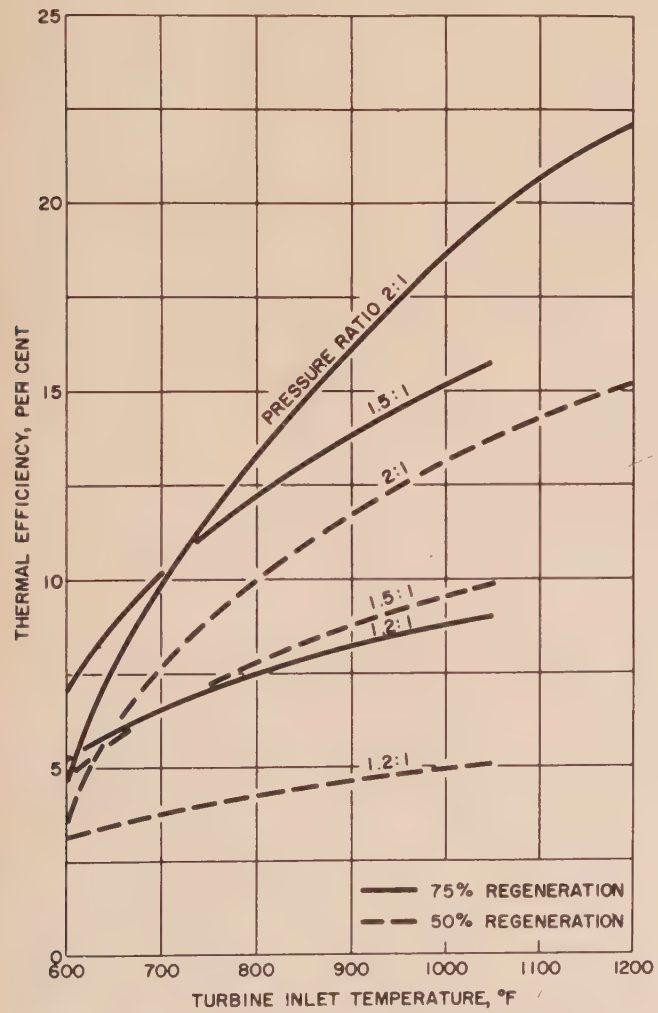

Fig. 21—Effects of regeneration and turbine inlet temperature upon thermal efficiency. Compressor and turbine efficiencies of 80% and compressor inlet temperature of 60 F

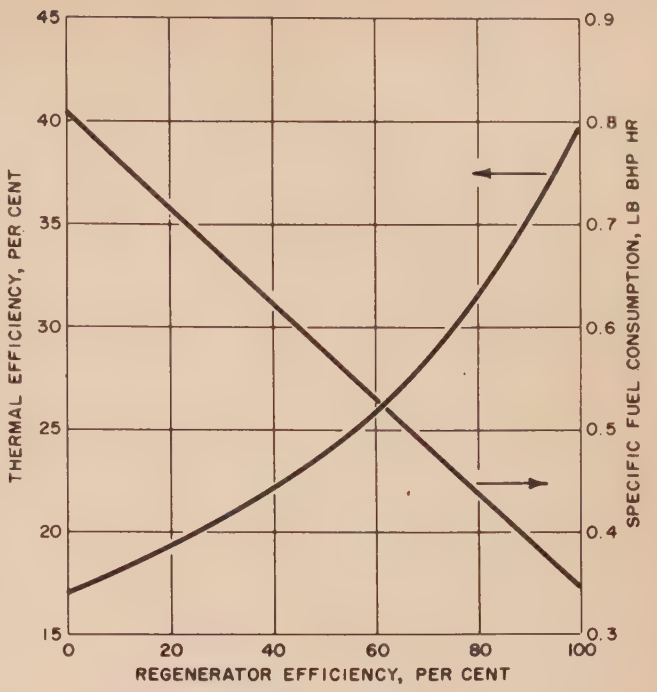

Fig. 22—Effects of regeneration upon thermal efficiency and specific fuel consumption. Pressure ratio 4/1, turbine inlet temperature of 1500 F, compressor inlet temperature of 60 F, compressor and turbine efficiencies of 80%, fuel 18,500 Btu lb lower heating value

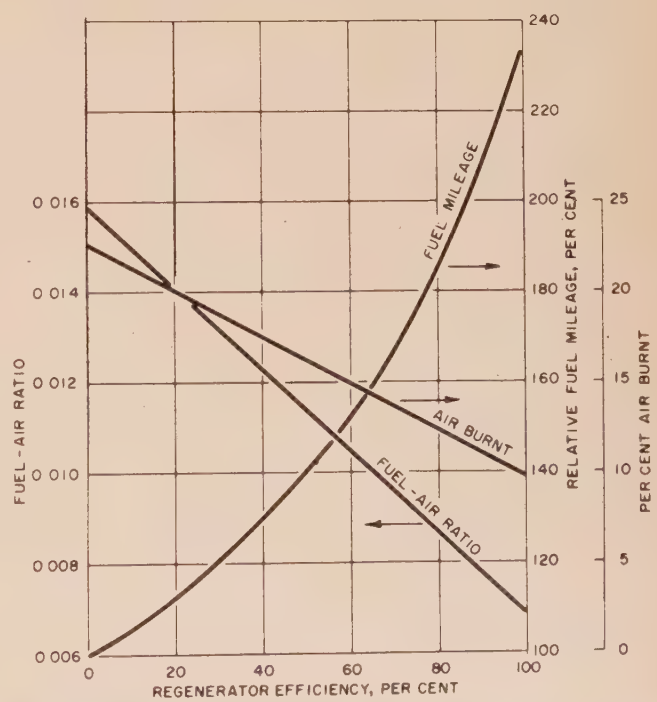

Fig. 23—Effects of regeneration upon relative fuel mileage, fuel/air ratio, and per cent air burnt. Pressure ratio 4/1, turbine inlet temperature 1500 F, compressor inlet temperature of 60 F, compressor and turbine efficiencies of 80%, fuel 18,500 Btu lb lower heating value, and stoichiometric fuel/air ratio assumed 0.070

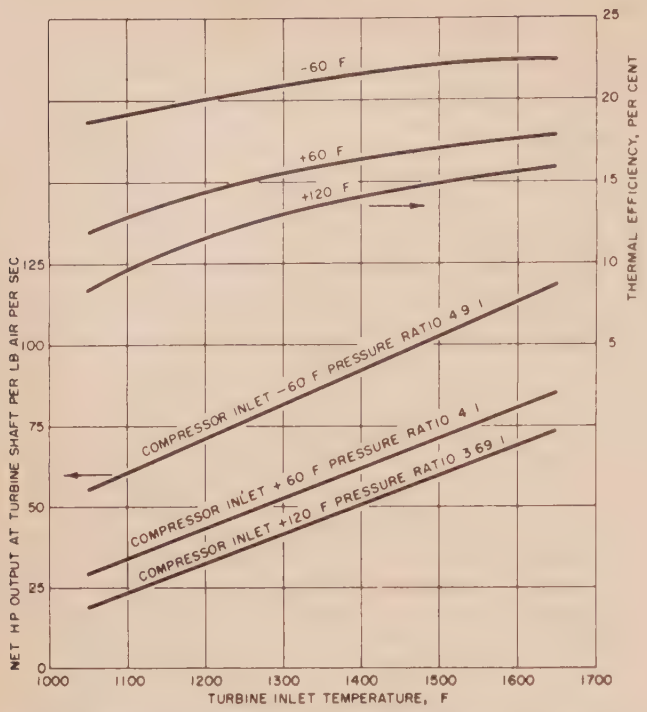

Fig. 24—Effects of compressor inlet temperature upon net power output per lb of air per sec and upon thermal efficiency at varying turbine inlet temperatures. Pressure ratio of 4/1 at 60 F compressor inlet temperature, compressor and turbine efficiencies of 80%

F turbine inlet, the regenerator will be cold initially and will go on storing heat until it reaches equilibrium, with the exhaust temperature corresponding to 1500 F turbine inlet. This means that the fuel/air ratio will have to progressively fall until equilibrium is reached. The reverse case of decelerating from 1500 F turbine inlet to the idle condition represents a disadvantage of the regenerator in vehicle use. The stored heat in the regenerator will probably result in the output shaft delivering significant unwanted power even if the fuel is cut off. Cases are on record of land-based turbines running for a considerable time on the stored heat after the fuel was cut off. With the free turbine type unit shown in Fig. 10 it appears, however, that relatively simple control measures can avoid this unwanted power and such control is discussed below.

The foregoing discussion of regeneration makes clear its phenomenal advantages and illustrates the reasons for the current intense design and development activity on this subject. In view of the advantages of regeneration the question may well be asked why the principle is not applied to aircraft turbines. A few serious efforts with turbo-prop engines have been made, but the necessary bulk and weight involved at the time of these efforts resulted in an unacceptable engine type. For high-speed, high-altitude aircraft, the high pressure ratio (10/1 to 12/1) engine appears to be a much better answer than the regenerative turbine, but the latter may ultimately find a place in helicopters and other low-speed, low-altitude aircraft.

(g) Effects of atmospheric temperature upon net output and thermal efficiency: The effects of atmospheric temperature upon the performance of a gas turbine are very pronounced both in respect to power output and to thermal efficiency. These effects are much more pronounced than those occurring in piston engines. In a gasoline engine, power output will sensibly vary inversely as the square root of the ratio of the absolute atmospheric air temperatures. Thus, increasing the intake air temperature from +60 to +120 F will only drop power output by about 5%, provided that the carburetion is adequately compensated and that knocking does not occur. In a gasoline engine with suitable carburetor compensation, variation of intake air temperature will have no significant effect upon thermal efficiency. In the diesel engine, data upon the effects of atmospheric air temperature upon output are largely lacking, but if one can judge by the effects of atmospheric density they should be much less than those of the gasoline engine. Provided that combustion efficiency is not affected, variation of intake air temperature should be sensibly without effect upon thermal efficiency. In the piston engine the effects of atmospheric air temperature are, in naturally aspirated engines at least, almost entirely due to variation in the weight of air taken in.

In the gas turbine the effects of atmospheric tem-

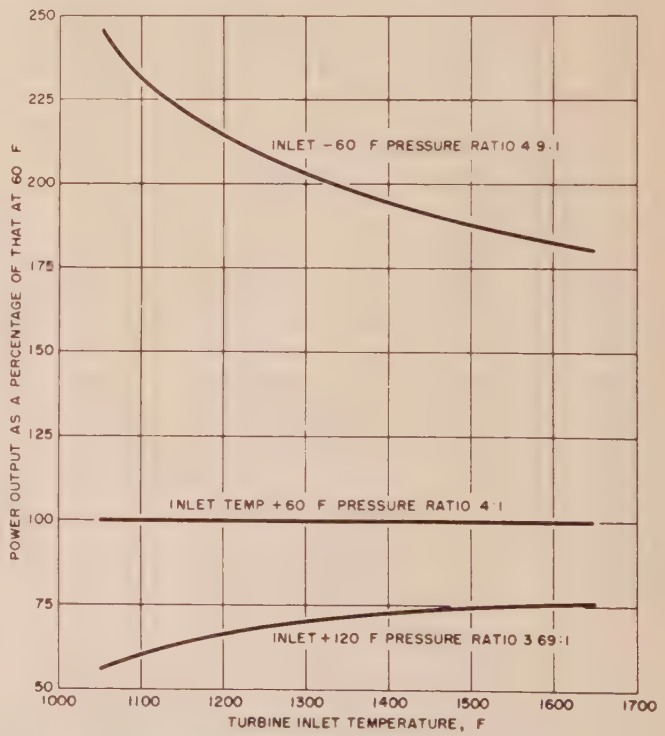

Fig. 25—Effect of compressor inlet temperature upon percentage power output at varying turbine inlet temperature. Pressure ratio 4/1 at 60 F inlet temperature, compressor and turbine efficiencies of 80%, single-shaft turbine with constant compressor speed, and volumetric efficiency of compressor independent of inlet temperature

perature are much more complex and much larger. First, increase of atmospheric temperature reduces the weight of air taken in by the compressor, and, secondly, it reduces the pressure ratio. Conversely, reduction of atmospheric temperature increases both the weight of air induced and the pressure ratio. Secondly, as the compressor intake temperature is increased, the compressor outlet temperature is also increased, and less fuel energy can be added to the air to produce a given turbine inlet temperature. The result of these effects is that, as atmospheric temperature increases, the gross power of the turbine wheel or wheels decreases due to a lower weight of gas and due to a lower expansion ratio (lower pressure ratio). While the work of compression decreases due to reduced pressure ratio, it becomes a greater proportion of the gross turbine wheel work. Gross turbine work minus compressor work is *net work*. The ratio: $\frac{\text{net work}}{\text{gross work}}$ is commonly referred to as "the work ratio" in gas turbine parlance.[18] As an example of "work ratio," a unit with a pressure ratio of 4/1 at 60 F compressor inlet temperature, compressor and turbine inlet efficiencies of 80%, and a turbine inlet temperature of 1500 F may be considered. Turbine and compressor speed is considered to be constant with varying atmospheric temperature, that is, a single-shaft unit such as Fig. 1 is assumed, and the comparison does not apply to a unit with a free turbine such as shown in Fig. 10. *Pressure rise* across the compressor (pressure ratio minus 1) is assumed to be proportional to air density at the compressor inlet. For atmospheric temperatures of −60 F, +60F, and +120 F the pressure ratios are 4.9/1, 4.0/1 and 3.69/1. At +60 F the compressor absorbs 60.2% of the gross turbine work and the work ratio is 39.8%. At +120 F the compressor absorbs 65.7% of the gross turbine work and the work ratio is 34.3%. At −60 F the compressor absorbs 48.9% of the gross work and the work ratio is 51.1%. The thermal efficiencies at atmospheric temperatures of −60 F, +60 F and +120 F are 21.9%, 17.0% and 14.8% respectively (see Fig. 24). The mass airflow is assumed to be proportional to air density at the compressor inlet, and the relative net power outputs at compressor inlet temperatures of −60 F, +60 F, and +120 F are 188%, 100%, and 73.7% respectively. It is of some interest to note that in this particular case the thermal efficiency is almost directly proportional to the work ratio. This relation is not generally applicable, although, in general, increasing work ratio will improve thermal efficiency at any given pressure ratio and turbine inlet temperature. At given pressure ratio, compressor and turbine efficiencies, increasing turbine inlet temperature increases both work ratio and thermal efficiency. However, adding a regenerator does not change work ratio (neglecting the mass flow and pressure losses discussed above) and materially improves thermal efficiency. Nonregenerative turbines with very high pressure ratios will have low work ratios and very high thermal efficiency at high turbine inlet temperatures. (This will only be true if high compressor and turbine efficiencies are obtained.)

The effects of atmospheric temperature discussed above are shown in Fig. 24 for turbine inlet temperatures of 1050 to 1650 F. In Fig. 24 the output data are given in terms of power per lb air per sec, that is, the effect of inlet temperature upon mass airflow is not included. In Fig. 25 the effects of temperature upon mass airflow are included, and in this case relative power is based upon the power available at any given turbine inlet temperature for a compressor inlet temperature of +60 F. The data in Fig. 25 are presented in another and possibly more convenient form in Fig. 26. There are insufficient points to really define the shape of the curves in Fig. 26, but the curves are sufficiently sound to show the pronounced trends. It will be noted that both the gain and loss of power due to decrease and increase of compressor inlet temperature are the more pronounced as turbine inlet temperature is reduced. Fig. 25 is sufficient to show,

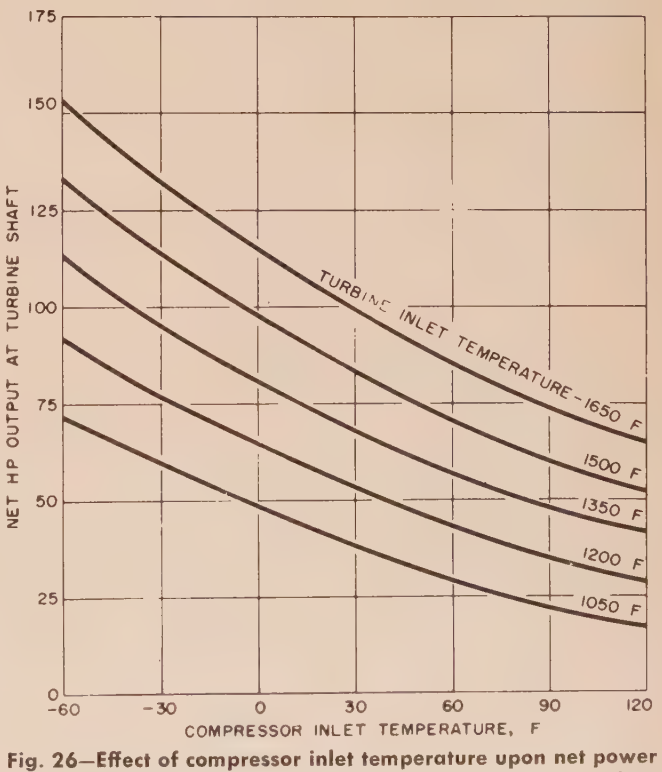

Fig. 26—Effect of compressor inlet temperature upon net power output at turbine shaft for a turbine unit having an airflow of 1 lb per sec at 60 F compressor inlet temperature. Pressure ratio 4/1 at 60 F inlet temperature, compressor and turbine efficiencies of 80%, single-shaft turbine with constant compressor speed, and volumetric efficiency of compressor independent of inlet temperature

[18] It is possibly of interest to consider the work ratio of piston engines. In a gasoline engine of 8/1 compression ratio at road load, the work ratio, on a brake power basis, may be as low as 35%. The work ratio of piston engines is, of course, mainly of academic interest. It is, however, worth discussion since many piston engine engineers jeer at the gas turbine because of its low work ratio, while ignoring the fact that it can be quite low in a piston engine.

in part, why the performance of the aircraft gas turbine improves so significantly at the low atmospheric temperatures prevailing at high altitudes.

Figs. 25 and 26 show that the effects of atmospheric temperature vary depending upon turbine inlet temperature. The effects also depend upon a number of other design factors. As compressor and turbine efficiencies decrease, increase of atmospheric temperature has a more unfavorable effect than shown in Figs. 24 and 25. Conversely, decrease of temperature has a more favorable effect. It can be shown that at 100% compressor and turbine efficiencies, that is, adiabatic compression and expansion, thermal efficiency is substantially independent of compressor inlet temperature. It has been shown that, at 100% compressor and turbine efficiencies, thermal efficiency is sensibly independent of turbine inlet temperature. The effects, with attainable compressor efficiencies, will also depend upon the design pressure ratio at 60 F, the higher this pressure ratio the more unfavorable the effect of increase of atmospheric temperature.

Atmospheric temperature will also affect thermal efficiency when a regenerator is used. As the temperature at the compressor inlet increases so will that at compressor discharge. Thus, for a given turbine inlet temperature less heat can be transferred from the turbine exhaust to the compressor discharge air. This effect will be partially offset, however, by the fact that increase of atmospheric temperature reduces the pressure ratio. It has been shown above that the lower the pressure ratio the greater is the effect of a regenerator upon thermal efficiency. In Figs. 24, 25, and 26 the effects of atmospheric temperature include the variation in pressure ratio which it produces. This discussion was originally written in terms of constant pressure ratio with varying atmospheric temperature. N. R. Quinn pointed out this error, which is, of course, very obvious when given sufficient thought. Quinn states that when a free turbine is employed, pressure ratio will vary substantially more than the relation used for Figs. 24, 25, and 26. This variation as set forth by Quinn is beyond the author's understanding, and the simple relation of pressure rise to atmospheric density has therefore been used. However, the variations of relative power output and thermal efficiency with varying atmospheric temperature are almost entirely due to its effect upon work ratio and only slightly affected by the change in pressure ratio. The data in Figs. 24 and 25 are almost identical with similar data computed for constant pressure ratio and for data computed upon the basis of the variation of pressure ratio given by Quinn. The effects of altitude upon turbine performance will, provided that atmospheric temperatures does not change, result in power being reduced directly in proportion to barometric pressure. However, since atmospheric temperature almost always is reduced as altitude increases, the available power will not drop anything like as fast as the barometric pressure. This has proved to be a significant advantage in gas turbine locomotives in mountain service.

Figs. 24, 25, and 26 are possibly misleading without explanation, as is Fig. 11. Figs. 24–26 do not show the effect of atmospheric temperature upon the full throttle power of a single turbine, but rather of a family of turbines each designed for a given turbine inlet temperature. A similar explanation has been given above in connection with Fig. 11

The Probable Type of Ground Vehicle Gas Turbine

Discussing the probable type of turbine unit for vehicle use involves the very dangerous field of prophesying turbine type, but fortunately not that of prophesying probable extent of the adoption of the turbine for ground vehicles. The rash predictions below apply to units of about 120 hp (2 lb air per sec approx.) minimum to those somewhat exceeding 1000 hp (about 14 lb air per sec). The author does not believe that gas turbines of less than 100 hp have any place in vehicles except those of very special type for very limited use.

In the light of present knowledge it appears that the type of turbine shown in Fig. 10 and embodying the free or power turbine is by far the most probable type for future road vehicle use. While the "single-shaft" turbine, that is, where the turbine wheel or wheels is or are directly connected to the compressor (see Figs. 1 and 5) is now successfully used for aircraft turboprops, it seems basically unsuitable for ground vehicle use. A single-shaft turbine in ground vehicle use will introduce extremely difficult and complex transmission problems unless electric drive is used (for example, in turbine locomotives).

The type of turbine shown in Fig. 10 (free turbine) may well be modified as progress is made. For example, the number of turbine wheels or stages may be increased, but this seems very doubtful and particularly so in the case of the turbine driving the compressor, since this would undesirably increase the moment of inertia. Likewise the number of compressor stages may be increased, but this again seems very doubtful on account of inertia effects, and also since the most effective pressure ratio appears to be well within the efficient range of a single-stage machine.

As regards turbine wheel type, the axial units shown in Fig. 9 may well be replaced with the radial or centripetal type. The radial type appears to have advantages in respect to simplicity, cost, ruggedness, and possibly in respect to cooling with air or liquid metals. The author has no data as to the efficiency of the radial type (the author prefers to avoid the use of the term centripetal since it is so lacking in euphony).

While the arrangement of compressor and turbine units shown in Fig. 9 seems to be that best suited to road vehicle use this, in the author's view, only applies to compressor and turbine arrangement. It is the author's *opinion* that the road vehicle gas turbine

without a regenerator has very limited fields of application. Such fields *may* be vehicles of very low weight, very short range and very high performance, and heavy vehicles operating continuously at nearly full power where power-plant weight and bulk must be kept to a minimum. For vehicles operating at a low average load factor such as passenger cars, it is the author's belief that a turbine unit without a regenerator is hardly worthy of serious consideration. Thus Fig. 10 shows the probable arrangement of the units involved in a vehicle turbine.

Most Effective Pressure Ratio—Since the pressure ratio at full load determines both compressor and turbine type, it appears appropriate at this point to consider the problem of pressure ratio in road vehicle turbines.

In a turbine unit without a regenerator, the most effective pressure ratio at full load depends upon compressor and turbine efficiencies, turbine inlet temperature, compressor inlet (atmospheric) temperature, and several other factors. It does not seem appropriate to further discuss the various factors beyond pointing out that turbine and compressor efficiencies will be markedly affected by the size of the unit (mass airflow) and by the permissible degree of mechanical complication.

In a turbine unit with a regenerator, the most effective pressure ratio will be much lower than when a regenerator is not employed. (See Fig. 19.) [2] Computation of the most effective pressure ratio for a regenerative turbine *for a vehicle* is quite beyond the author. Even though low pressure ratios tend to give the highest thermal efficiency with high efficiency regenerators there are several factors favoring use of a reasonably high full-load pressure ratio. One important factor is that of minimizing full-load specific air consumption since this will control the size of air and exhaust ducts, and intake and exhaust mufflers. Fig. 27 shows full-load specific air consumptions for pressure ratios of 2/1 to 6/1 and the problem is further discussed below. Even with a regenerator it is the author's *guess* that the most effective full-load pressure ratio will be close to 4/1.

Compressor Type and Efficiency—It is the author's belief that the centrifugal compressor will continue to be used and that the axial type will not come into use. The axial compressor is much more expensive than the single-stage centrifugal type, is much more readily damaged, and is much more affected efficiency-wise by dust, dirt, and oil leakage. The axial compressor in large aircraft turbines is significantly more efficient than the centrifugal type and in addition is suitable for higher pressure ratios. However, pressure ratios beyond the useful range of the centrifugal type seem unlikely in road vehicle turbines. Pressure ratios of up to 12/1 are in use in aircraft turbines, but it is the author's belief that such pressure ratios would not be useful in road vehicle use (they would be useful if compressor and turbine efficiency could both be raised to about 95% without regeneration). A further reason for not using the axial compressor for road vehicle turbines is the fact that in small sizes its efficiency is inferior to that of the centrifugal type. A European compressor authority, who has been responsible for successful production compressors of both types, informed the author in personal discussion that up to at least about 20 lb air per sec (roughly 1500 hp net output) the centrifugal type is more efficient than the axial type due to the high percentage of blade-tip leakage in the small axial.

Centrifugal compressors usually become less efficient as the size is reduced and as the pressure ratio per stage is increased. An efficiency of 77% in a single-stage jet-engine compressor with a pressure ratio of 4/1 and a delivery of 100 lb per sec can be considered good (this may contrast with efficiencies of over 85% at the best point for aircraft axial compressors of similar and much higher pressure ratio). At lower pressure ratios centrifugal aircraft compressors are now slightly exceeding 80% efficiency, but this involves expensive but justified construction.

The 2-stage, centrifugal compressor involves a very significant increase of cost over that of the single-stage machine (so the author has been informed by those who have built both). It is therefore the author's belief that the single-stage centrifugal compressor is the probable type for road vehicle turbines. To give the road vehicle turbine the benefit of the doubt and the benefit of development, the author is assuming that a compressor efficiency of 80% will be obtained. The horsepower involved in compressing 1 lb air per

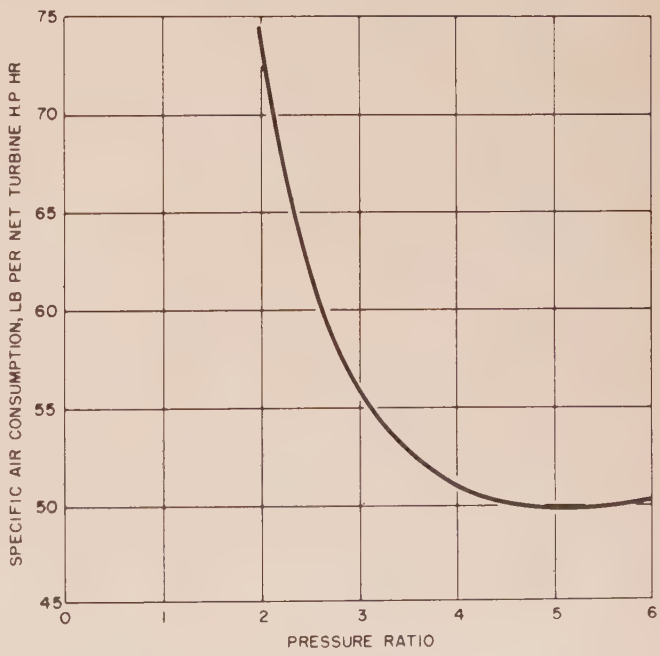

Fig. 27—Effect of pressure ratio upon specific air consumption. Turbine inlet temperature of 1500 F, compressor and turbine efficiencies of 80%, and compressor inlet temperature of 60 F

sec at pressure ratios of 1.2, 1.5, 2, 4, and 6/1 at various compressor efficiencies is shown in Fig. 28.

A pressure ratio of 4/1 in a single-stage, centrifugal compressor requires an impeller tip speed of the order of 1300–1400 fps. These tip speeds for a 6-in. diameter impeller give rpm in round figures of 50,000 to 54,000.

Turbine Efficiency—A turbine efficiency of 80% will also be assumed. Multistage aircraft turbines are achieving 85–90% efficiency at their best points. However, the road vehicle turbine is much smaller, and turbine efficiency tends to diminished with reduced size. Furthermore, the free turbine has to function as a torque converter which will reduce its efficiency since the maximum torque conversion ratio is about 2½/1. If the free turbine has an efficiency of 80% at the design rpm and is running with a torque conversion of 2/1 at 25% of design rpm the efficiency will then only be 40%. (The free turbine will, of course, develop as a maximum only about 40% of the total turbine wheel power).

Turbine Inlet Temperature—It is being assumed that a turbine inlet temperature of 1500 F will be the maximum permissible for continuous full-load operation of road vehicle turbines for the next few years. It is apparently possible to permit considerably higher temperatures during acceleration than can be tolerated continuously. There are indications, however, that excessive temperatures during acceleration produce thermal shock and other troubles.

The assumption of 1500 F as the maximum permissible turbine inlet temperature for continuous operation is predicated upon the idea that road vehicle turbines will be designed and developed with eventual volume production in view. Volume production will preclude extensive use of such critical materials as nickel, cobalt, and tungsten, which are curently usually necessary for turbine buckets of aircraft turbines operating at inlet temperatures in excess of 1500 F.

The assumption of 1500 F as the maximum permissible inlet temperature is made while bearing in mind ceramel (for example, titanium-carbide powder bonded with nickel or nickel-chromium alloy) buckets, buckets internally cooled with liquid metals, and hollow buckets cooled with air bled from the compressor. Hollow buckets cooled with compressor air were used by the Germans during World War II as a result of a chronic famine of critical materials such as nickel. Bucket cooling with air has been extensively investigated and reported upon by NACA. Aircooled buckets are now in limited service or shortly will be in British aircraft turbines. Bucket cooling, permitting increased turbine inlet temperature is particularly attractive in turboprop engines. Bucket cooling with air involves complex bucket construction and since it involves increased compressor work, this must be offset by increased turbine inlet temperature to maintain equal thermal efficiency, let alone to increase it. According to the grapevine, the increase of thermal efficiency presently obtained with aircooled buckets is about half that which would be obtained for a given increase of turbine inlet temperature, if the compressor was not bled for bucket cooling.

Steady-state turbine inlet temperatures of 2000 F may be just around the corner for aircraft turbines, but it does not appear to the author that anything much higher than 1500 F is probable for road vehicle turbines in the next decade. If the assumption of 1500 F maximum is in error, the text above is sufficient to indicate the advantages that will result from higher inlet temperatures.

Part-Load Performance—In the great majority of potential applications of the gas turbine to road vehicle use it would appear that the most important question is that of part-load thermal efficiency. In the majority of cases the specific fuel consumption of gas turbines is quoted at full load, and it is considered impolite to inquire as to the specific consumption at part load. It is usually considered to be even more impolite to inquire as to the gallons per hour consumed at idle. In the case of a passenger car developing 200 hp at the wide-open condition, but spending much of its operating life at an engine output of about 10 hp or less, these impolite questions have to be asked. Questions are one thing, but answers based on observed data for turbine engine vehicles are singularly lacking as will be seen below.

The piston engine appears offhand to have a considerable advantage over the gas turbine in respect to part-load fuel economy, since the compression ratio of the piston engine is independent of engine speed and engine output, whereas the pressure ratio of the vehicle type turbine varies with engine output. In vehicle practice this advantage of the piston engine is to some extent more theoretical than real, since while the compression ratio remains constant, the fuel economy cannot remain constant in practice due to the bugbear of engine friction. For maximum fuel economy the piston engine, at least the gasoline type, must be operated at nearly full throttle with power output varied by means of engine speed. Even with a magic transmission that would produce such a mode of operation, it is difficult to visualize a 200-hp gasoline engine which will operate at 5 hp (approximately 20 mph for a passenger car) and nearly full throttle. In practice, the lack of the magic transmission and the necessity for part throttle operation (with resultant marked increase of engine friction) results in the 200-hp engine usually having a thermal efficiency of the order of 10% to 13% at 5 hp, instead of the 25 plus % which could be obtained at full throttle.

In the gas turbine where engine power output is varied by means of engine speed and, of course, by variation of turbine inlet temperature, the pressure ratio varies simultaneously with engine speed. This variation of pressure ratio can be either an advantage or a disadvantage. It has been shown above in Fig. 16 that at 100% compressor and turbine efficiency, the

thermal efficiency increases with pressure ratio and is substantially independent of turbine inlet temperature. This is also substantially true at 95% compressor and turbine efficiencies. However, at the lower compressor and turbine efficiencies obtained in practice, reduction of pressure ratio with reduced turbine inlet temperature is advantageous in a nonregenerative turbine as shown by Fig. 16. In a turbine with a high efficiency regenerator and currently attainable compressor and turbine efficiencies, reduction of pressure ratio with reduced turbine inlet temperature is still more advantageous than it is with the nonregenerative turbine as shown by Figs. 19 and 20. Thus the characteristic variation of pressure ratio with turbine inlet temperature is, with a high efficiency regenerator, a happy combination for a turbine unit required to operate extensively at low load factors.

Fuel Consumption at Idle—The fuel consumption at idle can have a marked effect upon the average fuel mileage of a road vehicle turbine. As far as is known there are no published data for idling fuel consumption of vehicles equipped with turbines. If idling fuel consumption is to be judged by that of nonregenerative, single-shaft turboprops, and industrial turbines, the prospect is not encouraging. In such turbines the idling rpm may be as high as 90% of the rpm at rated power and the fuel gallonage one-third of that at rated power. Turbojets may idle at as low at 25% of rated rpm, and may use less than 15% of the fuel gallonage required at rated power.

In the case of the turboprop, the high idling fuel consumption discussed above is largely the result of choosing a high idling rpm so that rapid and certain acceleration to full power may be obtained. Rapid and certain acceleration to full power involves several factors of which the most important are compressor stall and surge, polar moment of inertia of the compressor and turbine wheels, and combustion difficulties such as the flame going out (blowout). Compressor stall and surge are unpleasant effects which will not be further discussed, but which are minimized if the range of rpm to be covered in acceleration is limited. The combined polar moment of inertia of the compressor and turbine rotating parts is high at the best, and as a result a large amount of energy is absorbed during acceleration. This energy storage consequently reduces the rate of acceleration. The effect is offset by limiting the range of rpm through which it must be accelerated. The combustion difficulties are related to the changes in air pressure and quantity of flow through the combustor, and the still larger changes of fuel quantity and pressure that result from large changes of rpm from the rated condition. The combustion problem of the aircraft turbine becomes increasingly difficult with altitude (reduced mass flow of fuel and air), and consequently it is largely designed around altitude performance with high idle rpm accepted as a price of good full-power behavior.

Idling at a high percentage of rated rpm means that a lot of air is compressed, and a lot of work is done on this air, with the result that a lot of fuel is necessary to do the work. In one turboprop it has been found possible to obtain a steady idle at 6% of rated rpm but this, because of acceleration time, cannot be considered for service. It may be mentioned that one centrifugal turbojet with a pressure ratio of 4/1 at rated power, idles at a pressure ratio of below 1.5/1.

In the vehicle turbine, the idling problem is distinctly different from that of the aircraft turbine and that of many industrial types. The use of a regenerator promises to very markedly reduce idling fuel consumption. Low idling rpm will be desirable to produce smooth performance at very low road speeds in a passenger car. However, even low idle rpm will not avoid unwanted power at low road speeds, when the turbine is fitted with a regenerator and has just slowed down from full power operation. This excess power due to stored heat in the regenerator will prevent the idle rpm from being reduced to its normal value. Several methods for eliminating this unwanted power are apparent. A simple one is a bypass valve between the compressor turbine and the free turbine (see Fig. 10). This waste gate would bypass the gas from the compressor turbine through the regenerator

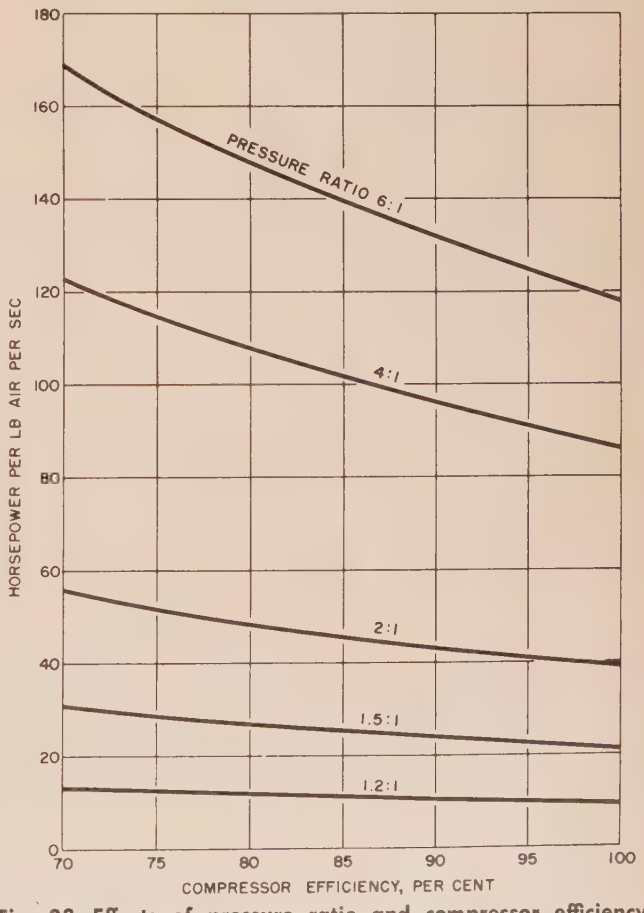

Fig. 28—Effects of pressure ratio and compressor efficiency upon power required to compress air from 60 F

and thus would partly conserve the heat in the turbine exhaust. This waste gate could also be used to eliminate creep when the vehicle is stationary. Without such a waste gate or another measure producing a similar result, high idle rpm is likely to be very objectionable in respect to creep.

Operation of the compressor at a low percentage of its rated rpm is desirable from the fuel economy standpoint at low vehicle speeds (level road) and at idle. Such operation of the compressor, however, penalizes vehicle acceleration and thus may be objectionable. It is reported that some experimental turbine passenger cars have a lag of about two sec, when nothing much happens, after full depression of the accelerator pedal. After two sec, however, it is reported that the nether regions break loose.

The problem of acceleration from idle is obviously complex. It is clear, however, that minimum polar moments of inertia of the compressor and its associated turbine are desirable. Low idle rpm may well introduce combustion difficulties, since pressure ratios of 1.2/1 and turbine inlet temperatures of 600 F are contemplated. Under these conditions combustion efficiency [19] may be seriously reduced from the 98% or higher that is attainable at rated power. In connection with combustion efficiency it may be mentioned that under high-altitude conditions in a wind tunnel, full-scale turbojets have shown efficiencies as low as 40%.

Road Load Fuel Economy—Actual road fuel economy published data of turbine vehicles are limited to the Boeing truck [10,11] and to the Chrysler passenger car discussed further below.

The Chrysler turbine with its fuel mileage of 14.9 mpg at a steady level road speed of 40 mph may be compared with current passenger cars equipped with automatic transmissions (the free turbine is, of course, an automatic transmission). The fuel mileage at 40 mph (steady, level road speed) was determined for five (1953–1954) different models of passenger cars having five different models of automatic transmission. The engines varied in size from 235- to 325-cu in. capacity. The lowest fuel mileage was 18.8 and the highest was 20.6. Fuel mileage was not related to either engine size or transmision type. On two of the engines idling consumption was determined as 1.05 gal per hr for one of the two 325-cu in. engines and 1.25 gal per hr for one of the three smallest engines. Chrysler have not published the idling fuel consumption of their turbine. The piston engine data are merely quoted for comparison and with the full realization that they could be greatly improved upon and would be improved, if fuel mileage of American passenger cars became a competitive marketing feature.

Computed road fuel economies of gas turbine passenger cars have been published by Wintringham [20] and by Beaufrere.[21] Wintringham gives a fuel mileage of almost 30 mpg over the speed range of 20–40 mph for a 140-hp regenerative turbine. The conclusions of Wintringham in respect to fuel mileage are in general agreement with unpublished investigations of a similar nature made by others.

Beaufrere (Ford) does not state the rated power of the regenerative unit he describes but shows that the best computed fuel economy is about 17½ mpg at about 40 mph, falling off fairly rapidly at higher or lower speeds. Kucher [22] does not give computed fuel mileages but does state the Ford Motor Co. viewpoint as follows: "Take the matter of fuel consumption. The very best that we can project employing known maximum component efficiencies, including high effective heat regeneration, brings us near to but not quite comparable with present internal combustion engine performance over the entire operating range." Curtice [23] has announced that General Motors has a turbine passenger car equipped with a regenerator and that the car will give fuel mileage comparable with that of current piston-engine cars.

Regenerators—The effects of the regenerator or heat exchanger upon the thermal efficiency of gas turbines are so pronounced that the use of a regenerator seems to be the obvious measure. It is particularly the obvious measure to employ since it can potentially, at least, largely overcome the low thermal efficiency of the gas turbine which is presently a major disadvantage. The potential advantages of the regenerator have long been known, and it may well be asked why the obvious has not been adopted. Regenerators for land-based or marine turbines have been and are in use, and their effects on efficiency are well known. The bulk, weight, and cost of regenerators of the tubular type which resembles a steam condenser are fairly well recognized, and this is the only type currently in service for marine and land-based turbines.

To the author, the regenerator appears to be the most difficult problem confronting the designer of road vehicle turbines. Attaining a compressor efficiency of 95% would be even more difficult and does not appear to be even beyond the horizon. However, such efficiencies, while very attractive, do not seem presently necessary for road vehicle turbines. Possibly further discussion of the major phase of the difficulty is warranted. In current passenger car turbine developments, regenerator efficiencies of more than 80% are being discussed and it may be worth pointing out some of the difficulties of obtaining this. The first and most obvious regenerator case is the one where the turbine exhaust loses heat to the compressor discharge air which gains heat until the two gas streams are in temperature equilibrium. This case is shown by A in Fig. 29 and will represent a regenerator efficiency of 50%, neglecting differences in specific heat. B in

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[19] Combustion efficiency is defined as the actual enthalpy increase divided by the enthalpy increase which would be obtained if all fuel constituents were completely oxidized (that is, to carbon dioxide, water, and sulfur trioxide).

[20] *Ethyl News*, July–August, 1955: "The Gas Turbine and Its Possible Application for Automotive Uses," by J. S. Wintringham.

[21] SAE *Journal*, March, 1955, and May, 1955, "An Exploration of the Automotive Gas Turbine," by A. H. Beaufrere.

[22] "Turbos for Vehicles," by A. A. Kucher. Presented at the Fifty-third Annual Meeting of the National Petroleum Association, September, 1955.

[23] Harlow H. Curtice, quoted in the daily press, December, 1955.

Fig. 29 represents countercurrent flow of turbine exhaust and compressor discharge air, with the air finally reaching turbine exhaust temperature which gives a regenerator efficiency of 100%. It is obvious that to get efficiencies of over 50%, methods involving the principle of countercurrent flow are essential.

There are few published data concerning regenerators for vehicle turbines. Chrysler have merely stated that their turbine automobile has a regenerator. Chao has described the Ford rotary regenerator in some detail.[24] Numerous patents have been issued and are being issued for regenerators of much more compact arrangement than the tubular type, but whether many of these are intended for vehicle use is not clear. It may be mentioned that one of the regenerator problems is that of its taking fire and destroying itself.[25] It is the author's opinion that the regenerator is a very large portion of the vehicle turbine problem.

Chrysler Regenerator—The recent publicity releases concerning the Chrysler passenger car turbine which uses a regenerator appear to put the use of regenerators for ground vehicles in an entirely different light from anything hitherto known. Chrysler has given no details of the regenerator but the published photographs show that its bulk is quite small. The Chrysler turbine installed in a Plymouth car is stated to give a fuel mileage of 14.9 mpg (2.68 United States gal per hr) at a level road steady speed of 40 mph. This is an outstanding achievement which appears to put the road vehicle gas turbine in an entirely different league from anything heretofore published. It may be assumed that the engine had a maximum (rated) output of about 120 bhp (which would be necessary to match the performance of the standard, 1954 6-cyl, piston engine Plymouth). It may also be assumed that at 40 mph the turbine was developing about 15 bhp and at this output the computed specific fuel consumption will be about 1.1 lb per bhp-hr for gasoline and about 1.2 lb per bhp-hr for kerosene (the fuel used was not specified). Either of these consumptions is extremely good for an open-cycle gas turbine operating at a load factor of 12½% (15/120). The thermal efficiencies (assuming 18,500 Btu per lb lower heating value) are 12.5% for gasoline and 11.6% for kerosene.

Overall Performance of a Complete Free Turbine Unit—The author has been curious to examine the performance parameters of a probable free turbine unit suitable for vehicle use.

The characteristics of the turbine unit chosen for analysis are as follows:

1. 4/1 pressure ratio at rated power.
2. 1500 F turbine inlet temperature at rated power.
3. 3 lb per sec mass airflow at rated power.
4. 80% turbine and compressor efficiencies at all pressure ratios and turbine inlet temperatures.
5. 60 F compressor inlet temperature.
6. Regenerator having an efficiency of 75% at rated power and 90% at a pressure ratio of 1.2/1. Regenerator efficiency, following Chao,[24] is assumed to be a straight-line function of mass airflow between pressure ratios of 4/1 and 1.2/1.

Compressor pressure ratio is assumed to vary in terms of rpm as shown in Fig. 30. Fig. 30 is based upon constant efficiency and the relation that work on the air varies as $(rpm)^2$. Compressor mass airflow is also shown by Fig. 30. This relationship is derived from the operating line of a turbojet with 4/1 pressure ratio. This compressor has a straight-line relationship between mass airflow and rpm but, as shown, the line does not go through zero rpm. This compressor does not follow the relationship between pressure ratio and rpm shown in Fig. 30.

Turbine inlet temperature is assumed to be 600 F at 1.2/1 pressure ratio and, to have a straight-line relationship in terms of pressure ratio between 1.2/1

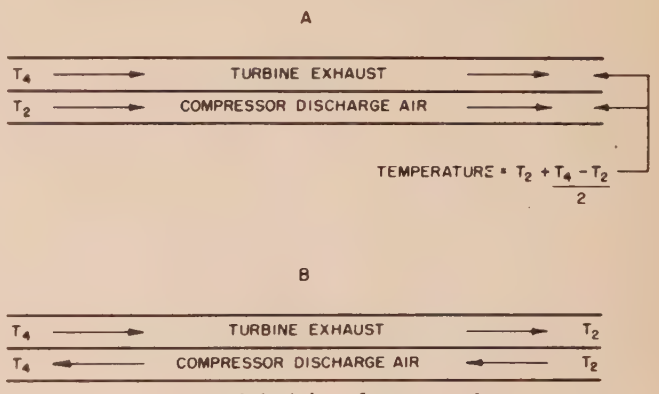

Fig. 29—Principles of regeneration

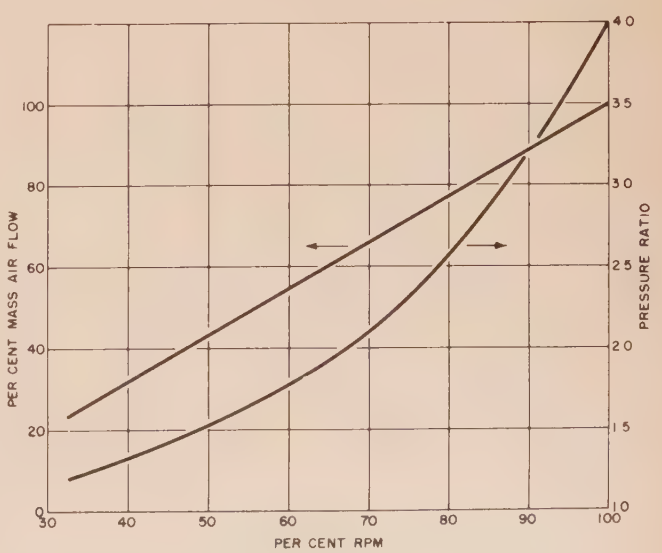

Fig. 30—Compressor characteristics of "probable" free turbine unit

24 Proceedings of the American Power Conference, Seventeenth Annual Meeting, Vol. XVII, 1955: "Research and Development of an Experimental Rotary Regenerator for Automotive Gas Turbines," by W. Wai Chao. See also "Automotive Industries," May 15, 1955, pp. 54–56.

25 *Engineering*, Jan. 29, 1954, pp. 146 and 147: "Gas Turbines at Beznau Power Station," by E. A. Kerez.

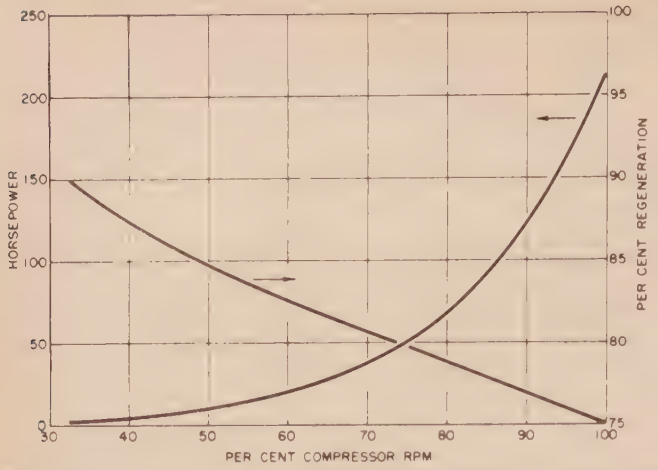

Fig 31—Power output and per cent regeneration versus per cent compressor rpm of 212-hp "probable" turbine unit

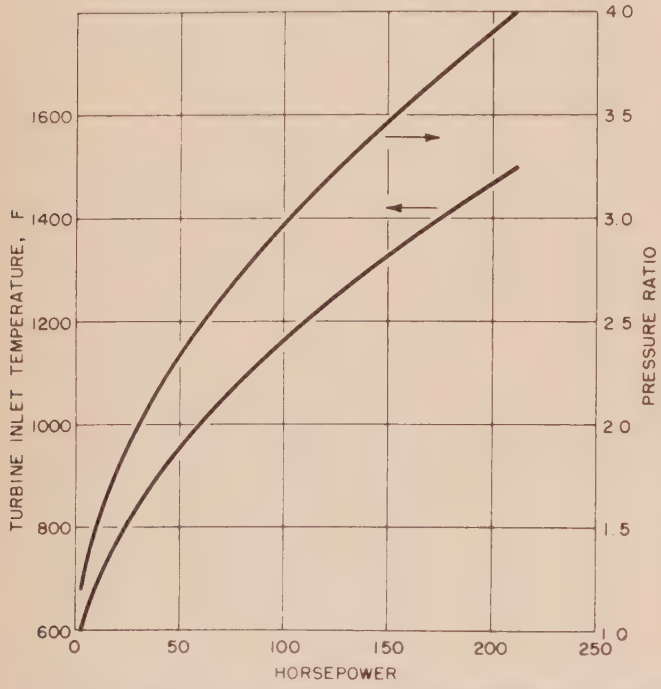

Fig. 32—Pressure ratio and turbine inlet temperature versus power output of 212-hp "probable" turbine unit

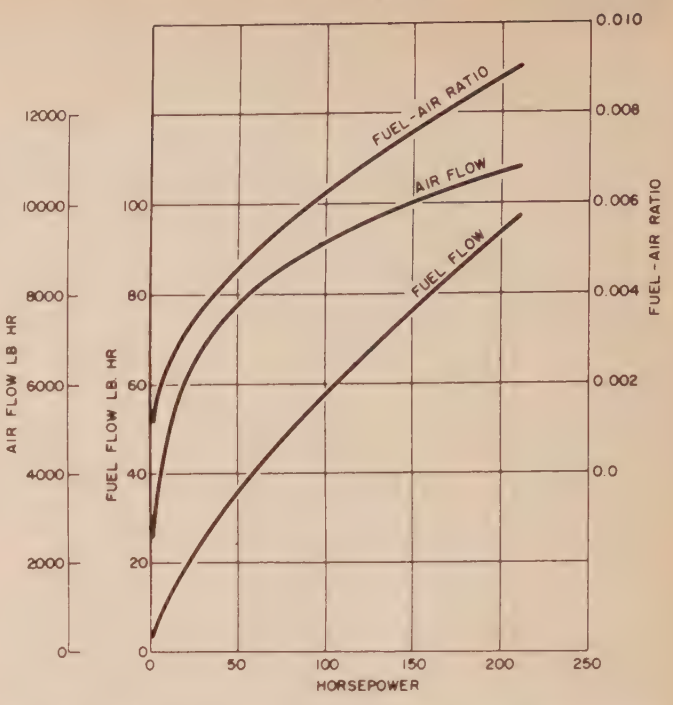

Fig. 33—Airflow, fuel flow, and fuel/air ratio versus power output for 212-hp "probable" turbine

pressure ratio and the 1500 F value at 4/1 pressure ratio. Derivation of the relationship between turbine inlet temperature and overall pressure ratio for a free turbine unit is beyond the author, and the relationship used is a slightly simplified version of one worked out by Quinn for the compressor and turbine conditions given above. Quinn points out that the straight-line relationship just happens to occur for the conditions chosen and is not a general relationship.

No pressure drops or leakages are allowed for, and the computation practices are those used earlier in this paper. The fuel is assumed to have a heating value (lower) of 18,500 Btu lb, and to have a sp gr of 0.75 giving a gallonage weight of 6.25 lb. Combustion efficiency is assumed to be 100%.

The performance characteristics are shown in Figs. 31–35 inclusive. Items of performance interest appear to be the enormous air consumption of nearly 5½ tons per hr at "full throttle" and the equally enormous consumption of about 1¼ tons per hr at idle. The fuel/air ratio of 0.00115 (air/fuel ratio 870/1) at idle is also of interest. If it is assumed that 5 hp is required by a passenger car at 20 mph and 15 hp at 40 mph, the fuel mileages work out at 20 mpg at 20 mph and 16.4 mpg at 40 mph. The idle (1.2/1 pressure ratio and 600 F turbine inlet) fuel gallonage comes out at slightly less than ½ gal per hr.

The whole analysis is at the best speculative, since so much depends upon the chosen values for component efficiencies, and this is particularly true in respect to thermal efficiency. Fig. 35 shows thermal efficiency versus power output for the regeneration efficiencies used in the analysis and also for efficiencies of zero and 100%. A 100% regenerator efficiency is entirely academic, but its comparison with the values chosen for the analysis indicates the very marked effect of a slight change, and particularly so at the very low load factors. Variation of compressor and turbine efficiencies will produce similar effects as has been shown above. The fuel mileages quoted above with trepidation are in considerable contrast with those of Wintringham.[20] Wintringham allowed for losses which the above analysis does not, but he used higher component efficiencies. The enormous influence of component efficiencies has been reiterated ad

infinitum by all analysts of the gas turbine problem.

Effect of Turbine Size Upon Road-Load Fuel Economy—The effect of varying rated engine output upon road-load fuel economy has been roughly computed for the "probable" turbine discussed above. With all unit efficiencies as given above, the effect of engine size has been computed for units of 1½-lb airflow per sec giving a rated power of 106 hp and for one of 6-lb airflow per sec giving a rated power of 425 hp. The effect was computed for the supposedly most unfavorable condition of 20-mph road load and the computations are based on Fig. 35. The results show that doubling the size of the turbine reduces fuel economy by 16% giving a mileage of 16.8 mpg. Similarly, halving the size of the unit increases fuel economy by 15% giving a mileage of 23 mpg. These relatively slight changes seemed unbelievably small, but an authority in the vehicle turbine field assured the author that the percentages, at least, are of the right order of magnitude.

Specific Air Consumption of Turbines—Specific air consumption of turbines has been discussed above under most effective pressure ratio and shown in Fig. 27 for a turbine inlet temperature of 1500 F with compressor and turbine efficiencies of 80%. The effects of turbine and compressor efficiences and of turbine inlet temperature are shown in Fig. 36. Specific air consumption will also be affected markedly by compressor inlet temperature but this is not explored here.

The data shown in Fig. 33 for the "probable" road

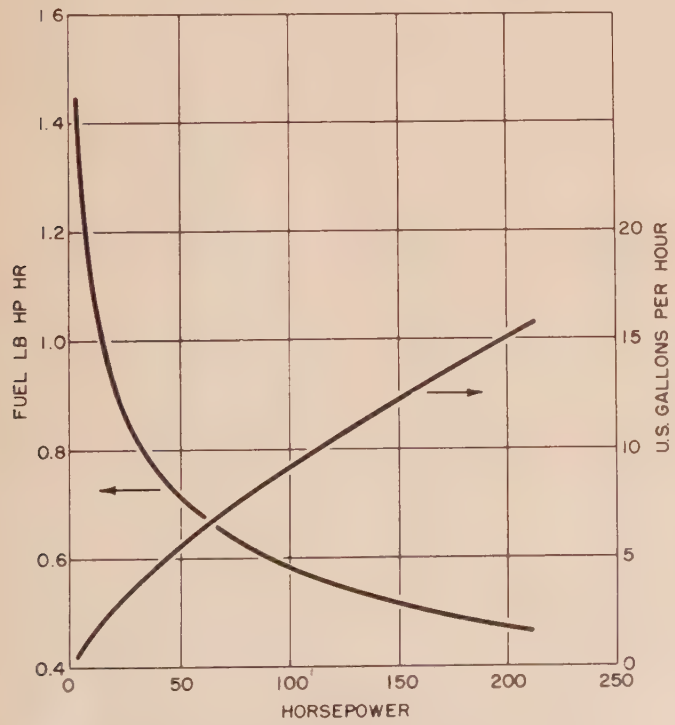

Fig. 34—Specific fuel consumption and fuel flow in gal per hr for 212-hp "probable" turbine

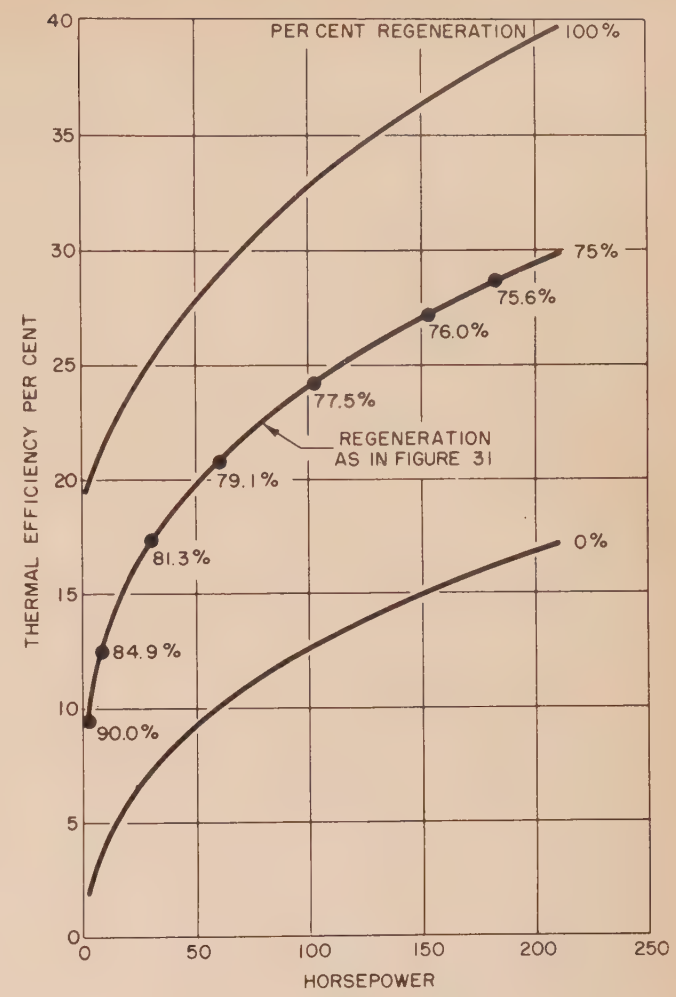

Fig. 35—Thermal efficiency of 212-hp "probable" turbine with zero, specified, and 100% regeneration efficiencies

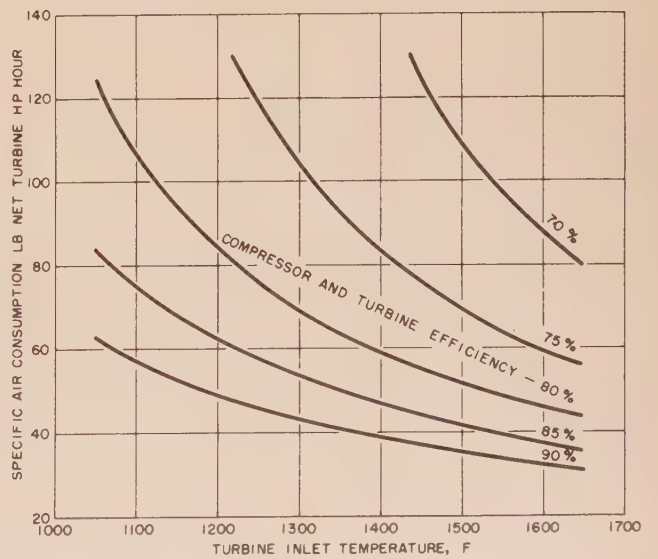

Fig. 36—Effects of compressor and turbine efficiencies and turbine inlet temperature upon specific air consumption. Pressure ratio 4/1 and compressor inlet temperature of 60 F

vehicle turbine of 212 rated hp and discussed above show the *relatively* enormous quantities of combustion air [26] which must be handled in a turbine vehicle. A gasoline engine of equal rated power would require about 1700 lb air per hr at full power, in contrast with 10,800 lb per hr for the turbine engine. These figures, at sea level and 60 F, correspond to about 370 cfm for the gasoline engine and about 2350 cfm for the turbine. Even at idle the "probable" turbine will use about 500 cfm.

The large air supply required by the gas turbine must be taken into the compressor with minima of heating and pressure drop, likewise the exhaust back pressure must be a minimum. Intake air heating and pressure drop not only reduce output but seriously affect thermal efficiency, and exhaust back pressure has the same effects. A regenerator has valuable effects in respect to the problem of handling the exhaust. In the case of the "probable" turbine discussed above, the regenerator will, at rated output (1500 F turbine inlet), reduce the exhaust volume by about one-third and reduce its temperature by about 500 F. In connection with exhaust volume and exhaust temperature of turbine vehicles, the remark of a witty friend may be quoted. Said friend watched a turbine vehicle which was not equipped with a regenerator and described it as "a heat wave with a vehicle under it." In a piston engine, intake air heating, air cleaner and carburetor pressure drop, and exhaust back pressure reduce output significantly but do not have any serious effects upon fuel mileage. The high frequency noises of the turbine air intake and exhaust probably involve difficult problems, if muffling is to be accomplished with minimum pressure drops. The high frequency problems plus the large volume to be handled make intake air handling and muffling, and exhaust disposal and muffling into a significantly more difficult job than that of a piston engine. The aerodynamic problems of handling large volumes of air with minimum pressure drops in the ducts of aircraft turbines have been shown to demand vigorous attention. It is not easy to improve turbine and compressor efficiency, but it has proved quite easy by bad duct design to lose such improvements after they have been laboriously obtained.

One bright spot in the problem of combustion air supply of a vehicle turbine is that the air will *probably* not need much cleaning even when a regenerator is used. Bolts, nuts, and stones must obviously be screened out. The centrifugal compressor fortunately appears to be immune to damage by ice. Another bright spot is that the total air required for combustion and engine cooling will be far lower for the turbine. This will at least be true at rated power but will probably not be true at idle.

Fuel Metering Systems—Since this paper is mainly concerned with performance, the mechanical problems involved in fuel metering are largely outside its scope. The ideal fuel metering system would involve a given position of the accelerator pedal defining a given turbine inlet temperature. For this desired inlet temperature the metering system must adjust fuel flow rate for variation of mass airflow due to barometer (including effect of altitude) and to atmospheric temperature; and variation of compressor discharge temperature due to atmospheric temperature and to heating of atmospheric air in the engine compartment. Variation of compressor discharge temperature ceases to be the significant variable when a regenerator is used, and the control variable then becomes that of regenerator discharge temperature. The fuel control with a regenerator has a difficult job, since a very wide range of discharge temperatures from the regenerator must be catered for at a given turbine inlet temperature. Thus the regenerator will be cold after prolonged idling, and if the engine is then opened to "full throttle" and kept there, the regenerator will start storing heat until equilibrium is reached. During this period of heat storage, the control must progressively reduce the rate of fuel supply. If the control problem parallels that of the aircraft turbine, it will also be necessary to regulate the rate of increase of fuel supply so that the compressor does not surge.

All in all, the fuel metering system for a road vehicle turbine would appear to add up to much more of a problem than a carburetor. It will, however, do things that a normal vehicle carburetor does not do, two of which will be compensation for atmospheric temperature and altitude. Misbehavior of a carburetor often means nothing more than either a nuisance or an excessive fuel bill. Malfunctioning of the fuel control of a turbine may mean such things as flying turbine buckets or melted combustors. Despite the problems of fuel metering control of the vehicle turbine, they appear to be vastly simpler than those of the aircraft turbine of either the propeller or jet type.

Combustion Systems—The problem of combustion in a gas turbine is a very difficult one. Initially the problem seemed the simple one of merely burning fuel in air as in a blowtorch, but experience has shown otherwise. The burner or combustor was probably the most difficult initial problem that Whittle had to overcome in his first jet engines. Von Ohain recognized the combustion problem and evaded it initially by using hydrogen.[4] In the light of present knowledge it would seem that the combustion problems of the gas turbine will turn out to be more complex than those of the gasoline or diesel engines.

The problem of the combustion chamber, known as the combustor in aircraft circles at least, may be considered first. As mentioned above, the major combustion difficulty of gas turbine combustion is due to the fact that the overall fuel/air ratio is so lean. A gas turbine with presently permissible turbine inlet temperatures does not burn more than about 25% of the oxygen in the air. If the fuel were uniformly mixed

[26] The term combustion air is used for all air which passes through the engine even though most of it does not take part in combustion and is really dilution air.

with the air, the mixture would not be combustible. The maximum F/A ratio used in open-cycle gas turbines is about 0.018 and about 0.04 is about the leanest that can be burned in a high-compression gasoline piston engine and then only with difficulty. Consequently, in a turbine the fuel has to be burned with only part of the air at approximately stoichiometric F/A ratio (about 0.07 for turbine fuels). It is desirable that combustion of this stoichiometric mixture be complete, before additional air is added to dilute the products of combustion down to the overall F/A ratio which will produce the desired turbine inlet temperature. While the maximum F/A ratio used is about 0.018, at low loads the F/A ratio in aircraft turbines is as low as 0.01, that is, it is burning only about 14% of the available oxygen.[27] In a vehicle turbine it will be necessary to burn much leaner mixtures than 0.01 F/A ratio if a regenerator is used. In addition to the difficulties due to the lean mixture is the fact that the gas speed through the combustor is probably about 50 fps even in the zone where stoichiometric mixture is being burnt. This 50 fps is far in excess of the speed with which flame will spread through quiescent mixture (known as the laminar flame speed). The work of NACA [28] indicates that the laminar flame speed of turbine fuels is of the order of 1.3 fps or 40 cm per sec. This flame speed will, of course, be greatly increased by random turbulence in the mixture, but this cannot be expected to increase flame spread in the mixture to the rate at which the gas is passing through the primary zone of combustion. Thus, since the flame speed is much less than the velocity of the mixture through the combustor (primary zone), the flame can be expected to move down the axis of the combustor and out through the turbine, producing blowout. This can and does happen at times in aircraft turbines when combustion conditions are difficult (such as at very high altitude). Fortunately, by suitable turbulence in the primary zone the flame is apparently made to circulate back into the unburned and partially burned mixture as shown in Fig. 7. This circulation produces a stable flame in primary zone and is a product of swirl produced by vanes at the primary air entrance and turbulence produced by the combustion process itself and by the fuel spray where fuel is injected by a spray nozzle. That a flame circulation process similar to that shown in Fig. 7 occurs in practice appears to be fairly well established even though visual observation is difficult.

In a vehicle turbine the combustion problem does not have some of the limitations of the aircraft turbine, since weight and space requirements will not be as difficult as in the aircraft type. Cost will, however, be important in the vehicle turbine as will the necessity to avoid critical materials. It is the author's opinion that prolonged and steady operation at very low turbine inlet temperatures as in a passenger car turbine in dense traffic with efficient combustion will prove to be a difficult but not insuperable problem in the vehicle turbine combustor. There are those who think that the high temperature of the combustion air, which is the result of a regenerator, will ease the combustion problem of the vehicle turbine. The high temperature of the air prior to fuel injection which is the result of the regenerator would appear to be of assistance in combustion. This may, however, be offset by the fact that the greater the temperature rise produced by the regenerator, the less the fuel to be injected and the leaner the overall fuel/air ratio.

In a vehicle turbine, efficient combustion is desirable at all turbine inlet temperatures for reason of fuel economy. Inefficient combustion and particularly so at low turbine inlet temperatures, may be very objectionable on account of odorous (aldehydes?) exhaust. Unpleasant exhaust odor may be more important than increased fuel cost due to inefficient combustion. As a matter of fact, it is possible that inefficient combustion may produce a highly unpleasant exhaust without a significant effect upon the fuel bill. In this matter time will tell. In some cases turbojets emit fairly black smoke despite highly efficient combustion, the loss in combustion efficiency due to smoke is often too small to be determined.

No attempt will be made to prophesy the most suitable type of combustor for the vehicle turbine. It is obvious however, that efficient combustion will be an important factor in vehicle turbine performance.

Fuels for Turbine Vehicles

Since this paper is mostly concerned with performance, only limited discussion of the fuel problem appears to be appropriate.

With the advent of the gas turbine the idea arose that any liquid that would burn could be used as a fuel. While residual (bunker) fuels are used in marine, land-power station, and locomotive gas turbines, it has been shown that such fuels can produce many operating difficulties. Bunker fuels often produce deposits which rapidly reduce the available power, the deposits in some cases produce rapid and destructive corrosion, if vanadium is present which it frequently is.

The aircraft turbine has shown that the more volatile the fuel, the easier the *combustion* problem; the overall fuel problem of the complete aircraft is another matter, however, and in this case kerosene may be greatly superior to gasoline. The land-based power station turbine has also shown the advantages of fuel volatility, since methane (natural gas) has been shown to be greatly superior to bunker fuels.

The aircraft turbine has also proved to be very

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27 The combustion problems of the gas turbine and the diesel engine are similar in that neither burn all the oxygen in their combustion air supplies. Both seek to completely burn fuel at approximately stoichiometric F/A ratio and then to uniformly dilute the combustion products with the residual unburned air. In the diesel engine the range of overall F/A ratios will cover the range from about 0.05 (about the maximum for smokeless combustion, that is, burning about 75% of the oxygen) down to 0.01 or leaner.

28 *Industrial and Engineering Chemistry*, Vol. 46, No. 10, October, 1954, p. 2151: "Effect of Molecular Structure on Combustion Behavior," by L. C. Gibbons, H. C. Barnett, and M. Gerstein.

choosey about the combustion properties of the hydrocarbons used in its fuel, and the most suitable fuels appear to be those which give cleanest burning in a lamp using a wick. The aircraft turbine has shown a particular dislike for aromatics boiling at over 400 F.

Assuming very widespread use of road vehicle turbines, it is the author's *opinion* that the fuel used will resemble either motor gasoline or JP4, which in practice is about two-thirds gasoline. In either case, low aromatic content would be desirable as would straight-run products which, however, would probably be unavailable in the event of a large demand. For road vehicle use, gasoline or JP4 have many advantages over heavier fuels in respect to low freezing point, ease of handling, settling and removal of dirt, ease of starting, and so forth, in addition to the combustion advantages discussed above.

The above discussion cites the author's *opinion* that the road vehicle turbine has a distinct fuel problem. The problem, however, would not appear to be a difficult one for the petroleum industry to meet, and the *potentially* available supplies would seem to be greater than that of either current motor gasoline or high-speed diesel fuel.

Forgetting cost and specific fuel consumption, methanol probably is the most nearly ideal liquid fuel for a gas turbine. The ideal fuel, if all practical considerations are forgotten, is probably hydrogen preferably metered as a liquid. It has a very high flame speed, has a very wide range of combustibility, and would produce no deposits or corrosive residues. Von Ohain's turbojet engine appears to have been the first to be tested and, as mentioned above, avoided the known combustion difficulties of liquid fuel by using hydrogen.[4]

Conclusions

The author, having been rash enough to express opinions as to the most suitable turbine type for vehicle use, may as well summarize these views since this does not involve further sticking out of his neck.

These opinions may be summarized as follows:

1. Vehicle turbines with acceptable part load fuel economy appear, in the light of present knowledge, to be not only technically possible but to have been produced experimentally.

2. The vehicle turbine will differ basically from the aircraft gas turbine since the vehicle type will, in the great majority of potential applications, embody a regenerator.

3. The vehicle type will use the free turbine which is an effective torque converter and constitutes a major portion of an acceptable automatic transmission.

4. In further contrast to the aircraft turbine where high pressure ratios are most efficient, the vehicle type will use only a moderate pressure ratio which is most effective for use with a regenerator. The moderate pressure ratio predicted for the vehicle turbine is readily obtained with a simple, single-stage centrifugal compressor.

5. Maximum continuous turbine inlet temperature will be limited to about 1500 F by availability of materials. The matter of materials has received only limited discussion above but it may be emphasized here that even 1500 F will be difficult in the light of present knowledge, if the vehicle turbine comes into extensive use.

6. The power output and thermal efficiency of gas turbines are very adversely affected by high atmospheric temperature and conversely markedly improved by very low temperatures. (This is not an opinion but a fact supported by much service data.) The effects of atmospheric temperature upon the power output of gas turbines may be more than five times as great as in piston engines. The power loss of the turbine due to atmospheric temperature is, however, offset by a lower loss than a piston engine due to increased altitude.

7. Efficiencies of turbine, compressor, and regenerator are of extreme importance, as has been said so many times by others in the past. Currently attainable compressor and turbine efficiencies are adequate for pasenger car use provided a regenerator efficiency of about 90% can be obtained at very low load factors. At low load factors, a 1% gain in regenerator efficiency has a marked effect upon fuel mileage. The regenerator is the most difficult design and development problem facing the vehicle turbine engineer.

8. The combination of a turbine unit in which the pressure ratio is reduced as the load is reduced is, with a highly efficient regenerator, a happy combination for a turbine engine which must operate efficiently at very low load factors.

9. Many difficult problems await solution in the vehicle turbine (they may have been solved in secret) such as: minimum lag in acceleration from rest, suppression of creep, and an adequate and simple fuel control which will compensate for altitude and the widely varying heat output of a regenerator. The very important possible difficulty of turbine engine cost has not been discussed.

10. An efficient regenerator has very marked and very desirable effects in reducing the temperature and volume of the exhaust of a gas turbine. The exhaust of a gas turbine at low load factors contains very low percentages of combustion products such as carbon dioxide, water, and sulfur oxides. While, at idle or near idle, the exhaust gases may consist of as little as 3% carbon dioxide, and so forth, in 97% air, this does not ensure that the exhaust will not have an offensive odor.

Apologies

This paper is full of the author's opinions, views, and beliefs. Considering the author's exceedingly

limited gas turbine background, this is hardly surprising. However, even in the well-developed field of aircraft turbines, widely experienced authorities today often qualify their statements with "I think" or "I believe."

Acknowledgments

The author is very appreciative of the very considerable assistance he has received in the preparation of this paper. In particular he acknowledges the extensive help of N. R. Quinn of the Bristol Aeroplane Co., Ltd.

Reviews by many piston engine engineers who are without turbine experience have been exceedingly helpful. The editorial assistance of J. R. Howard has been invaluable, and the paper would have been much better arranged and expressed had time permitted extensive further editing. Thanks are due to General Motors Research Laboratories for permission to use Fig. 9.

The author would in particular acknowledge the help of Ethyl Corp. which has checked computations, supplied test data, prepared illustrations, and printed the text. Despite this help, Ethyl Corp. is in no way otherwise associated with the paper which is entirely the author's responsibility, and the views expressed therein are his and his alone.

DISCUSSION

Simplifying Assumptions Result in Neglect of Losses in Practical Engine

—C. A. Amann

General Motors Corp.

WHILE there are a few statements in the author's paper which might provoke some disagreement, he is to be congratulated on presenting a very thorough introduction to the elements of gas turbine cycle analysis. The comprehensive part load analysis of a gas turbine engine, such as has been undertaken by the author, is indeed a laborious and time-consuming operation. Our research staff has programmed this problem for an electronic computer in such a way that the performance characteristics of a newly modified compressor, or turbine, or regenerator can be fed into the computer. The effect of that modification on overall engine performance can then be determined without actually building the modified component into the engine. This program, which condenses six man-months of hand computation per engine configuration into 15 min of time on the computing machine, has made it practical for us to study more thoroughly the complex interrelationship of variables involved in the cycle. The author has circumvented the six months of hand computation by making a number of simplifying assumptions.

Table A—Effect of Losses on Engine Performance

	Output, hp	Bsfc, lb/hp-hr
Ideal Engine	212	0.465
+Pressure Loss	178	0.535
+Leakage Loss	164	0.556
+Mechanical Loss	151	0.602

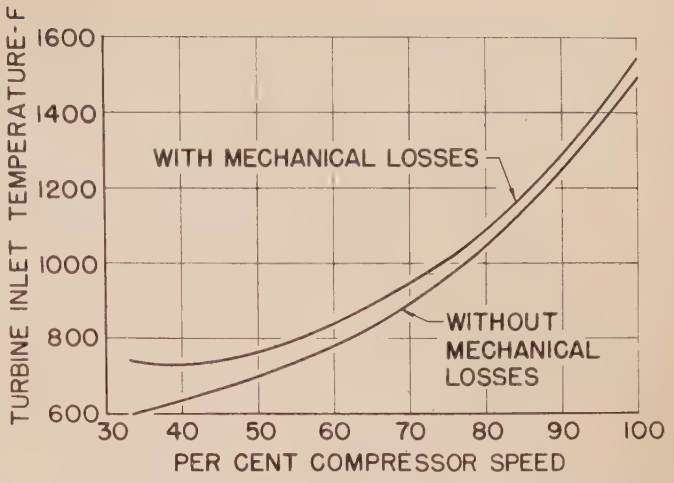


Fig. A—Effect of mechanical losses on turbine inlet temperature

One of the most significant of these assumptions is the neglect of various losses which inevitably occur in a practical engine. While these losses do not materially affect most of the trends indicated by the calculation, they do significantly affect the magnitude of some of the results. The losses might be divided into three types: pressure losses, leakage losses, and mechanical losses.

Pressure losses resulting from flow friction, from changes in cross-sectional area or direction of flow, and from heat transfer are always present in the air-intake and -exhaust ducting, the regenerator, and the combustion chambers. The result of these losses at a fixed compressor pressure ratio is to reduce the pressure ratio, and hence the output, in the turbine. Since the compressor continues to absorb the same energy in spite of the losses, less energy is left for delivery to the output shaft. The author's hypothetical engine has a compressor pressure ratio of 4 and a flow rate of 3 lb per sec with a turbine inlet temperature of 1500 F. Under these conditions it delivers 212 hp with a specific fuel consumption of 0.465 lb per hp-hr as indicated in Table A. If the compressor pressure ratio, flow rate, and turbine inlet temperature are fixed but the turbine pressure ratio is reduced to a typical value of 3.6 to allow for pressure drops, the 212-hp engine becomes a 178-hp unit, and the specific fuel consumption rises to 0.535 lb per hp-hr.

The second source of loss is leakage of high pressure air. This air, which has already had considerable work expended on it in the compressor, may leak across the regenerator seals or be carried over in the regenerator matrix to the low pressure turbine exhaust gas. Some of it may be used intentionally to cool the turbine disks. Regardless of how it is lost, it demands more work from each pound of gas flowing through the turbine in order to satisfy the requirements of the compressor. This leaves still less energy for the output shaft. If a leakage allowance of 4% of compressor discharge air is made, what was a 178-hp engine a moment ago now delivers 164-hp, and at an increased specific fuel consumption of 0.556 lb per hp-hr.

The third source of loss is mechanical. Bearings and accessories on the compressor shaft might typically consume 8 hp in a 200-hp engine. Of this 30% might go to the electric generator, regenerator drive, oil pump, fuel pump, and tachometer generator, the rest going directly to friction. Perhaps 3% of the power which is developed in the the output turbine is lost in the reduction gear box. Including all these losses, the author's 212-hp engine has an output of 151 hp, and the specific fuel consumption has risen from 0.465 to 0.602 lb per hp-hr. The conclusion to be drawn here is that a practical engine having the design characteristics used by the author might exhibit a minimum specific fuel consumption some 30% poorer than that indicated in the paper while requiring 40% more air to deliver the same power output.

The author states in his paper that he is convinced that turbine inlet temperatures as low as 600 F will be necessary for reasonable fuel economy at light loads. He then uses a linear variation

of turbine inlet temperature with pressure ratio, from 1500 F at full load to 600 F at idle, as a basis for his part load analysis. Unfortunately an engine of fixed geometry may not choose to idle at a turbine inlet temperature as low as 600 F. It has been our experience that the plot of turbine inlet temperature versus compressor pressure ratio is fairly linear above a pressure ratio of 2 or 2.5, but such has not been the case all the way down to idle pressure ratio. Some of the losses neglected by the author for simplification help to prevent this. In Fig. A the turbine inlet temperature of the author's engine is plotted against compressor speed. Now let us add to the compressor shaft a mechanical loss which varies as the square of compressor speed and reaches 8 hp at full load. The turbine inlet temperature curve shown with mechanical losses results. Minimum turbine inlet temperature is reached at about 40% speed. Futher reduction in compressor speed results in increased turbine inlet temperature, and 600 F is never reached. That an accessory load has this effect has been very graphically demonstrated in General Motors' gas turbine-powered turbocruiser. In this vehicle loading the electric generator and service air compressor at idle speed increases the turbine inlet temperature 100 F.

In an actual engine pressure and leakage losses also contribute to the shaping of the turbine inlet temperature curve. In addition the compressor and turbine efficiencies do not remain constant as in the author's engine, but at idle are usually below their maximum values. Because of these and other factors it is not unusual for minimum turbine inlet temperature to occur above 40% compressor speed. Idling the engine at speeds below that at which minimum turbine inlet temperature prevails is perfectly acceptable as long as overheating of engine parts does not occur, but high idle temperature does adversely affect acceleration of the engine from the idle condition.

Use of Gas Turbine in Passenger Cars Would Require Further Improvements

—D. P. Barnard
Standard Oil Co. (Ind.)

THE petroleum industry is intensely interested in the fuel requirements of the automotive gas turbine. We must be prepared if, and when, it is accepted. Mr. Heron suggests that JP4 would be a suitable fuel. However, our economic studies show an incentive to use heavier fuels if they can be burned without undue penalty. Further, we feel it will be a necessity to include cracked components in any diet for an extensively used automotive gas turbine.

A question most important to us, which Mr. Heron has not discussed, (and to which, incidentally, we have as yet heard no answer from any source), is, "Why should the gas turbine be considered as a future passenger automobile engine?" On the basis of the entire powerplant of the engine, the transmission, and all the necessary accessories, just what does the gas turbine offer to justify the intense interest in it for the passenger automobile? Of course, it may lend itself to novelty and glamor appeal and might, therefore, take its place with other expensive sales aids. Other than such nonfunctional appeal, however, it would seem that the turbine's inherent lighter specific weight is its main potential advantage. We presume that a turbine might save anywhere from two to four pounds per actual horsepower, or about 500 lb of powerplant weight per car. Perhaps it might also save a bit of space, but it would seem that this difference would not be too great. Mr. Heron's figures show, as have other calculations and a very limited amount of experience, that a gas turbine can be a little better or a little worse than the gasoline engine on the basis of operating expense—primarily fuel cost. In any case fuel costs are probably competitive enough so that this difference would be hardly important enough to impress the American motorist.

To date the American motorist seems to have judged appearance, performance, comfort, and reliability more important than first cost, car size, or fuel economy. It would appear that the turbine must give performance which is at least as good as that of the piston engine on each and every point, and that reliability must also be as good even in the face of abuse and maladjustments. We would think that the slow throttle response of the turbine (as we have thus far seen it) must be markedly improved before it can hope for widespread public acceptance.

If the turbine permits a reduction in powerplant weight it might lead to a more attractive automobile at no overall increase in cost. Obviously such a possibility justified almost any development effort, although, as we are aware, such a possibility seems anything but assured. Our impression is that the turbine's chance of replacing the gasoline automobile engine on any extensive scale will be slim indeed unless costs are equal to or lower than the piston engine automobile.

One technical point in Mr. Heron's paper would seem to call for more comment. Mr. Heron says that he neglected pressure drop. Some figures indicate that a 10% increase in pressure drop will spoil fuel economy by a like amount. Therefore, Mr. Heron's efficiency figures may be a little high. We do agree, however, that this would not be a major item in determining acceptability of the turbine.

We would welcome the author's comments on these questions:

1. What are the main incentives toward a turbine powerplant for the passenger automobile?
2. What fuel economy relative to that prevailing with the piston engine is judged acceptable?
3. What fuel characteristics are deemed necessary?

Author's Closure To Discussion

MR. AMANN shows, in rather devastating fashion, that the simplified treatment used in the paper can be misleading, and I am glad that his discussion is on record as a warning.

I can only wish that he had included a curve showing per cent compressor speed versus power output and specific fuel consumption. Such a curve would serve to show if the turbine passenger car has a chance of equalling the fuel mileage of current models.

In defense of the simple treatment used in the paper I would remark that in the case of the passenger car piston engine, the power delivered at the rear end of an automatic transmission is often a long way below dynamometer horsepower. The simple treatment is mostly understandable by the older piston engine engineer and is sufficiently accurate to give him a reasonably sound understanding of gas turbine processes. The more elaborate procedure with allowance for all losses is sufficiently complex so that it confuses the older piston engine engineer in addition to being beyond my understanding. Had it been within my understanding I could not have spent six months in working out the case for a single engine. I may say that comments on the present paper by experienced turbine engineers have shown that the formal treatment of turbine processes often fails to penetrate the understanding of many very well-educated engineers of middle age and beyond.

I hope that one of these days, the actual road operating parameters for an advanced turbine passenger car will be published along with comparable data for an equally advanced piston engine car.

Dr. Barnard poses a very sound question in "Why a gas turbine for a passenger car?" In the light of present knowledge, the disadvantages are numerous and obvious. There are certain advantages that are possible or potential; First, as Dr. Barnard says, it will be lighter than the gasoline engine; secondly, it will be a better cold weather engine. It also offers potential possibilities of having better exhaust characteristics than the gasoline engine. It is also possible that engine plus transmission will be less complex than a gasoline engine plus automatic transmission.

As regards acceptable fuel economy, my opinion is that this will be slightly worse in miles per gallon than current models with cost per mile essentially equal. In other words, turbine fuel should be slightly cheaper than motor gasoline.

As regards fuel characteristics, I agree with Dr. Barnard that cracked products will have to be used if passenger car turbines come into extensive use. I still think that an end point of 550 to 600 F will be desirable for smokeless and odorless exhaust in a passenger car with the inevitable stop and go operation.

As regards the fact that my figures do not allow for losses in computing performance, I think Mr. Amann's discussion is the best answer to Dr. Barnard's query.

The Continental Mark II

—A Design Story—

H. F. Copp, Ford Motor Co.

This paper was presented at the SAE Annual Meeting, Detroit, Jan. 13, 1956.

IN June, 1952, our product planning and executive committees recommended the creation of a new car for the "prestige" market—a new Continental, embodying the aesthetic appeal and the dignified elegance of its predecessor, the Lincoln-Continental of the early Forties. In accordance with their recommendation, a new and separate organization was established to design, manufacture, and merchandise the new car. This paper tells the story of the first of these steps—the design of the Continental Mark II (Fig. 1).

Fig. 1—Lincoln-Continental Mark II

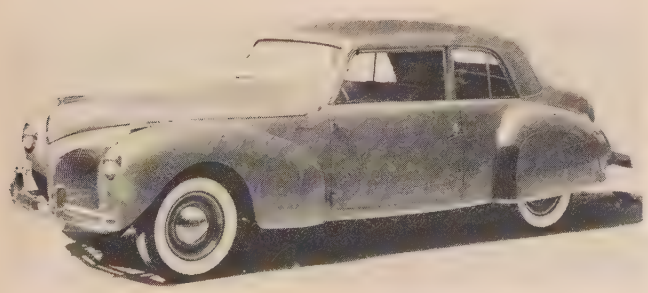


Fig. 2—Lincoln-Continental of 1941

At the very beginning of this program, four major design objectives were set forth:

1. The new Continental was to have distinctive styling. The styling was to be based upon clean, classic lines, rather than modernistic innovations.
2. The new car was to include technological advancements and special features which would enhance its value to the customer in terms of comfort, durability, safety, performance, and prestige.
3. The overall height of the car was not to exceed 58 in.
4. There was to be maximum interchangeability of chassis parts with Lincoln, including engine, transmission, rear axle, and suspension.

Preliminary studies indicated that distinctive styling could best be achieved by adhering to the fundamental proportions of the earlier Continental, shown in Fig. 2. Since these proportions were "classic,"

STYLING the Continental Mark II with a very low silhouette without sacrificing leg room, seating comfort, and driver visibility was accomplished by using a specially designed frame and a 3-joint driveline. These two features are the basic engineering innovations which make possible the Continental's distinctive classic styling. As a result of this structural analysis, Ford has produced an automobile that, according to the author, has technical advancements that are of value to the consumer.

Fig. 3—Conventional frame structure showing ordinary seating conditions

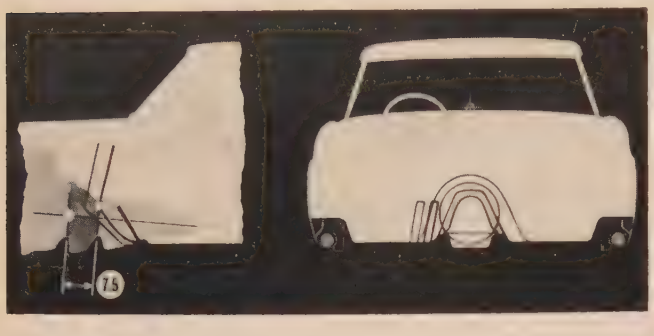


Fig. 4—Removal of frame structural members from foot areas resulting in acceptable seating standards in a lower car

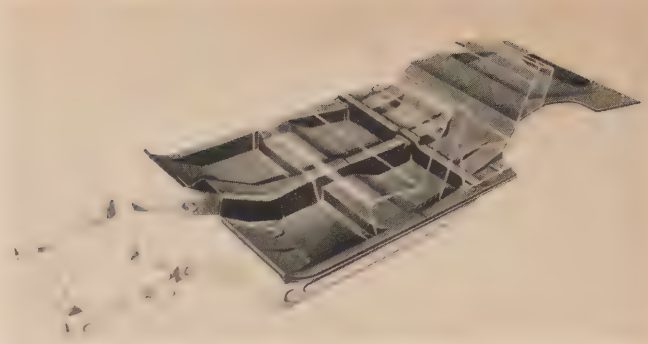


Fig. 5—"Ladder" type frame permitting deep foot-wells in floor pan

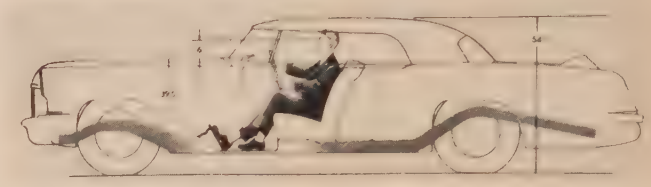


Fig. 6—Effect of 54-in. height on forward visibility and styling proportions

rather than "modernistic," it became apparent that our aim should be to create a "modern formal" design, as opposed to the informal and more spectacular "futuristic" design, typified by such cars as the X-100 and XL-500. Even the use of the Mark II designation (based on the assumption that the Lincoln-Continental produced from 1940 to 1948 constituted the Mark I series) implied that the design must be an enduring one, and that it should not be subject to yearly model changes. According to our interpretation, the "modern formal" design was characterized by a long hood, a short greenhouse, and low overall height.

Seating Package Development

In designing a car with a low overall height, the main difficulty lies in providing a comfortable seating arrangement. Conventional frame structure, as in Fig. 3, prevents lowering the floor pan appreciably, and chair-type posture is sacrificed if the seats are merely lowered with respect to the floor. A different kind of frame structure was clearly indicated, if seating standards established for the new Continental were to be achieved.

With an overall height of less than 58 in., it is desirable to remove the frame structural members from the foot areas, as in Fig. 4, so that the floor pan may be lowered to provide an acceptable floor-to-cushion height. Even if this is done, however, a new problem arises in obtaining sufficient width in the front-seat foot-wells. This is because of the increased width of the tunnel over the torque-converter housing as the lowered floor approaches the full diameter of the housing. An acceptable arrangement of accelerator and brake controls is not possible with the full "step-down" design unless the converter housing is ahead of the toeboard.

Fortunately, the long hood styling objective afforded an opportunity to explore a new type of frame-underbody construction which would also satisfy the other "modern formal" styling aims of low overall height and a short greenhouse. With a long hood, the front-seat passengers were moved far enough to the rear to allow placement of the toeboard and dash behind the large diameter of the converter housing. Then, when the frame side rails were shifted outward to a position immediately adjacent to the rocker panels, the floor pan could be lowered to the limit dictated by reasonable ground clearance. This resulted in a "ladder" type of frame, in which the members were arranged to permit deep foot-wells in the floor pan, as in Fig. 5. In addition, this design afforded a more compact passenger compartment, helping us to fulfill the short greenhouse objective.

With this type of frame-underbody construction, it became possible to build a much lower and more comfortable car than had at first been antcipated. In fact, if we had taken maximum advantage of this construction it would have been possible to design a car having an overall height of 54 in., but this was not feasible. The height of the hood was already fixed at 39.5 in. (determined by the minimum ground clearance beneath the engine, the height of the engine, and the minimum air cleaner-to-hood clearance). Good forward visibility dictated that the driver's eye-level should be at least 8 in. above the hood level. Also, from the styling viewpoint, a proper

proportion must be maintained between the overall height and the hood height.

Fig. 6 shows how forward visibility and styling proportions would have been affected by a 54-in. height. Although it is not readily apparent, the seat thickness would also be reduced in order to maintain sufficient headroom. The overall height of the new Continental was finally established at 56 in. by the addition of headroom and roof thickness to the 8-in. dimension over the hood.

Fig. 8—Structural plastic body model, ⅜-scale

Styling

Our own detail styling studies were supplemented by those of four independent consulting stylists. The consultants were requested to submit from one to three designs. Control was exercised to insure the submission of these designs in uniform color, on uniform perspective grids, and in accordance with certain critical chassis-body design dimensions, but complete freedom was permitted in hood length, greenhouse length, and design theme.

Early in 1953, our own styling organization and the four consulting stylists together submitted a total of 13 individual designs. These designs, exhibited in coded form, were reviewed individually by the members of a management committee, and each member was asked to select the design which he considered best for development into a final design. One of the three designs submitted by our own stylists was unanimously chosen by the committee for development in full-size clay. Another design was accepted because it complemented the modern styling of other Ford passenger cars, and yet retained the extremely important characteristics of formality and tradition.

The full-size clay model was approved, subject to minor revisions, in June, 1953 (Fig. 7). While it was being completed, a ⅜-scale structural plastic model was built, as shown in Fig. 8, utilizing the relatively new process of vacuum-forming thermoplastic sheet over plaster and wood models. In addition to providing a means for early evaluation of body structural

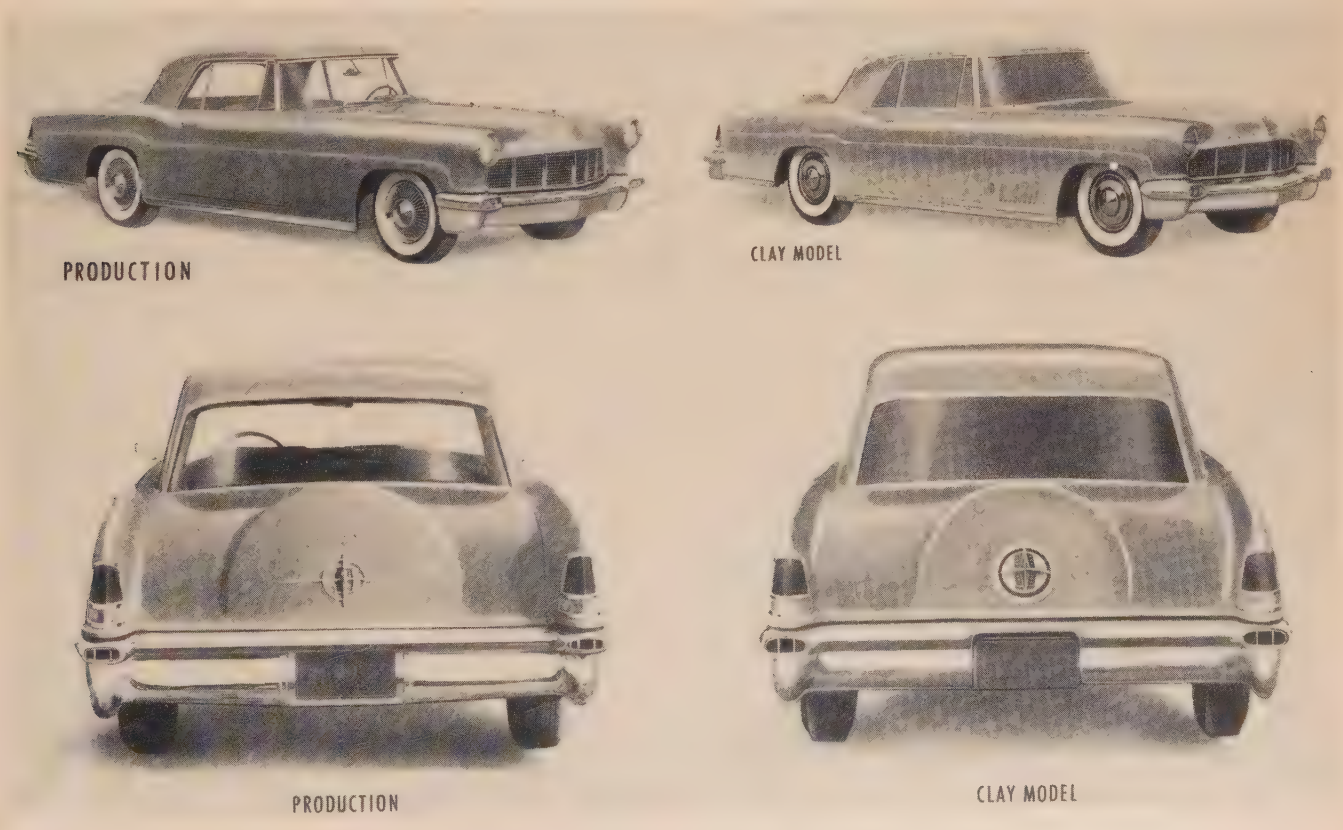


Fig. 7—Clay model (as approved in June, 1953) compared with production Continental

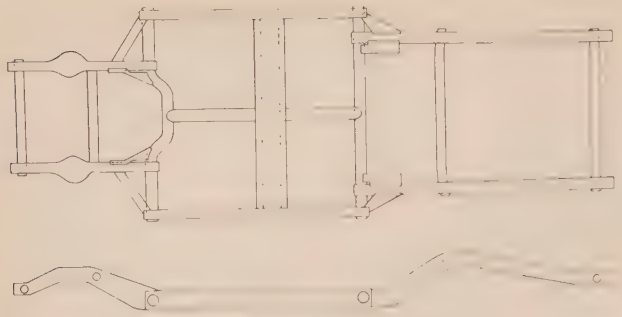


Fig. 9—First full-size experimental frame

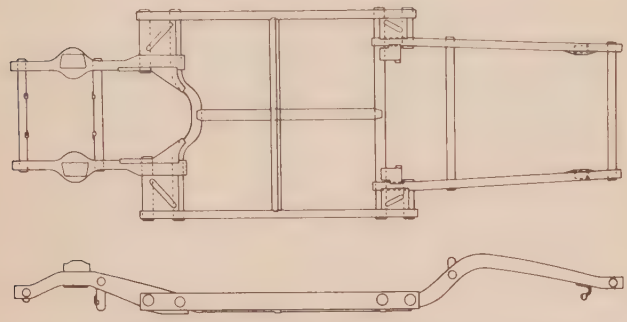


Fig. 10—Second frame design showing extended central section side rails at front and rear

problems, the plastic model served as a planning aid to the manufacturing and purchasing departments.

Frame Structure

As previously stated, the seating package dimensions in the new car ruled out the use of the conventional X-member brace in the frame. In addition, continuous side rails were not feasible because of the necessity for 16-in. right-angle offsets behind the front wheels, where the side rails must move outward to allow maximum foot-well space in the front seat floor. These limitations created a challenging problem in the attainment of adequate torsional rigidity.

Extensive analyses were made of the structural possibilities of this new "double-ladder" type of frame. Tenth-size drawings were used to study general arrangements and for structural analyses, and 3/8-scale plastic models were made of some of the designs. The plastic models made it easy to study new shapes and joints because they could be made in less than a week, and they were relatively inexpensive.

The first full-size experimental frame (Fig. 9) was built in three sections. The front section comprised double-flanged box-section side members, joined by two tubular cross-members. The front section was attached to the central section by a tubular cross-member which passed through the side members of both sections. At its center, this third cross-member was offset to the rear to make room for the transmission housing. Tubular braces angled forward from the ends of the center-section side rails to the side rails of the front section, and gussets reinforced the joints between the front-section inner rails and the third cross-member.

Like the front section, the central- and rear-section side members were box sections, joined by tubular cross-members, but the wider central section was further reinforced by a tubular center member, or "backbone," connecting the third, fourth, and fifth cross-members. The offset between the central and rear sections, although not so severe as the front offset, was heavily reinforced with channel section gussets. This was done because the fifth cross-member had to take a large measure of the wheel reaction through the rear-spring front hangers, which were to be mounted on the inside gussets.

Test results on the first frame were very encouraging. Torsional rigidity was more than twice that of the average 1952 Lincoln hardtop frame, and bending deflection was about 24% less. At 594 lb, this was the heaviest of the experimental frames to be built, yet a front end "quick-twist" test showed that it had a torsional rigidity-to-weight ratio of 8.29 lb-ft per deg per lb—unusually high for any frame without an X-member. In the front end "quick-twist" test, the frame is secured at the rear-spring front hangers. A support is placed under one of the front spring towers, and a weight of approximately 200 lb is suspended from the unsupported front spring tower. The deflection is measured at each front corner in this way and gives us a reliable index of frame rigidity in a relatively short time.

As we suspected, the most critical areas of this frame were at the connections between the discontinuous side rails and, in the second frame design (shown in Fig. 10) we extended the central section side rails at front and rear and joined them to the front and rear sections with short tubular members to form double trunnion connections. The small rectangular spaces thus formed were enclosed by capping plates at top and bottom. This construction resulted in a substantial reduction in bending deflection, as well as increased torsional rigidity, in these critical areas. However, the overall strength of this frame was somewhat less than that of the first design

Table 1—Comparison of Frames as to Torsional Rigidity and Gross Weight

	Maximum Torsional Rigidity, lb-ft/deg	Gross Frame Weight, lb	Torsional Efficiency, lb-ft/deg/lb
Experimental Frame No. 1	3189	594	5.36
Experimental Frame No. 2	2466	436	5.65
Production Frame	3013	535	5.63
1956 Lincoln Hardtop Frame	1995	358	5.57

because of reduced metal thickness in all principal members.

While the second frame design was being tested, some minor changes were made in seating package dimensions. The "A" point of the rear-seat cushion was raised slightly, permitting us to employ continuous side rails for the central and rear sections of the frame. At the same time, the engine was moved back 1.5 in., due to a relocation of the passengers. This meant a deeper offset in the third cross-member. Since this was impractical, a new construction was indicated.

In the third frame design (shown in Fig. 11) this cross-member and the longitudinal center tube were replaced by a box-section Y-member. On each side, the connection between the front- and center-section side rails was changed from the double trunnion to a larger diameter tube, which passed through the front side rail and was joined to one branch of the Y-member. As before, gussets were used to blend these joints smoothly together, and channel sections were welded to the tube between the center and front side rails. A capped, inverted hat section formed the fourth cross-member in place of the tubular member of the second design. Except for its smaller size, this member was similar to the one employed in the first frame.

Torsional and bending rigidity tests on this third frame design (Fig. 12) supported our original choice of an enclosed rectangular section for the fourth cross-member, which proved to be quite effective in torsion, but which had little value as a bending member. The new Y-member, on the other hand, contributed effectively to both bending and torsional rigidity because of the more efficient juncture between the discontinuous side rails at the dash. In bending, this front connection works as a torsion member and, through the Y-member, reduces the bending moment at the center of the fifth cross-member.

During all this time, we had been aware that one of the most critical problems facing us was the control of front-end shake, due to the combination of an unusually long hood and the relatively inefficient stress transference characteristics of the wide offsets between the front and central frame sections. We reasoned that control of front-end shake could best

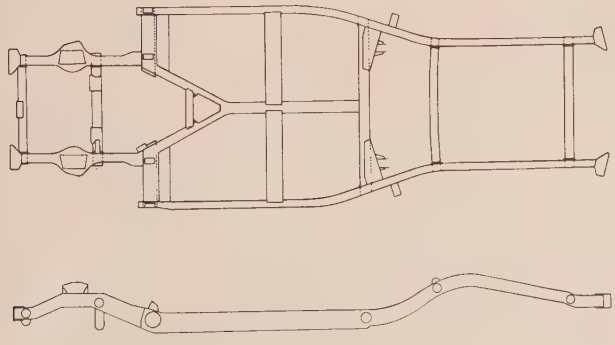


Fig. 11—Third experimental frame using box-section Y-member

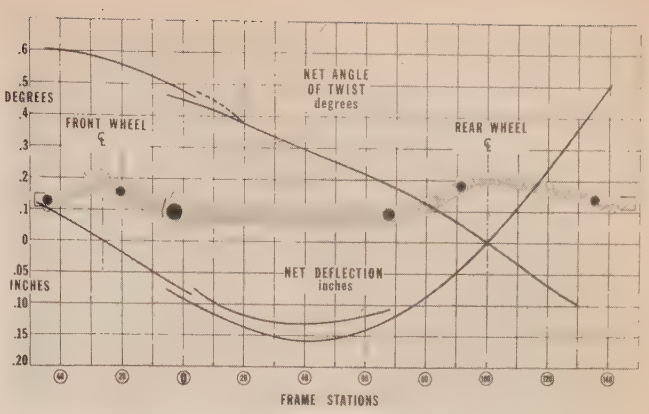


Fig. 12—Torsional and bending rigidity data from tests of frame using Y-member design

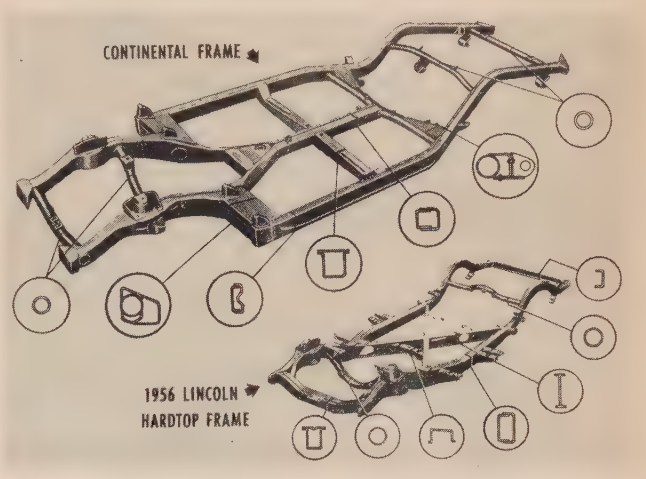


Fig. 13—Production Continental frame assembly (upper) compared to 1956 Lincoln hardtop frame (lower)

be obtained by providing extremely high torsional rigidity in the frame between the front of dash and the centerline of front wheels. As the frame design progressed through its various stages, we were gratified to see that torsional rigidity in this critical area was constantly increasing (from 13,430 lb-ft per deg in the first experimental frame to an average of 17,667 lb-ft per deg in the production frames).

With a few minor revisions, this design became the production Continental frame. In Table 1, it is compared with the 1956 Lincoln hardtop frame as to torsional rigidity and gross weight.

Although the weight of the production frame is rather high, its efficiency, in terms of the torsional rigidity-to-weight ratio, is much greater than we had expected to achieve in a ladder-type frame. In any case, we were convinced that the degree of torsional rigidity obtained is largely responsible for the excellent stability of the new Continental. We are also confident that the extra torsional stiffness designed into the frame between the dash and the centerline of front wheels was the controlling factor in reducing front-end shake, since it is practically nonexistent in the production car.

In Fig. 13, the production Continental frame as-

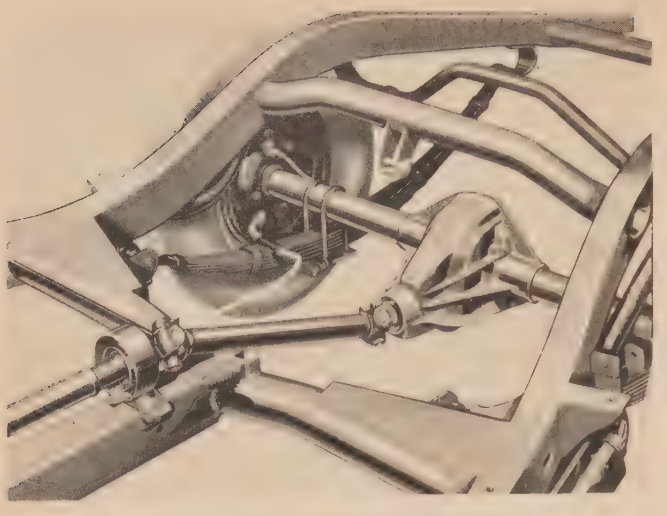


Fig. 14—Rear portion of driveline

Fig. 15—Use of 2-joint shaft would require fifth cross-member to be reshaped to accommodate shaft during rebound. As shown above, this would reduce rear-seat foot room

sembly is illustrated, along with an inset of the 1956 Lincoln hardtop frame for comparison. Overall static rigidity tests of the assembled frame and body show that the Continental frame contributes almost 50% of the total rigidity in both bending and torsion. The contribution of the 1954 Lincoln frame, on the other hand, amounts to only about 30%.

Driveline

In order to provide the lowest possible tunnel height, a 3-joint driveline was used in the new Continental. The center universal contains the slip-joint, which allows fore-and-aft movement of the rear shaft. Just ahead of the center joint, a rubber-lined mounting for the front shaft is bolted to the central Y-member of the frame (Fig. 14). The shaft turns in a single-row ball bearing which is supported in this mounting. The special design of the new frame lends itself admirably to the employment of a 3-joint driveline, since it allows the shaft to be supported at its most critical point by a vital frame structural member.

This arrangement afforded a gain of 1.25 in. in the thickness of the rear-seat cushion at its "A" point above the tunnel, in comparison to the dimension obtained with a conventional 2-joint shaft installation. At the same point on the front-seat cushion, the gain amounted to 0.6 in. The reduction in tunnel height at the front edge of the rear cushion was 2.08 in.

Had a 2-joint shaft been used, the fifth cross-member in the frame would have had to be reshaped to accommodate the shaft under rebound conditions, as shown in Fig. 15. This would have seriously reduced rear-seat foot room, as the illustration shows. Further, since the Continental was to be a low-slung car with top-speed capabilities in excess of 100 mph, the critical speed of a long shaft with two joints would probably be within the upper speed range of the car —an undesirable, if not intolerable, feature.

Once the decision was made in favor of a 3-joint driveline, we began experimenting to find the best and most workable arrangement of elements. Most of the dimensions governing the driveline geometry, such as frame height, ground clearance, and tunnel height and width were already fixed. The chief variables left for us to work with were the height of the rear engine mount and the rear-axle-pinion nose angle, which could be varied between jounce clearance to the tunnel and rebound clearance between the shaft and the fifth cross-member—a total of 5 deg. (See Fig. 16.)

We first approached the problem from the standpoints of the pinion nose angle and the design of the center universal mount. Our test car was a mechanical prototype, and the procedure was simple and straightforward. We tried nearly every angular change within the imposed limits, making minute changes in the garage, then going out on the road to test results. However, there seemed to be no angle that would eliminate rumble, which occurred during wide-open throttle acceleration at speeds of 13–18 and 25–30 mph.

Several kinds of center universal mounting were evaluated, along with mounts of varying hardness. As it turned out, the nature of the mounting was not nearly as important as the selection of a rubber having the right natural frequency. To avoid resonance with large body panels, we chose the highest possible frequency that was consistent with good vibration-damping efficiency.

In exasperation, we removed the rear engine mount from the car, and with a hacksaw, we cut off about three-quarters of an inch. When it was put back into the car, the low-end shudder had disappeared. Needless to say, we followed up this more or less accidental experiment with a complete series of tests in which the rear engine mount height was varied in $\frac{1}{16}$-in. increments. Surprisingly, however, the first angle turned out to be the one angle at which the offensive vibratory forces at the center bearing were practically eliminated. This was later verified when the engineer-

ing research department made an analysis of the angles we had arrived at experimentally. Their check showed that the vibratory load was indeed at its minimum.

A rudimentary investigation of the effects of phasing the front shaft yokes had indicated no benefit to be derived. But, when an engineering research department analysis showed that proper phasing could reduce the vibratory load at the center bearing even below the level we considered acceptable, we checked the eight spline positions in combination with the other variables. With possible variations of 5 deg at the pinion nose and ⅜ in. at the first joint, we could have tested 240 combinations. By judicious selection, however, we narrowed the choice down to a number that could be tested before the car was put into production. Results of these tests in our mechanical prototype car showed that driveline vibrations were reduced to a completely acceptable minimum when the front-shaft yokes were phased at 90 deg.

About this time, our first engineering prototype became available. In contrast to the "cut down" 1953 Lincoln body on the mechanical prototype, this car incorporated one of the first Continental bodies, with different body-panel resonance characteristics. Much to our disappointment, we found that the driveline geometry we had so painstakingly worked out was unsatisfactory for the engineering prototype. Resonant periods we had previously considered negligible were accentuated, and an annoying vibration of 58 mph compelled us to begin our testing all over again.

In the tests that followed, we were finally forced to depart from the analytical approach developed by the research department when it was found to be applicable only to center bearing vibrations at low speeds. We hope there will soon be a mathematical solution to the entire problem of driveline vibrations —one which does not require the use of constant-velocity joints.

After repeating many of our first tests, we finally arrived at a completely acceptable arrangement of driveline components. The pinion nose angle was established at 2 deg below horizontal at curb height, the rear engine mount height was set at $\frac{3}{16}$ in. below its original point, and the front shaft yokes were phased at 0 deg, or line to line. The height of the rear engine mount was found to be the most critical dimension and, for production, a special gaging fixture was worked out to insure its close control. This fixture (shown in Fig. 17) contacts the frame assembly at the positions of the front engine mounts and the center universal mount and indicates the relative height of the frame at the rear engine mount. Any discrepancy is recorded and, when the engine is installed, shims are added between the transmission extension and the rear engine support plate, as shown in the inset. The angle of the pinion nose is also closely controlled in production by the use of shims.

At this point, it may seem that we were preoccupied with vibration problems. To a certain extent that is true, but only because the overall rigidity of this car, in conjunction with the high degree of insulation around the body, made it exceptionally quiet in operation. This had the effect of emphasizing any periodic vibrations out of proportion to their actual amplitude, with the result that special attention had to be given to even the smallest tremors.

Ride Control

Early in the design program, it was established that the new Continental must have an unusual combina-

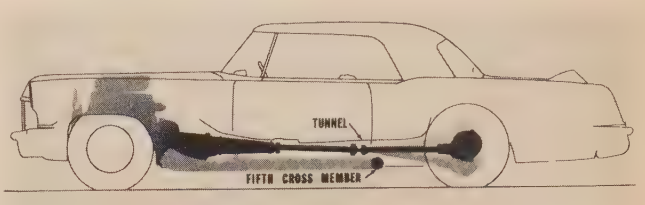


Fig. 16—Driveline design limitation

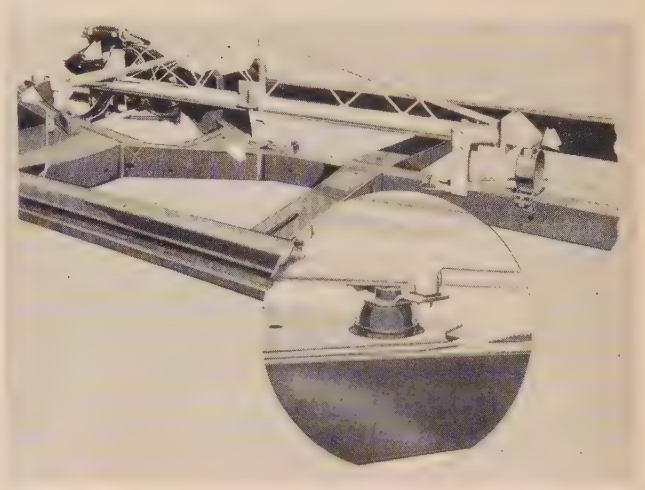


Fig. 17—Gaging fixture used during production to insure close control of height of rear engine mount

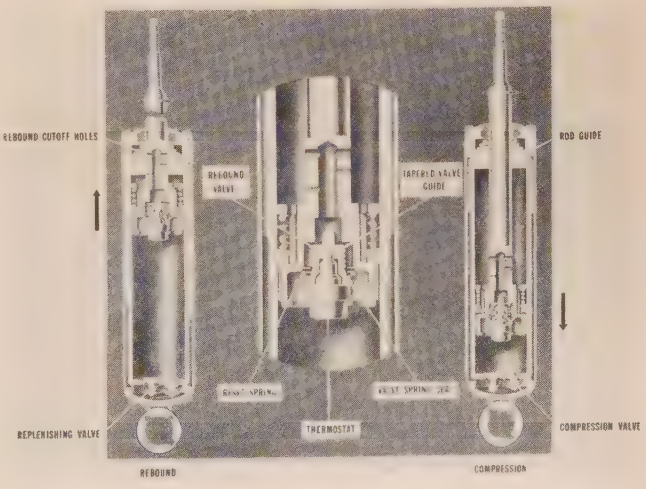


Fig. 18—New fully automatic, speed-compensating shock absorber

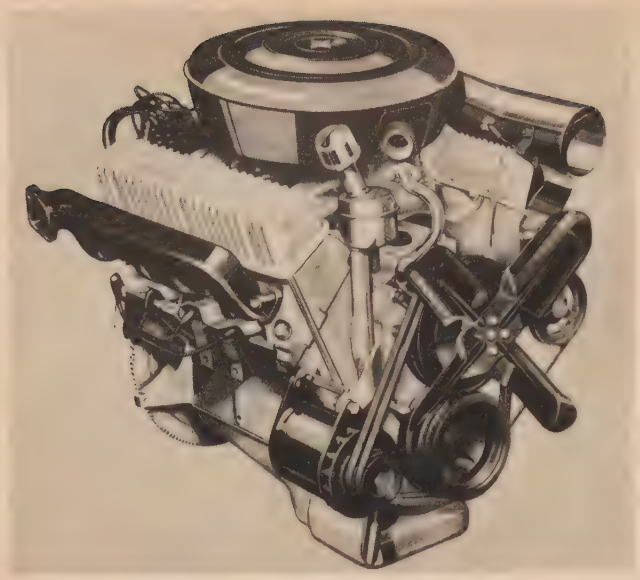


Fig. 19—Modified 1956 Lincoln engine which has been adapted for Continental

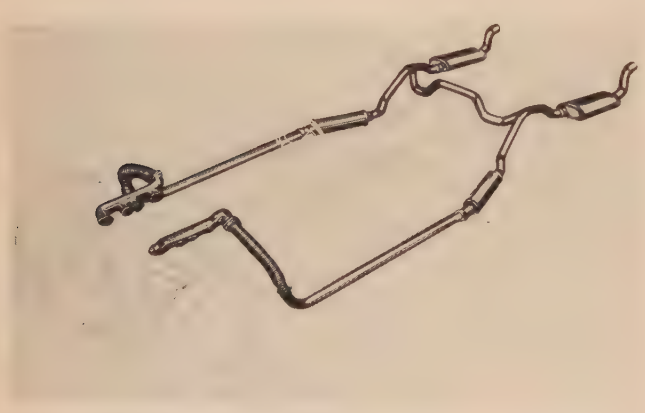


Fig. 20—Arrangement of dual exhaust system in new Continental

tion of riding qualities. For maximum comfort in city driving, a "boulevard ride" is required, while safety and stability at high speeds demand a firmer ride control. Ultimately, these conflicting requirements led to the invention and development of a fully automatic, speed-compensating shock absorber (illustrated in Fig. 18).

Development of this shock absorber was based upon our belief that the best results would be obtained through the use of a small orifice with a wide range of control provided by the valve spring load. As a consequence, the clearance around the valve guide pin became the orifice. A thermostatically controlled rebound valve provides constant shock-absorbing action with negative fade characteristics by controlling the tension on the valve spring as the operating temperature varies. The thermostat is a pellet type, containing a wax which is impregnated with a metallic salt. Since it is fully operative within a few degrees, it will adjust quickly to temperature increases caused either by the rapid, short-stroke operation associated with high-speed driving, or the more violent long-stroke operation when driving over extremely rough roads. Positive readjustment to the cold setting is assured by the inclusion of a reset spring. Exhaustive road testing proved that this speed-compensating feature was required only on the front wheels.

Both front and rear shock absorbers incorporate a device which limits rebound travel. As the shock absorber nears the rebound limit, two large fluid escape ports in the piston rod are cut off by the closely fitted rod guide. Further movement cuts off a medium-sized port and, finally, a small one. This action provides a progressively increasing resistance until the smallest hole is closed, resulting in a condition of hydraulic "lock" which effectively prevents further movement. The severity of rebound "strike-through" is thus reduced, improving the ride on rough roads, and eliminating the need for rebound bumpers.

The relocation of the greenhouse toward the rear also made it necessary to shift the attachment point of the rear-spring front eye rearward in order to maintain the package advantages originally planned. This was accomplished by changing the relative lengths of the front and rear sections of the spring, making the front section 6 in. shorter than the rear section. An incidental advantage of this arrangement is that it reduces the effect of windup torque because of the increased stiffness of the shorter front section, thus helping maintain proper alignment of the axle universal joint under high-input torque loads. Five full-length liners are used in each rear spring. These are made of a "beater-mixed" plastic paper compound which has a very high resistance to water absorption, insuring quiet operation in winter as well as summer. Threaded metal bushings are used in the rear-spring shackles since tests proved that, in the Continental, they contributed to controllability and tended to reduce "mushiness," especially at high speeds.

Engine

A modified 1956 Lincoln engine is used in the new Continental. The engine is pictured in Fig. 19. Because of the "step-down" frame design and the low hood, it became necessary to deviate from the engine position established for the 1956 Lincoln. The mounting angle was reduced from 4 deg 30 min to 3 deg 18 min, and the engine was lowered 1.5 in. This made a special oil pan necessary in order to provide the required volume of oil for extremely steep grades, as well as to reduce the overall height of the engine. To create a distinctive appearance, finned, cast-aluminum rocker covers are used.

Exhaust System

The arrangement of the dual exhaust system in the new Continental is illustrated in Figs. 20 and 21. Because the floor structure occupies all the space between the frame side rails, it was necessary to route

the muffler inlet pipes around the outside of the frame. In order to provide clearance between the body rocker panels and the frame, the side rails have indentations along their lengths in the central section. Small resonators are located ahead of each rear wheel and, to provide additional resonator capacity in the small amount of space available, a crossover pipe connects the two systems just ahead of the mufflers. The crossover pipe functions somewhat like an accumulator in equalizing the shock waves of alternate exhaust impulses, and helps to reduce the noise level of the exhaust system at idle and low speeds.

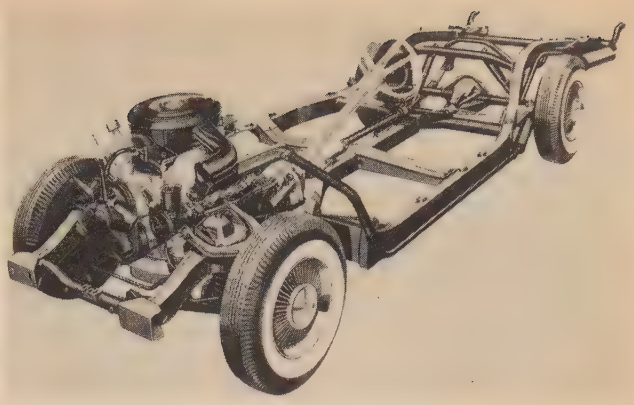


Fig. 21—Continental Mark II chassis

Body Structure

The Continental body structure contains many interesting features which grew out of its unconventional styling proportions. To begin with, a new approach was taken to windshield pillar construction. By locating the pillar totally inboard of the windshield glass, and by devising a new method for retaining and sealing the glass, we were able to design a pillar of extremely narrow section. As shown in Fig. 22, the windshield pillar is made from two heavy-gage stampings, arc-welded along their entire length to form a closed section structure. The use of heavier gage metal and the inboard pillar arrangement resulted in a windshield pillar and vent window frame whose combined cross-section, including all moldings and seals, can be enclosed within a very small circle. In fact, the width of the pillar is no greater than the combined width of the seals.

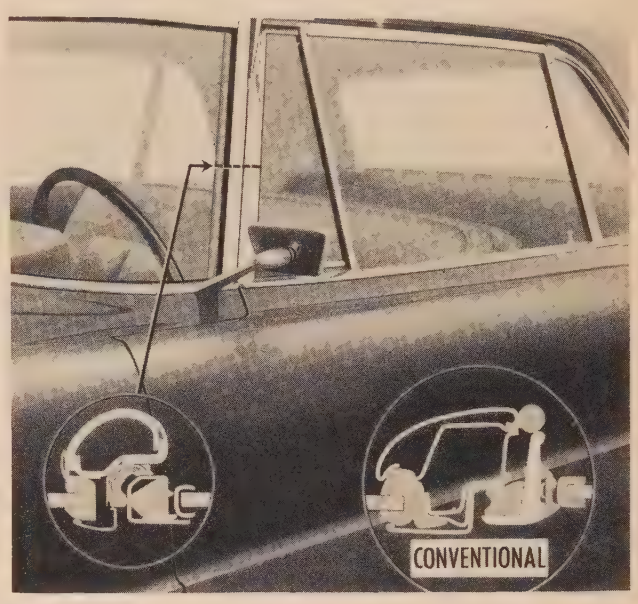


Fig. 22—New approach to windshield pillar construction (shown in insert at left)

At the top, the pillar is welded to both upper and lower surfaces of the roof side rails. At the bottom, the pillar structure flows underneath the upper cowl panel into the front body hinge pillar. This structure permitted us to carry the windshield glass around to the pillars in one continuous sweep, unbroken by corner radii. Also, we joined the windshield glass to the pillars in the same plane with the side windows, presenting a flush surface to the air flowing around the corners. The windshield glass is retained by means of conventional weld flanges along the upper and lower edges. The ends, at the pillars, are allowed to float in the rubber seals which are retained by the finish moldings.

To add to the overall rigidity of the windshield pillar structure, and to minimize the possibility of squeaks, the instrument panel was designed as a structural member and is welded in place. Further rigidity is gained in the cowl area by the triangular truss which is formed by a structural member which connects the hinge pillar to the cowl and to the forward end of the outer rocker panel.

In order to obtain the maximum width of foot wells in the floor pan, we again chose an unconventional approach. A compact grouping of the chassis frame side rails, muffler inlet pipes, and body rocker panels provided an appreciable gain in this area. As shown in Fig. 23, the body inner rocker panel is welded to the

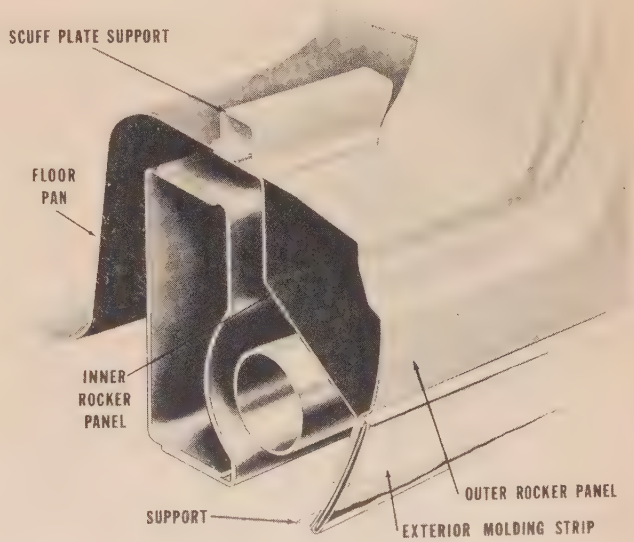


Fig. 23—Cross-section showing body inner rocker panel welded to underbody above frame side rail

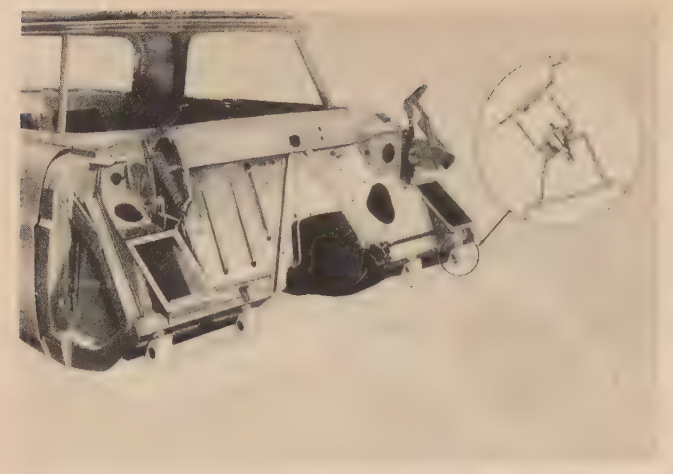


Fig. 24—Attachment of body to chassis frame at dash

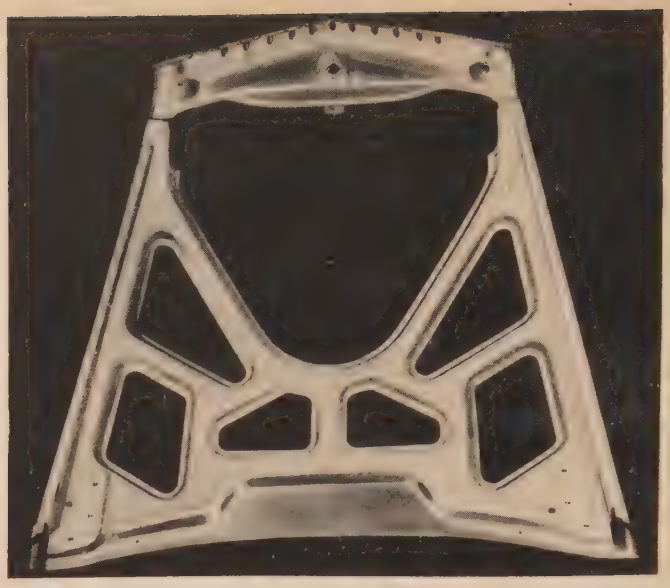


Fig. 25—Production hood assembly

underbody above the frame side rail. The lower end of the inner pannel is welded to the outer rocker panel, forming a closed section whose shape permits the rocker panel to be placed very close to the muffler inlet pipe. The frame outer side rail has an indentation along its length from the dash to the front face of the wheelhouse, and the muffler inlet pipe occupies the space thus made available. Both inner and outer rocker panels are made of corrosion-resistant, copper-bearing steel, and the inner surfaces of the box section are bonderized and painted as extra insurance against corrosion. The exterior molding strips have inboard-attached supports which are removable to permit servicing the exhaust system.

Fig. 24 shows the method used to attach the body to the chassis frame at the dash. A structural support, extending from the body rocker panel inward to the driveline tunnel, is welded to the toe panel at the dash intersection. Attachment is made to four heavy-gage stools welded to the chassis frame, a construction which replaces the conventional dash brace.

An extra-rigid hood assembly was essential because of its great length and the flatness of the hood panel surface. The production hood assembly (Fig. 25) weighs 90 lb and comprises the hood panel with side reinforcements and an inner panel structure, welded to the hood. A sound-deadening material is cemented to the underside of the hood panel before the inner structure is welded to it. The high degree of structural rigidity built into the hood and the front-end sheet metal to which it latches is undoubtedly responsible for the comment which has been made to the effect that the closing of the hood is reminiscent of the closing of a vault door.

Interior Design

One of our prime objectives in interior design was to give the driver the feeling that he "belonged" to the

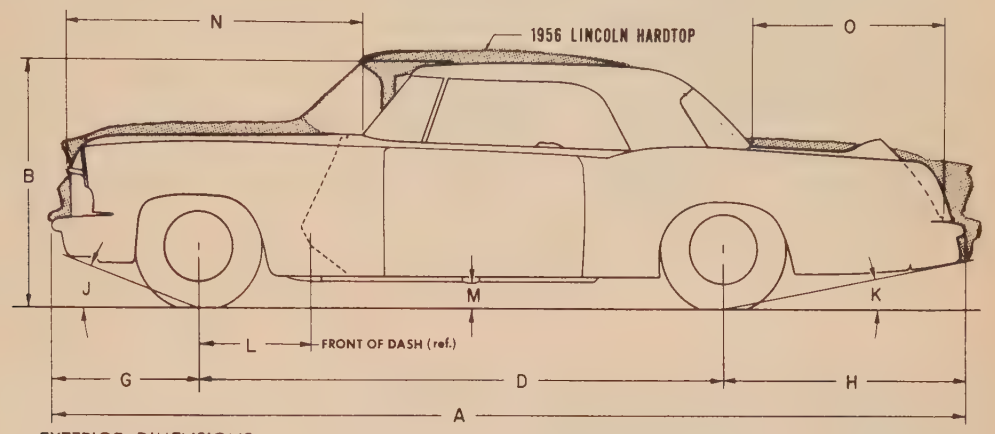
Fig. 26—Instrument panel of new Continental

car—that he and the car were an entity. To achieve this, we placed the windshield close to the driver, allowing wide angles of visibility in both horizontal and vertical planes. This feeling of intimacy is further enhanced by the placement of controls and instruments. A low steering column angle puts the wheel in a near-vertical position and, to insure the maximum distance between the rim of the wheel and the driver's eye level, a 17-in. wheel (1 in. smaller than the Lincoln steering wheel) is used.

Instruments are grouped in a horizontal row of four circular dials, directly in front of the driver (Fig. 26). It is our belief that the round dials have a clean, mechanical appearance that lends itself admirably to the functional styling theme we endeavored to maintain throughout the car. In addition to the usual instruments, a tachometer is included, since we found that it was one of the most often requested additions to the instrument panel of a car such as the Continental. Another convenience is the warning arrow which points to the fuel gage. The arrow glows red whenever the quantity of fuel in the tank drops below approximately three gal. The instruments are cov-

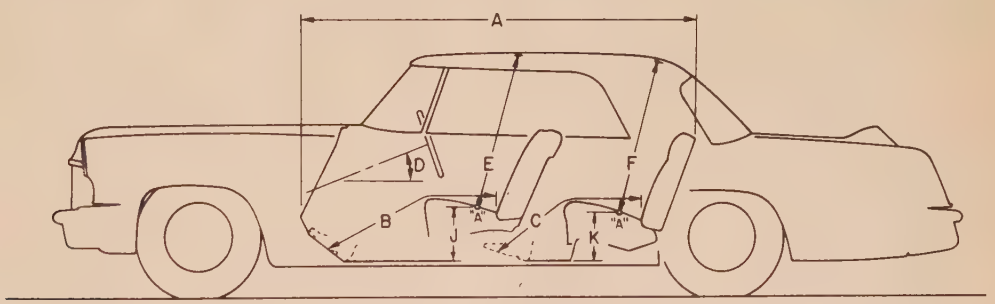
ered by a leather-trimmed die-cast hood, designed to prevent instrument light reflections in the windshield at night. The hood is easily removable to permit instrument service or light bulb replacement.

Seat cushions in the Continental are of full foam rubber construction and are mounted on a flat spring frame. The full foam rubber provides a better shock-insulating effect and permitted us to design a firmer cushion. Depression of the front-seat cushion at the "A" point by a person of average weight amounts to only 4 in., as compared to 5.1 in. in the 1956 Lincoln hardtop. Rear-seat cushion depressions at the



EXTERIOR DIMENSIONS	Continental Mark II	1956 Lincoln Hardtop
A—Overall Length	218.5	222.8
B—Overall Height	56.0	59.6
C—Overall Width	77.5	79.9
D—Wheelbase	126.0	126.0
E—Front Tread	58.5	58.5
F—Rear Tread	60.0	60.0
G—Front Overhang	35.2	35.0
H—Rear Overhang	57.2	61.9
J—Angle of Approach	19° 3′	20° 15′
K—Angle of Departure	11° 30′	11° 15′
L—Front of Dash to ℄ of Front Wheels	26.0	18.8
M—Minimum Ground Clearance	5.9	6.8
N—Hood Length	70.8	55.9
O—Deck Length	45.4	50.1

Fig. 27—Comparison of dimensions of 1956 Lincoln hardtop and new Continental



INTERIOR DIMENSIONS	Continental Mark II	1956 Lincoln Hardtop
A—Interior Length	94.6	96.1
B—Front Leg Room	42.5	44.8
C—Rear Leg Room	39.1	40.0
D—Steering Column Angle	20° 35′	27° 7′
E—Front Head Room	35.1	34.8
F—Rear Head Room	34.6	33.0
G—Front Hip Room	59.6	62.4
H—Rear Hip Room	56.6	53.0
J—Front Seat Height at "A" Point	11.7	9.9
K—Rear Seat Height at "A" Point	11.5	10.7
Curb Weight	5000	4633*

*Premiere Series, with normal complement of accessories

Fig. 28—Exterior air inlets of air-conditioning system

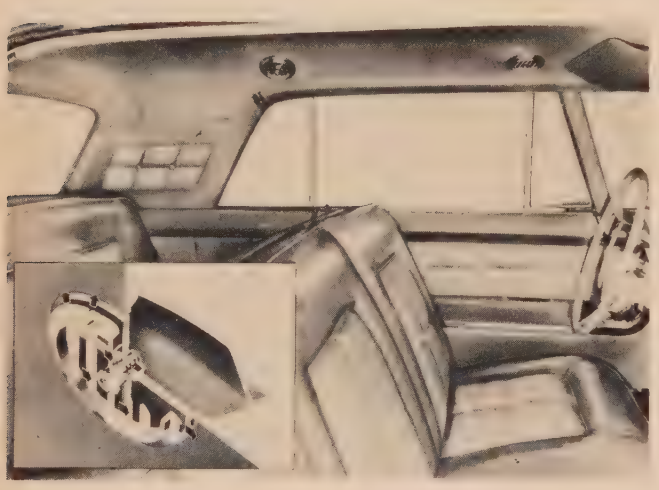


Fig. 29—Concealed interior duct system lets in cool air through four adjustable roof outlet registers

"A" point are 3.82 in. for the Continental and 4.2 in. for the Lincoln.

Referring to the dimensions shown in Fig. 27, it will be noted that leg room in the Continental was not sacrificed because of the low silhouette styling. Chair-height seating is provided in both front and rear seats by the step-down frame design, and rear-seat leg room approaches that found in the popular 4-door sedans. The accelerator and brake pedals are situated in approximately the same plane with the toe pan under the driver's left foot, so that there is equal room for both legs. This was accomplished by means of a depression in the toe panel adjacent to the driveline tunnel.

Air conditioning is currently being installed in about 75% of Continental production. In the Continental system, the condenser is located in front of the radiator, and the evaporator is at the front of the luggage compartment, underneath the package shelf. The air inlets, which are the only items visible on the outside of the car, are located in the kickup area of each quarter panel, as illustrated in Fig. 28. These inlets are brass castings, brazed to the inside of special quarter panels which replace the regular quarter panels when the car is built.

Cool air is directed through a completely concealed duct system to four adjustable roof outlet registers (Fig. 29). The amount and direction of cool air introduced into the car is controlled by rotating a vaned grill in each register, as shown in the inset in Fig. 29. Air-conditioning controls are located in a console below the radio control panel and speaker grille. They are integrated with the heating and ventilating controls, eliminating the necessity for an auxiliary panel.

The accepted yardstick for measurement of the automobile, by the industry and presumably by the public, is nowhere evident in the Continental. It was our belief that "prestige" was in no way associated with size, power, and chrome. The Continental does not have the most chrome, the most horsepower, or the greatest size. Rather, the emphasis is placed on elegance, and not only the elegance of appearance. Certainly, an essential requirement of elegance is enduring value—rhinestones are no substitute for diamonds. Not only must a "prestige" car look elegant, but it must feel elegant to the owner, and it must grow in his esteem.

It would have been easier to grasp at the accepted approach in designing the new Continental. It could have been larger than any current car, it could have been *fully* chrome plated, and it could have boasted the highest horsepower rating. Also, it would have been fun to have designed a technological laboratory on wheels.

It is obvious from what already has been said in this paper that we chose none of these approaches. The style of the car is not based on size and/or chrome. Technical innovations were included only where the design indicated that they would be of real value to the customer and would not jeopardize field service, since with a "prestige" car good field service is as important as new car quality. Those who will be interested in owning the Continental expect a degree of satisfaction beyond the ordinary.

ORAL DISCUSSION

Reported by L. H. White

Ford Motor Co.

A. K. Hannum, Thompson Products, Inc.: Why wasn't a unitized body construction used on the Continental instead of a separate frame?

Reply: This type of construction was considered and discarded for the following reasons:

1. Limited development time.
2. A convertible model was originally projected, but insufficient time was available for development of structural members on a model of this type.

ORION

A Gas-Generator Turbocompound Engine

Ralph J. Hooker, General Electric Co.

This paper was presented at the SAE National West Coast Meeting, San Francisco, Aug. 7, 1956.

IN June, 1950, we started a powerplant development program for the U.S. Army Ordnance Corps. The project was named "Orion" after the constellation Orion, the hunter of Greek mythology. This contract was with the Research and Development Branch of the Tank Automotive Center, Detroit Arsenal requiring the development of a turbogas-generator engine for application to tanks and other track vehicles.

The need for such a powerplant study was apparent when it was reported by our field forces that our tank equipment seemed inadequate in comparison to enemy equipment. Enemy equipment was generally diesel-engine driven.

The contract was written to cover various phases of the overall program, and each phase was to run consecutively and blend, one into the other. There were four basic phases.

These four phases included analytical cycle studies, single-cylinder engine design, construction and test and design of a full-size engine, and running an engine in a tracked vehicle. This last phase was being

THE "Orion" gas-generator turbocompound engine consists of a supercharged, regenerative aircooled, 2-stroke-cycle opposed-piston diesel engine driving two centrifugal compressors. One of these compressors is for combustion air with fine air filtration, while the other is for cylinder cooling with much less filtration.

The gas-generator engine has a bore of 4¼-in. diameter and a stroke of 5⅞ in. x 2. The engine turns at 2340 rpm, and the combustion air compressor turns at 37,000 rpm while the cooling air compressor turns 17,000 rpm.

The cylinder is cooled with air at nearly the supercharge level and at an equivalent temperature because this air later does work on the turbine. The cooling airflow is about 3½ times the combustion airflow. These two airstreams join in a plenum chamber downstream from the engine, and the mixture temperature is about 500 F. This hot gas stream then goes to the power turbine, which is mechanically free of the gas generator.

The combination of the engine and the two compressors forms the gas generator. The turbine is the source of power in this powerplant.

Table 1—Comparison of Orion and Gasoline Powerplant Performance

	Present Gasoline Engine	Predicted Performance of Orion[a]
Engine	AV 1790-5	Orion 1000
Cycle	4-stroke	2-stroke
Gross Power	810	600
Power (after Torque Converter), hp	540	600
Specific Fuel Consumption (Based on Power after Torque Converter), lbs per hp-hr	0.9	0.5
Volume of Engine, cu ft	75	58
Volume of Powerplant (Engine, Oil Cooler, and Fans), cu ft	85½	63
Fuel	Gasoline	Diesel fuel
Fuel Tank Capacity	100%	180%
Range (Simulated Combat), miles	100%	180%
Idling Fuel Rate, gal per hr	5.2	2.6

[a] Assuming installation in M-47 hull.

actively worked on when orders were received to cancel all outstanding work and complete the project on Dec. 15, 1955.

It was proposed by the general engineering laboratory that one of the compound engine designs might attain the dual objectives of greater power density as well as improved efficiency, even to the extent of approaching diesel-engine thermal efficiency. At the conclusion of the study phase, the regenerative-cooled cycle was adapted as best fitting all the requirements and the inferred meaning of the specifications. This cycle was essentially as follows. The opposed-piston engine drove two air compressors, one for the combustion air which was required to be finely filtered and a cooling air compressor requiring less filtration. The cooling air compressor operated at approximately the supercharge-pressure ratio, and the air was discharged at a corresponding temperature. The idea of the regeneration was that the cooling air would absorb the heat rejected from the cylinder wall, and a certain percentage of this heat is converted to work in the turbine. The two airstreams, combustion air and cooling air, joined together downstream from the engine at a common pressure and at a mixture temperature consistent with the airflow and fuel setting. The resulting hot gas stream then entered the turbine from which was derived all the useful mechanical power.

In addition, it was felt that the torque-speed characteristics of a turbine would be an advantage, leading to improved types of transmission. Therefore, the Orion powerplant must be considered more in the light of a torque converter than as a gas-turbine powerplant. This will also be more evident as the use of the "pneumatic" transmission is discussed later in the paper. In the initial stages of this development the specifications were very broad, and this allowed quite a wide variation in interpretation by the engineers of the Research and Tank Automotive Center of the Ordnance Corps.

The specifications were as follows (Horsepower originally specified was 900 hp, and later decreased to 600 hp.):

1. Turbine shaft horsepower—600.
2. Minimum life—500 hr.
3. Specific fuel consumption—approximately 0.5 lb per hp-hr.
4. Cooling—engine must be aircooled.
5. Fuel—diesel fuel oil (to include gasoline at later date).
6. Water tightness—fording rivers and submerged operations.
7. The subject powerplant was to fit into the existing engine compartment of the M-47 tank.

To compare the predicted performance of Orion with an existing gasoline powerplant, Table 1 is presented.

Guided by the specifications, a study was made of many conceivable forms of gas generator-gas turbine powerplants to obtain the maximum power density. In order to rate these various powerplant configurations, a number called the "factor of merit" was used to grade the power density of each configuration. In accordance with the dimensions of the engine compartment in the M-47 tank of 3 ft x 4 ft x 5 ft = 60 cu ft and the initial requirement of 900 turbine shaft hp with additional auxiliary power for radio, radar for gun control, and hydraulic power for turret control, this would mean a gas-generator output of about 1200 gas hp. Gas horsepower is defined as the power produced by the gas when expanding through a turbine of 100% efficiency from manifold pressure and temperature to the ambient atmospheric pressure. Therefore, for any powerplant to satisfy the power density, it would have to produce at least 20 gas hp per cu ft of engine envelope. This output was given a rating of 1.0. The comparative evaluation of several engine designs is shown in Fig. 1.

It is interesting to reflect that, five years after this chart was made it is still valid. It also points out an additional interesting fact—no powerplant using a reciprocating compressor regardless of cylinder arrangement, except the "free piston," has the necessary high "factor of merit" to satisfy the requirement of power density. The power requirements were later reduced to 600 turbine shaft hp; thus in the completed Rigel I and II engines, the power density was 9.4 turbine shaft hp per cu ft of envelope.

The results of this program, shown as test results in the curves at the end of this paper, can be compiled into a series of observations as follows:

1. An aircooled balanced-cycle compound powerplant of 600 hp can be built and operated at the specified design point.
2. From these test results the specified specific fuel consumption of 0.5 could have been improved.
3. Aircooling an opposed-piston supercharged diesel-engine cylinder is feasible. While too much piston cooling was being done, this could have been reduced by at least one of the many "hot-crown" piston designs under consideration.
4. With revised cycle conditions, a one-compressor design would have shown better performance by reducing the number of high-speed shafts and gears.

In other words, the mechanical efficiency would be higher.

5. The accumulator type of fuel-injection system can give excellent service and can be designed to satisfy any fuel-flow requirement.

6. It is feasible to use the technique of the Howard Foundry to make iron cylinders of almost any design.

7. Chrome-plated piston rings on the iron liners showed signs of being able to give long, low-wear life.

8. With proper tension on the oil-control rings and with sufficient drainage capacity this Orion engine can be made as oil-tight as any other engine.

9. The idling fuel consumption is very low, subject to the other demands of speed and power for the auxiliary equipment.

10. The engine could be easily heated with hot air for cold starting, but absolute caution would have to be taken to see that it is clean filtered air.

11. Because of its high compression ratio, large number of high-speed gears and shafts, and long piston skirts, starting power will be high. The two attached starters are of 29 hp each.

12. Test results show that our turbine specific fuel consumption was within 12% of the specified 0.50. The reason for this deficiency can be accounted for in the lack of trapped air in the cylinder, too high a pressure drop across both the cylinder ports and the cooling fins, too high heat rejection through the piston crown, poor mechanical efficiency, and compressor efficiencies below design. By normal developmental improvement in all of the above respects, the turbine specific fuel consumption would be well below the specified 0.50.

13. A reaction gas turbine will run successfully without a shroud band; otherwise choking will result due to carbon buildup under the band.

14. It would be a simple matter to convert the Rigel engines into the single-compressor design, by the judicious use of bolted-in diaphragms and deflectors.

15. A hot gas generator of this design should not have a built-in lubricating oil sump tank, because of the heat rejection to the oil.

Since this development was terminated prior to completion of the last phases, there are many incomplete details that should be pursued to a final conclusion.

All of the engines were named after stars in the Orion constellation. The first engine, Alpha, did not fit into the finally accepted cycle and will not be discussed. The second engine was named Beta. The third engine was Gamma, and the last and full-size engine was named Rigel after the bright star in the knee of the hunter.

Beta Engine

A single-cylinder engine designated Beta was obtained for testing the active components of cylinder, piston, and fuel-injection system and the proposed method of aircooling the power cylinder. The Beta engine was motored and then fired on Nov. 21, 1952. This engine was originally constructed by Junkers in Germany as a single-cylinder, opposed-piston test engine. Upon receipt of the engine, it was modified

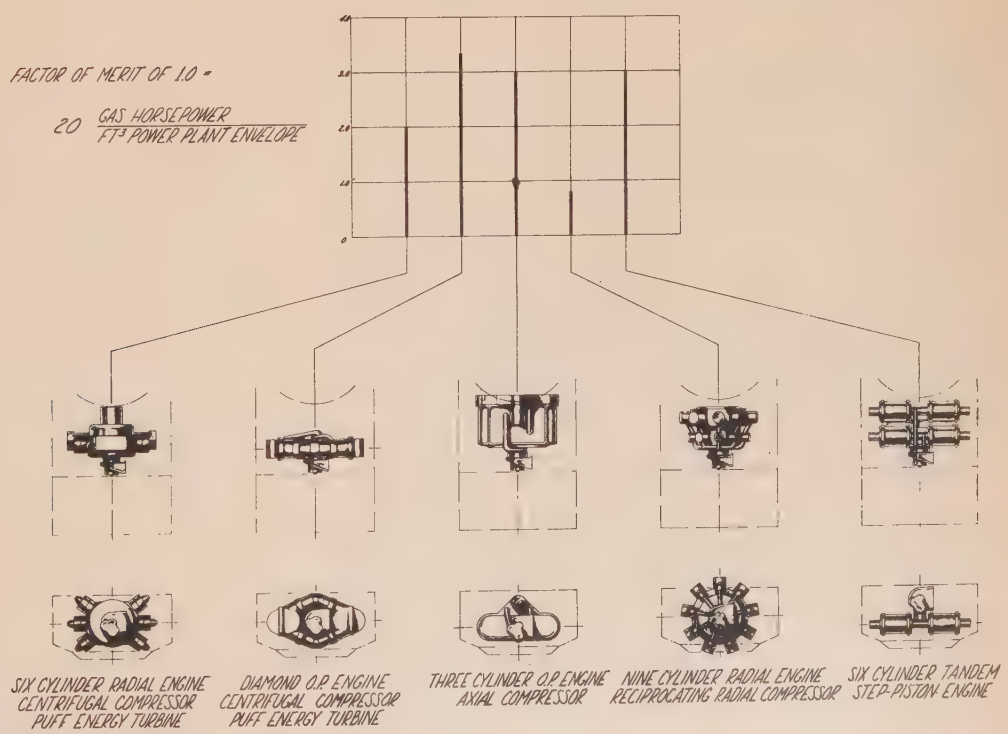


Fig. 1—Powerplant arrangement evaluation

to suit Orion requirements. This consisted of reboring the frame for the Orion cylinder since the outside diameter of the Orion aircooled cylinder was larger than the outside diameter of the original Junkers liquid-cooled cylinder.

Summary of Developments of Beta Engine

Testing started on the Beta engine Nov. 21, 1952.

1. Tests on Beta demonstrated the feasibility of cooling the cylinders with hot, high-pressure air (240 F at two atmospheres) at least two ways. Tests also proved that exhaust bridges need no auxiliary cooling means. The bridge temperatures at full load can be held at or below 500 F.

2. Preliminary tests indicate that uniflow cooling is possible.

3. Pear-shaped pistons improved combustion and permitted use of larger scavenge ports resulting in decreased pressure loss across the cylinder.

4. Hot-crown-type pistons gave unsatisfactory service. Further development is necessary to determine a satisfactory piston design utilizing a "heat-dam."

5. The diesel specific fuel consumption was improved to a value of about 0.38 lb per hp-hr and therefore met the requirements of the cycle analysis.

Cycle data for the Beta engine is given in Table 2.

6. Wristpin life was marginal when splash lubrication was used, but with pressure lubrication the service life was good.

7. Oil consumption was improved from 150 to 1500 hp-hr per gal by oil scraper ring variation.

8. Initial tests on the Beta engine indicated the need for injection-equipment development. The fuel-injection system will be discussed in detail later.

Fig. 2 shows the engine frame without cylinder or crankshafts assembled. Shown in Fig. 3 is, in some detail, the 3-piece cylinder assembly which is installed from the scavenge end of the frame. The exhaust end is fixed to the frame by a lock nut. Four shoulders on the cylinder mate with seats in the frame to form annular spaces for air passages. Figs. 4 and 5 show the cooling airflow through the cylinder for the two different configurations tested. In the "split-flow" cooling tests, a muff on the scavenge end separated cooling and combustion air as shown in Fig. 4.

The pistons, connecting rods, and crankshafts are installed as a unit. The gears on the crankshafts mesh with idler and power take-off gears, and the crankshafts are positioned by doweled main-bearing caps. The output shaft rotates at approximately one-half crankshaft speed.

Table 2—Single-Cylinder Full-Load Data

Combustion Air:	Flow	0.55 lb per sec
	Temperature	250 F
	Pressure	30.7 psia
Cooling Air:	Flow	2.0 lb per sec
	Temperature	220 F
	Pressure	To match combustion air pressure
	Bhp	125
	Ihp	160
	Speed	2340 rpm
	Bore	4¼ in.
	Stroke	5¾ in.
	2-cycle	Opposed piston

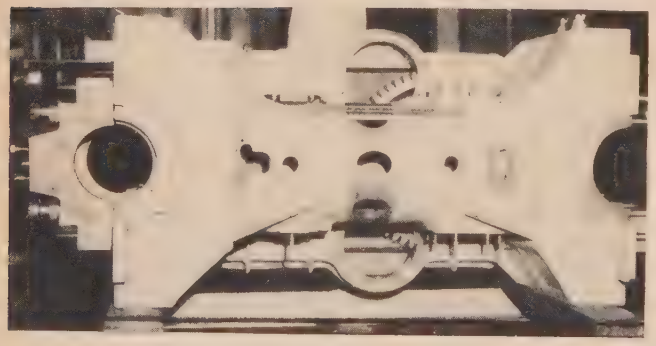


Fig. 2—Engine frame without cylinder or crankshafts assembled

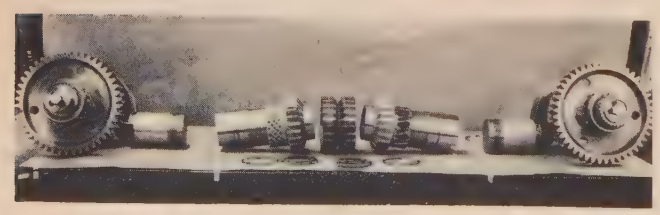


Fig. 3—Components of 3-piece cylinder assembly

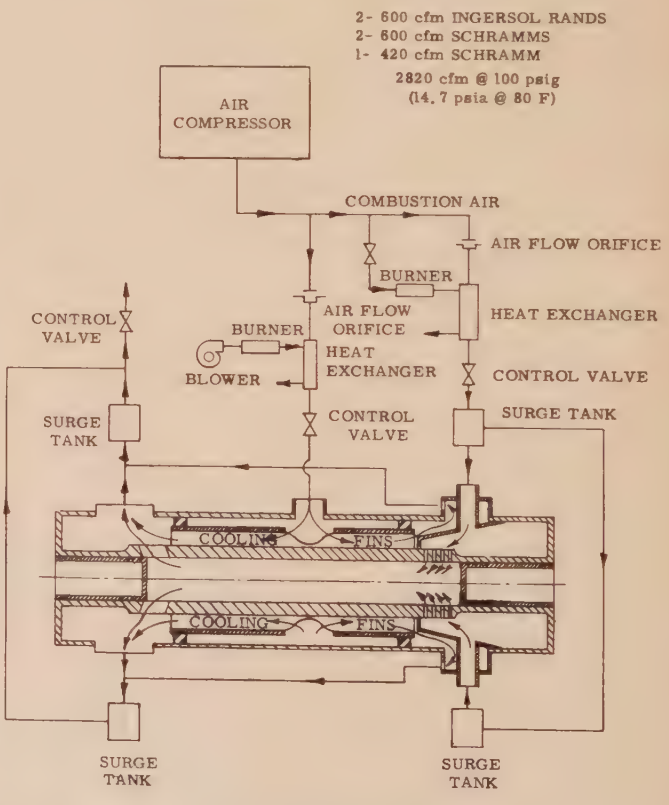


Fig. 4—Cooling airflow passage through cylinder for "split-flow" configuration

The fuel injectors are screwed into the cylinder liner and "O" ring seals to prevent cooling-air leakage. The fuel-injection pump is driven through a gear train by the exhaust crankshaft. Because of a torsional failure of the pump drive train early in the operation, a flywheel was added to the free end of the fuel-pump camshaft and a rubber coupling installed between the pump and gear train. Injection timing was checked by a strain gage on the injection tubing so that actual injection timing was known.

Variable timing was obtained by using a reworked Bosch variable timing mechanism. This device contributed greatly to the study of timing effects on fuel consumption.

The original Orion cycle analysis indicated the desirability of two compressors, one for cooling and another for combustion air. The Beta engine tests were started with cylinders using the "split-flow" cooling as shown in Fig. 4. The cooling air for this type of cooling entered at the center plane of the cylinder and divided into two streams which flow axially to the exhaust and scavenge ends. Probably less than one half of the cooling airflow cools the exhaust end of the cylinder while the remainder of the cooling air cools the scavenge end due principally to the pulsating effect of the exhaust "puff." The cooling air supplied was one-sixth of the design flow for the 6-cyl component research unit.

Following tests on the split-flow system, "uniflow" cooling was investigated. In this system the same cooling air cools the complete length of the cylinder

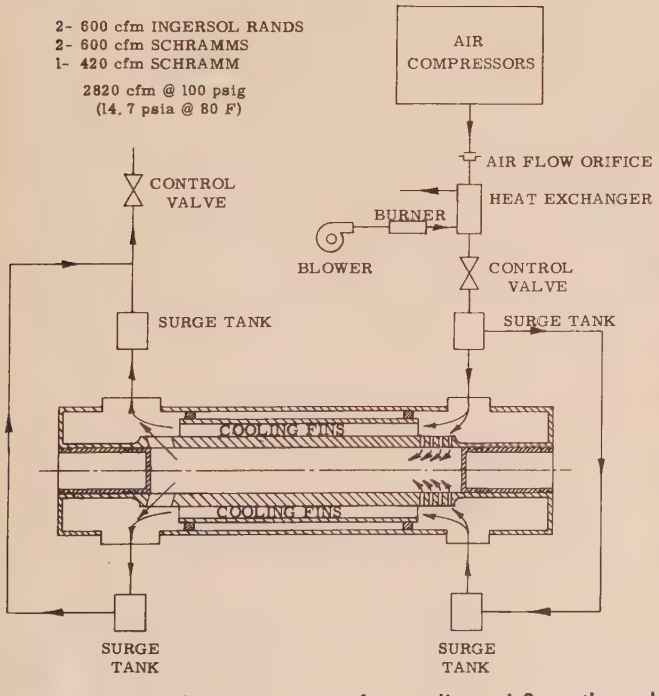


Fig. 5—"Uniflow" arrangement for cooling airflow through cylinder

cooling fins. The system is shown schematically in Fig. 5.

The major difficulty with the "uniflow" system was balancing the flow between the cooling and combustion air systems. Balance was obtained by restricting the discharge of the cooling fins. The attractiveness of this scheme lies in the simplification of the system by the elimination of one compressor.

Initial tests were made with a cylinder having air-cooled exhaust port bridges. The holes can be seen in the exhaust port bridges shown in Fig. 6. The cooling air system was arranged so that two sources were possible, the combustion air system and the main shop air-supply header. This arrangement was made to permit bridge-cooling variation without changing the combustion air supplied to the engine in addition to using a portion of the combustion air for bridge cooling. The cooling air was supplied to the muff surrounding the bridges through a pipe extending through the frame of the engine and connected externally to either of the two supplies previously mentioned. Fig. 6 shows the exhaust end of a bridge-cooled cylinder with the muff assembled.

Tests indicated that as combustion efficiency improved bridge cooling was unnecessary; therefore, subsequent cylinders were designed without bridge cooling.

The scavenge ports on the first cylinders consisted of various patterns of drilled holes. The original design consisted of holes spaced in such a manner as to give two symmetrical, but opposite, rotating swirl paths in the cylinder. The original engine design included two fuel injectors per cylinder; thus a double-swirl pattern seemed to be desirable. Definite double-swirl patterns were observed on the piston crowns after running the engine with this liner, but the pressure drop across the ports was excessive when compared to the design value for the cycle. Fig. 7 shows the swirl patterns.

A second scavenge-port configuration consisted of a series of holes drilled to give a single swirl pattern. Pressure drop across this port was less than the

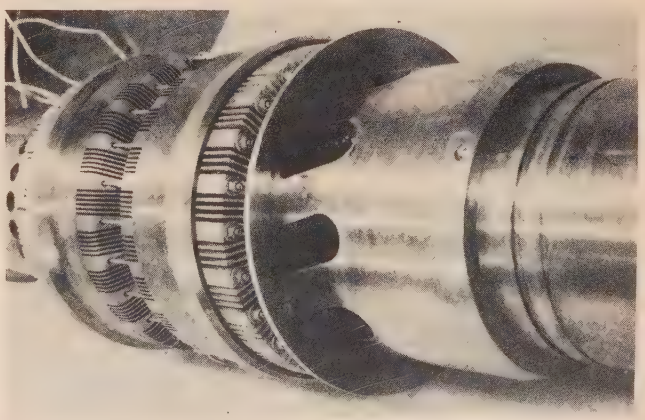


Fig. 6—Exhaust end of bridge-cooled cylinder with muff assembled

Fig. 7—Double-swirl pattern on piston crown resulting from design of drilled holes on scavenge ports

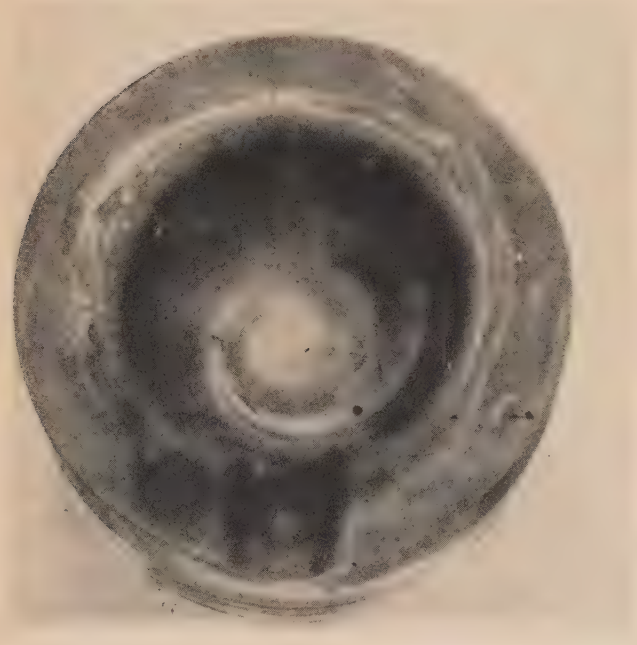


Fig. 8—Single-swirl pattern resulting from another design of drilled holes

"double swirl," but was still greater than the design value. The single swirl is shown to good advantage on the "pear" piston crowns, shown in Fig. 8.

The facilities for testing the Beta engine consisted basically of the following items:

1. Engine and dynamometer mounting (Fig. 9).
2. Resistance load for the dynamometer.
3. Diesel-generator set for motoring power.
4. Motor-generator set for field-control power.
5. Diesel-driven air compressors for external air supply (both cooling and combustion air).
6. Motor-driven pumps (external) for lubrication and cooling oil.
7. Motor-driven pump (external) for fuel oil supply.

The engine was connected to the dynamometer by a rubber coupling which was manufactured by Goodrich Tire and Rubber Corp.

Instrumentation

Temperatures were measured by Iron-Constantan thermocouples utilizing a Brown multipoint indicator.

Pressures were measured by:

1. Water manometers.
2. Mercury manometers.
3. Bourdon-tube pressure gages.
4. Heisse gages.

Flow was measured by:

1. Airflows were measured according to ASME specifications for airflow measurement.
2. Cooling and lubricating oil flows were measured by calibrated rotometers. Difficulty in reading the rotometers was encountered. To eliminate this difficulty, a reservoir tank and weigh tank were utilized to weigh the oil rate. This system was much more satisfactory.

Gamma Engine

Since the Beta engine predated the layout and complete design of the Rigel engines, it could only be considered as a jury-rigged test vehicle for obtaining a quick overall survey of the cycle possibilities. Therefore, as the design of Rigel progressed and as test information was available for incorporation into Rigel, a second test engine, known as Gamma was built. This engine was as closely equivalent to a 1-cyl section of the full-size Rigel engine as possible. Provision was made in its design to use the same cylinders, pistons, crankshafts, mainbearings, and connecting rods together with the same fuel pump drive, fuel pump, and other injection parts as were used on Beta.

This engine was to concentrate upon the active thermal parts of the cycle rather than the mechanical parts, which had been amply demonstrated on the Beta engine.

Summary of Gamma Engine Developments

Testing started on Aug. 5, 1953, using the same instrumentation, air supply, and other facilities as were itemized under Beta.

1. The pressure drops and air distribution between the cooling fins of the two ends of the cylinder were verified by test to satisfy the calculated predictions.
2. The pear-shaped pistons were given their "long-time" running tests and demonstrated that the final design should be made of stainless steel to prevent burning. Chrome plating the crown above the

ring belts did provide a short-time cure, but would eventually spall-off due to the high thermal stresses on the bond surface. Chromallized crowns ran without burning for over 100 hr.

3. The final work on improving the lubricating oil consumption was performed on this engine. With chrome-plated oil control rings on the cast ductile iron cylinders with 160-psi contact pressure or higher, the oil consumption was improved from about 80 to over 1500 hp-hr per gal. The lower figure was obtained when the project first started, and the rings' application was faulty.

4. The accumulator fuel-injection-system development was finalized on this engine. The performance satisfied the cycle requirements, and its life was not affected by the higher temperatures of the aircooled cylinder.

5. The mechanical design of this engine was more suitable as a test engine. It was easier to change test parts, and in addition, the gradual improvement of the cycle and of parts in general made it possible to accumulate more running time. As a result, this engine was used to run tests requiring longer periods of time such as the lubricating oil consumption tests and cylinder-wall tests.

It was first thought that both engines could be kept running, using each for a somewhat different type of problem, but due to the ease with which Gamma parts could be changed it took preference over the Beta engine. It should also be pointed out that because Beta was not representative of the Rigel engine in airflow pattern through the engine, the data obtained from it were always looked upon with a jaundiced eye. This, in itself, was a good reason for building the second test engine.

Both the Beta and Gamma engines used the same service systems and the same air supply. Consequently, when one engine was running, the other was idle; therefore, since Gamma turned out to be the better test vehicle, it accumulated the running time. After approximately 400 hr on the Gamma engine, a crankshaft gear failed. It was a fatigue failure, starting at the root, where the root radius blended into the tooth flank. Shotpeening and better quality control would prevent future failures.

Rigel Engine

The Rigel engines represented the culmination of the fourth phase of the "Orion" project. In these engines were assembled the separately developed components which were used to demonstrate and finalize the proposed regenerative-cooled gas-generator-turbocompound engine. Even though these components, consisting of cylinders, pistons, compressors, turbines, and so forth proved satisfactory on the single-cylinder test engines or on the compressor and turbine test stands, there were always present the peculiar matching problems when these components were applied to the multicylinder engine. These problems consisted of maldistribution of air between the two compressors due to cylinder cooling fins or scavenge-port design, mismatching of the turbine nozzle to the engine's total air demand, crankshaft phasing studies—and on and on. These problems can only be solved on a full-size engine.

It is seen from the evaluation chart, Fig. 1, that the opposed-piston type of engine had a very high rating, and in addition the use of a supercharge pressure ratio of slightly over two meant that advantage could be taken of the high specific performance of the centrifugal compressor. Six hundred turbine shaft hp requires 6 opposed-piston cyl in length, and while the performance "factor of merit" of such an engine was high it was found that by cutting the 6-cyl opposed-piston engine in two and folding one half back over the other, the resulting package of an acceptable

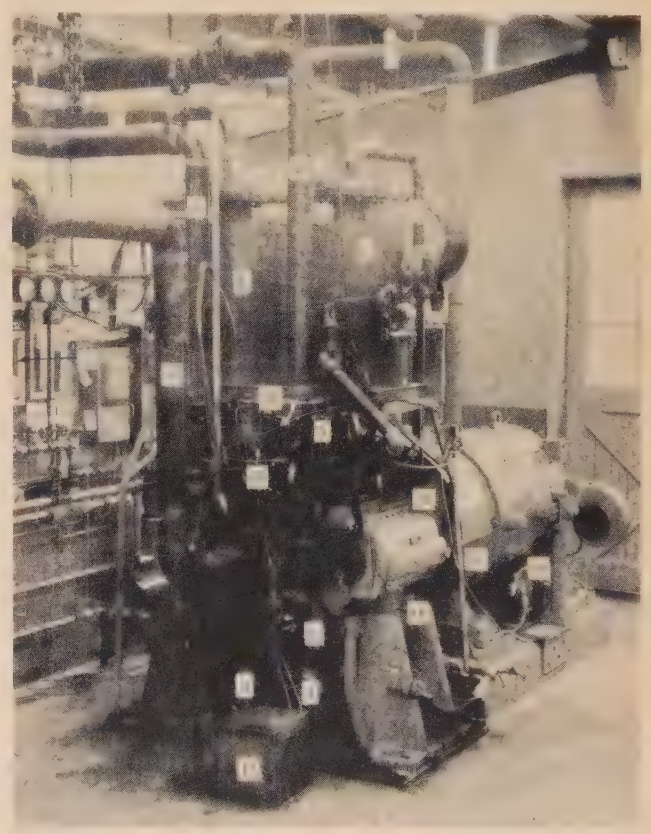


Fig. 9—Engine and dynamometer mounting for testing Beta engine.

Component Identification: (1) Scavenge air supply pipe, (2) Upper scavenge surge tank, (3) Lower scavenge surge tank supply line (air), (4) Lower scavenge surge tank (air), (5) Scavenge air supply lines—surge tank to engine, (6) Exhaust bridge cooling air supply line, (7) Engine exhaust line, (8) Upper exhaust surge tank, (9) Lower exhaust surge tank, (10) Cooling air supply line, (11) Cooling air discharge lines, (12) Lubrication oil rotometer, (13) Piston cooling oil rotometers, (14) Crankcase oil drain lines (one per crankcase), (15) Fuel injection pump, (16) Emergency stop solenoid (fuel rack), (17) Cooling oil tank for injection pump, (18) Lubrication oil pump and motor for dynamometer, (19) Flywheel, (20) Dynamometer

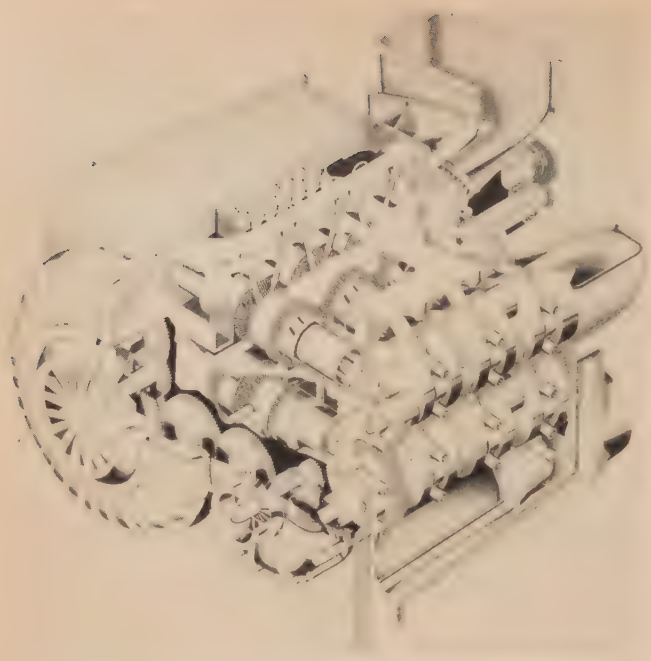


Fig. 10—Cutaway view of Rigel engine

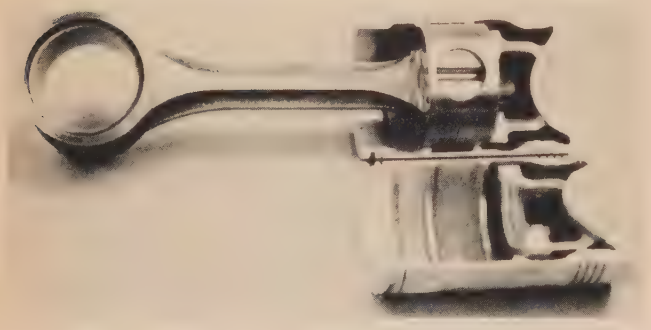


Fig. 11—Connecting-rod details

rectangular dimension and no larger in height than the compressor is in diameter. An artist's drawing of the Rigel engine is shown in Fig. 10.

In essence, the Rigel engine is a rectangular package, approximately 35 in. high x 47 in. wide x 67 in. long, consisting of two rows of three horizontal cylinders, with four 3-throw crankshafts all geared together. On one end are the two centrifugal compressors. These are driven from the gear train that ties all four crankshafts together. On the opposite end of the engine is the power turbine. The discharge from the two air compressors enters their respective conduits, flows through the cylinders, and into the turbine. The turbine is mechanically free from the gas generator proper; therefore, its output in speed and torque are functions of the load only and are self-adjusting in accordance with the gas flow from the gas generator. Hot gas flow from the gas generator is a unique function of the throttle setting.

The connecting-rod construction permits squeezing two cylinder rows together approximately 3 in. This is the result of saving two bolt-boss dimensions on the two rods that are bound to roll together on some

Table 3—Comparative Performance of Piston Engines

Type of Engine	Make of Engine	Type of Cooling	Maximum Shaft Hp per Sq In. of Piston Area
Diesel 2-stroke	GM locomotive 567-B	Liquid	1.89
Diesel 4-stroke	Alco 244	Liquid	2.28
	Alco 251	Liquid	2.54
Diesel 2-stroke	Junkers Jumo 205-E	Liquid	3.73
Gasoline 4-stroke	P & WA 4360	Air	5.23
Diesel 2-stroke opposed-piston uniflow	Orion	Air	5.28
Gasoline 4-stroke (sleeve)	Napier Sabre VII	Liquid	6.5
Gasoline 4-stroke	Wright Compound	Air	6.97
Diesel 2-stroke (loop scavenge)	Napier Nomad	Liquid	9.0

one of the 12 crank throws. The crankshafts are built up of sections bolted together with Gleason curvic face splines transmitting the torque. The bolt tension produces a compression load on the spline teeth of sufficient force to withstand the bending moments. Fig. 11 shows connecting rod details.

The two fuel nozzles shown extending at an angle to each other from the center of each cylinder (Fig. 10) have, since this drawing was made, been replaced by one single nozzle. This will be discussed more in detail later. The six fuel-pump discharge fittings were reduced to three as mentioned above. It will be noticed that the exhaust ports are in evidence on the right-hand end of the upper row while the exhaust ports are to the left-hand end on the lower row. This cylinder arrangement assures a more even temperature distribution and reduces frame distortion.

To engineer an entirely new powerplant, there will be apparent certain outstanding problems that must be solved. These problems in the case of Orion and in the order of their importance are discussed in detail below and listed here as:

1. Aircooling—(a) cylinder, and (b) exhaust port bridges.
2. Combustion efficiency—(a) cylinder, (b) ports, (c) injection system, and (d) pistons.
3. Pressure drop.

Cylinder Cooling

In the beginning, the biggest problem was the ability to cool a diesel engine with air. Since there was no engine of this type available on the American market at that time, the opinion was generally expressed that there must be something wrong with this idea. These arguments persisted in spite of millions of aircraft engine horsepower flying every day. For this reason the cooling problem was most thoroughly analyzed. Table 3 gives some very important com-

parisons between engines of various makes, ratings, and thermal loading.

With the specification of aircooling it was considered worth while, in the light of reliability and life, to use a larger percentage of "pneumatic" transmission and reduce the brake mean effective pressure. To balance the engine and compressor power, a higher bmep is required with fan cooling. The division of work between the two air compressors is discussed in the Appendix. In addition, if the heat picked up by the cooling air could be made to do work on the turbine blades then a certain credit could be given to the heat balance over the engine. This is the regenerative-cooling idea; the heat picked up from the cylinder walls just about balances out the losses in pressure drop through the fins.

The reason for using two compressors was predicated on two facts; the first is that the combustion air must have finer filtration with a consequently higher filter pressure loss than required of the cooling air. Secondly, the pressure drop through the cylinder ports is greater than the drop through the cylinder cooling fins, and consequently the two airstreams operate at different pressure ratios. The combustion air compressor pressure ratio was 2.25 as against 2.06 for the cooling air compressor.

The layout of the cylinder cooling fins can be deduced from Fig. 4. Fig. 12 shows the temperature distribution along the axis. Because cylinder cooling was such a controversial question, several tests were made to verify the heat-transfer calculations as well as to measure the pressure drop and to demonstrate the streamline pattern. Both tests are shown in Figs. 13 and 14 respectively.

The next area that was considered vital in the life expectancy of the engine, due to high temperature, was the exhaust port bridge. As can be seen from Fig. 12, these temperatures tested out at about 500 F. To have an alternate, various cooling schemes were laid out, and several were tested on a single-cylinder engine. One of the first successful schemes is shown in Fig. 6, where air is blown into the port cavity through the small drilled holes. The air prevented the hot gas from contacting the port walls as well as absorbing heat from the port and cylinder wall. When the engines first started, the fuel consumption and

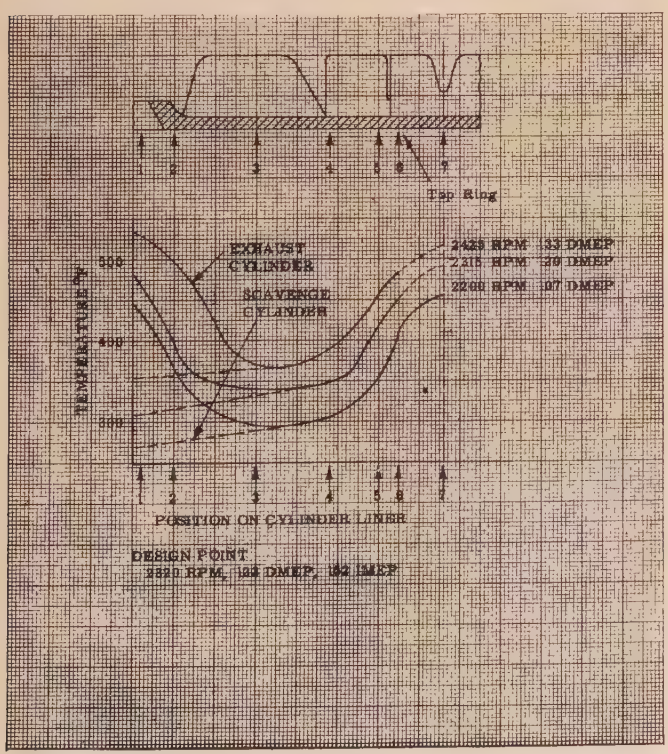


Fig. 12—Temperature distribution at inner surface of Beta engine cylinder liner, two nozzles. Temperatures are extrapolated from results of tests run May, 8, 11, 12, and 13, 1955

Fig. 13—Test setup for verifying heat-transfer calculations

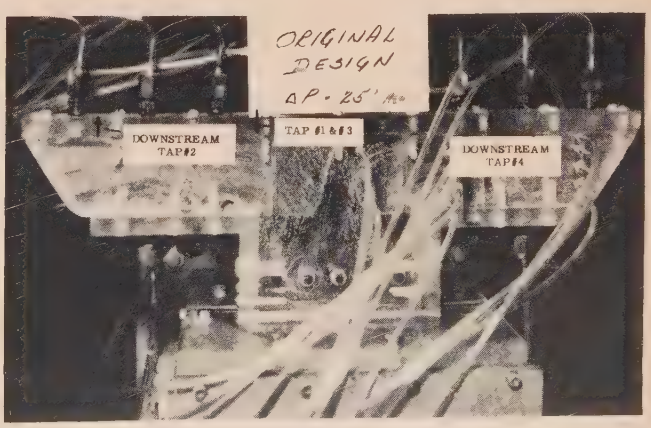


Fig. 14—Test setup for measuring streamline pattern and pressure drop

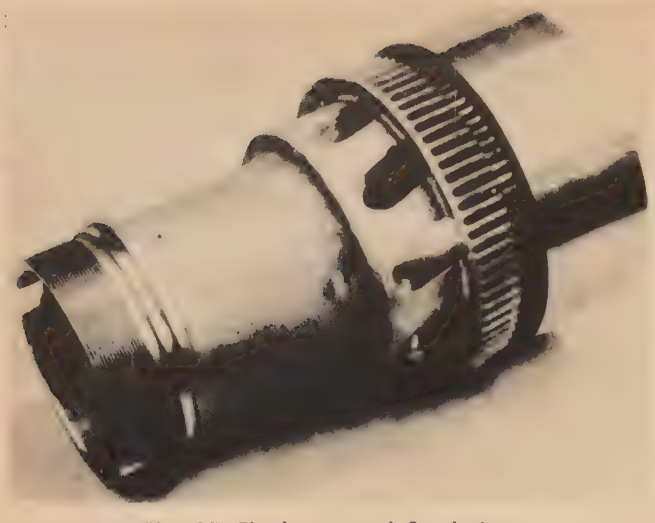


Fig. 15—Final port and fin design

the exhaust temperatures were high. As a consequence, the port temperatures were also high, which was of some concern. However, as soon as the specific fuel consumption began to improve, as will be discussed later, the bridge temperature also came down with the result that this cooling method was discarded. The port and fin design adopted as final is shown in Fig. 15.

Cylinder Design

The cylinder program was so intimately interconnected with other elements of the combustion system that it is hard to discuss one without the other. The development of cylinders was nevertheless carried out in three phases.

1. Design and running an original cylinder for preliminary test results.
2. Design changes to obtain the required performance.
3. Systematic design program to obtain the best possible cylinder.

The first cylinder designed is shown in Fig. 16. This was a one-piece steel sleeve with shrunk-on cooling fins. The scavenge ports were drilled for a "double swirl" as shown in Fig. 17.

It was immediately apparent that the one-piece cylinder was too costly for our development programs. Therefore, to increase the availability and re-

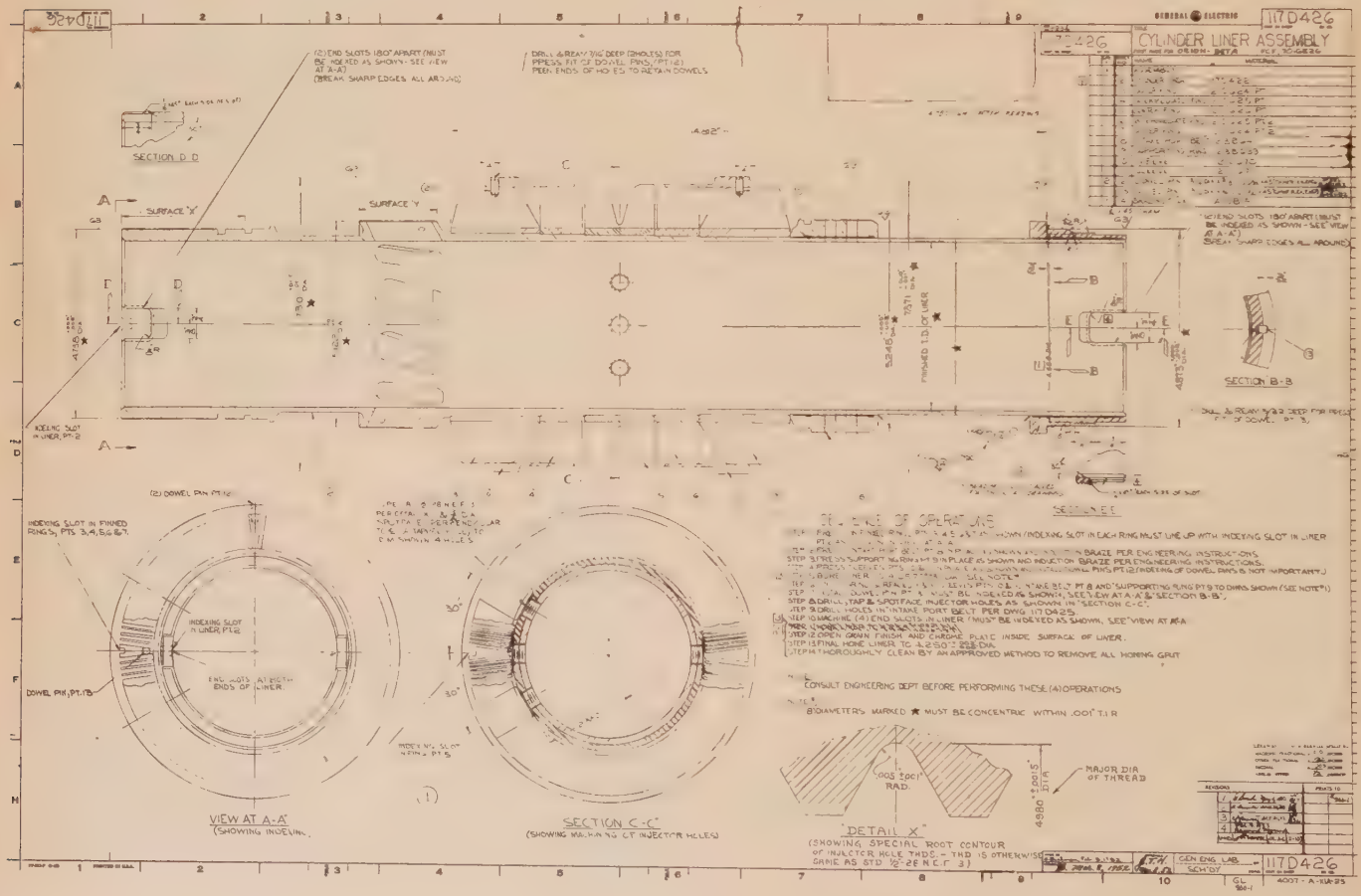


Fig. 16—First cylinder-liner assembly for Orion Beta engine

duce the cost, a 3-piece construction was developed as shown in detail in Fig. 17. In this cylinder an Al-Fin muff was cast onto the cylinder sleeve, and the fins were then milled into it. The fin spacing is 0.145-in. pitch x 0.050-in. fin thickness. The two end pieces, the scavenge and exhaust halves, were shrunk into the center finned section, which is made of beryllium copper. This material was chosen because of its high-temperature strength and excellent heat-transfer qualities. This center fin is shown in Fig. 18, and the fin spacing is 0.100-in. pitch x .040-in. fin thickness. The three pieces were bolted together by the slender tie-rods made from bicycle-spoke material. The shrink forces were sufficient to hold the cylinder together, but the tie-rods were an additional safety measure. All of the early cylinders were porous chrome-plated until the all-steel and the ductile iron cylinders were made.

The "double swirl" was not satisfactory, neither combustion-wise or pressure drop-wise. The next port drilling was for the single swirl as shown in Fig. 19. Other variants are shown in Fig. 20. These two port designs were a big improvement over previous ones.

The Al-Finned cylinders with porous chrome-plated bores were requiring too long a time to manufacture. It was decided to investigate the all-steel cylinder, made of 4140 steel and heat-treated to R$_c$35. This design was of the same essential dimensions as the Al-Finned cylinder, except that the fins were milled from the solid steel. A photograph of a

Fig. 18—Center fin

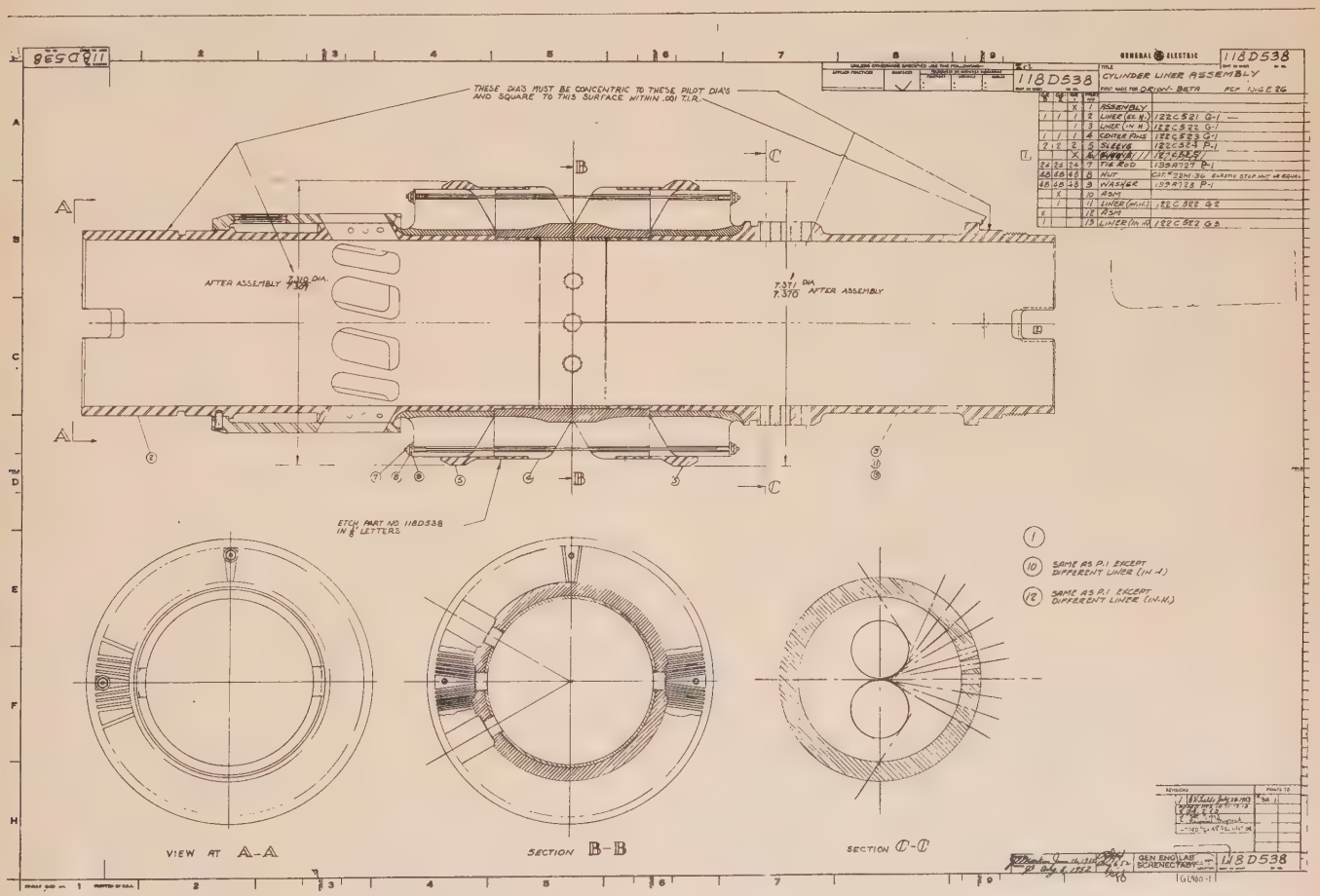


Fig. 17—Three-piece cylinder-liner assembly showing scavenge ports drilled for double swirl

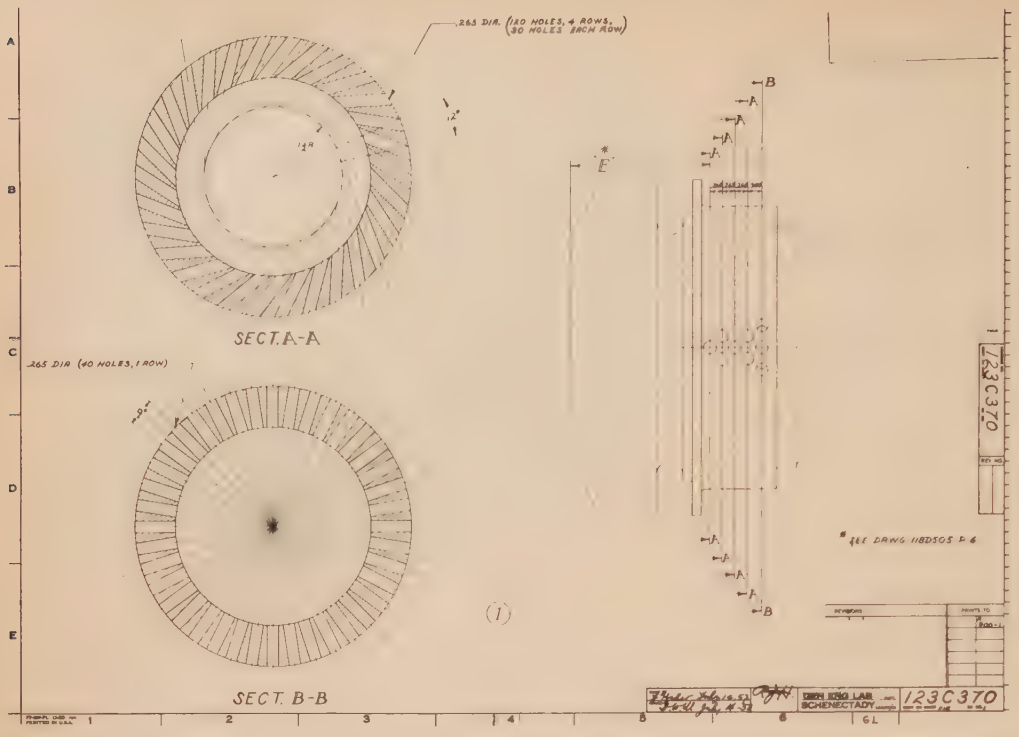


Fig. 19—Port drilling for single swirl

finned exhaust half is shown in Fig. 21. This was one of the earlier cylinders without bridge cooling. During this same period, exhaust port bridge cooling was of great concern, and it was advisable to have, prepared and available, other designs ready for test. A special cylinder with cooling fins on the port bridges was made. This cylinder is shown in Fig. 22.

The cooling air is deflected upward by a stamped-out shield that was welded to the fin. In the cylinder of Fig. 22 it is shown that this deflector is a part of the solid steel section from which the fins are milled. The exhaust port bridge temperatures were not as high as expected, and consequently this cylinder was never run.

Tests demonstrated that the engine needed two things; first, single swirl and second, low pressure

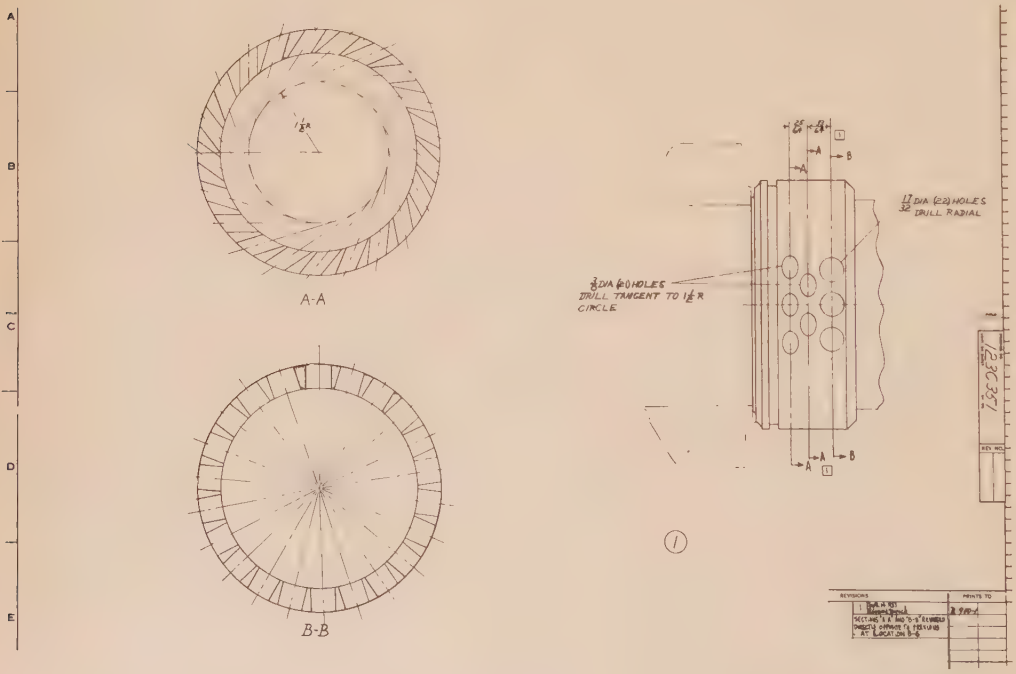


Fig. 20—Third port drilling design

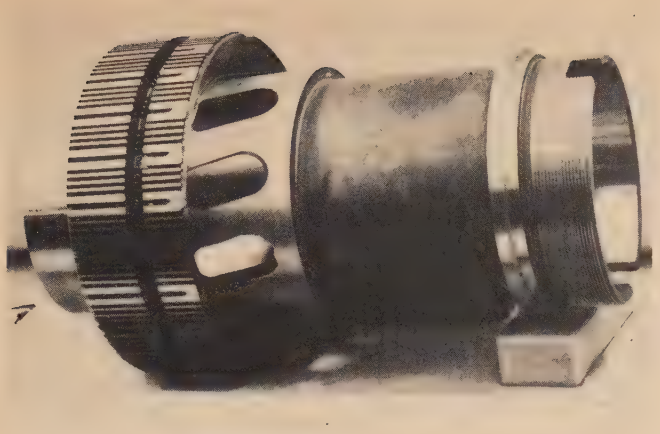


Fig. 21—Finned exhaust half of all-steel cylinder

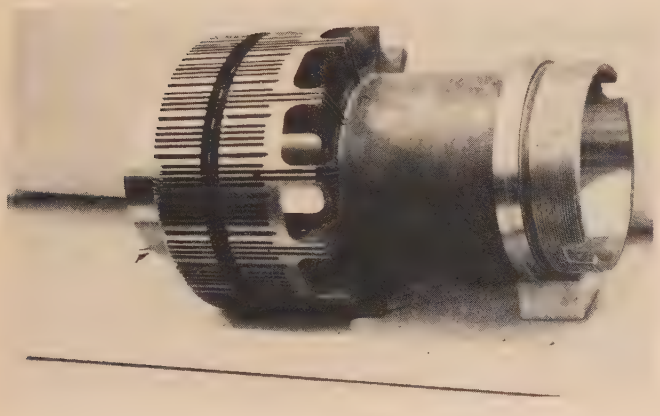


Fig. 22—Cylinder with cooling fins on port bridges

drop. Consequently, the next cylinder design was a tangential ported all-steel cylinder as shown in Fig. 23. This proved to be the best port design; therefore, other methods of manufacture could now be devised.

Since the beginning of the 3-piece cylinder design it was a continued hope that their cost and procurement time could be reduced.

Before the acceptance of the tangential ports, the next step was to investigate a cast cylinder. After much investigation, the Howard Foundry in Chicago was found to be sympathetic to the idea of trying to cast a completely finned half cylinder of ductile iron. The assembled cylinder is shown in Fig. 24. These same cylinders were later "honey chromed" as shown in Fig. 25.

In the discussion of the Beta engine, mention was made of the uniflow-cooled cylinder. One of these cylinders was built and tested and is shown in Fig. 26. In addition, it was a radial-ported cylinder for investigating the "squish-swirl" piston effect on combustion. While the combination of this "squish-swirl" piston and cylinder was nothing to be enthused over,

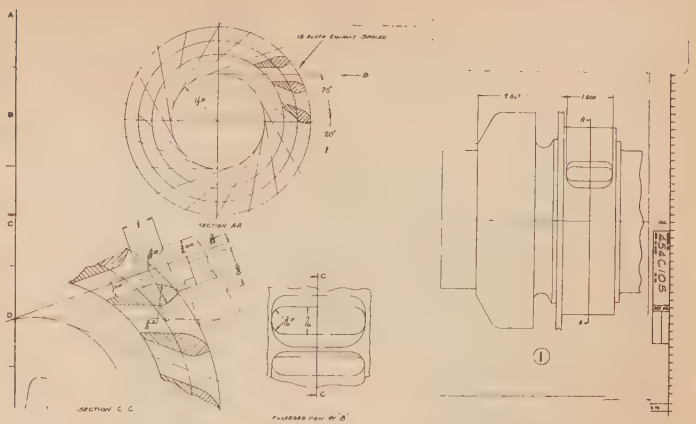


Fig. 23—Tangential ported all-steel cylinder

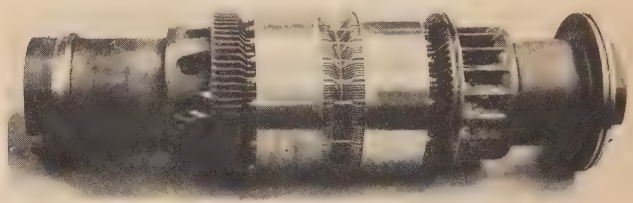


Fig. 24—Assembled cast cylinder

Fig. 25—Honey-chromed cast cylinder

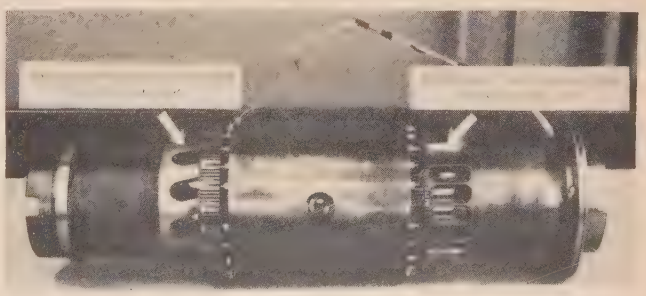


Fig. 26—Uniflow-cooled, radial-ported cylinder

Fig. 27—Ported cylinder liner designed to give low pressure drop of combustion air across liner

it nevertheless seems to have a potential that is worth while in reducing the cylinder port drop for a compound type of engine. This cylinder and matching piston were run just once because the project was finished before it was again available for test.

As the testing continued on the single-cylinder and the Rigel engines, with good data being obtained and analyzed, it was evident that the Orion cycle could be much improved if it were possible to redesign the engine and combine, in one cylinder, the following items:

1. Increase the fuel/air ratio in the cylinder.
2. Decrease the port drop across the cylinder.
3. Combine the combustion air and the cooling air compressor thereby saving high-speed bearings and gears.

Tests of ported cylinders showed that it is possible to get very low pressure drop of the combustion air across the cylinder liner. The cylinder liner pictured in Fig. 27 gave an equivalent pressure drop of 2.6 psi. By moving the ports 0.3 in. toward the center of the cylinder in the cylinder shown in Fig. 28, the pressure drop decreased from 6.5 to 4 psi (equivalent pressure drop). Moving the ports by the same amount on the cylinder of Fig. 27, the equivalent pressure drop would have been 1.5 psi. Since the pressure drop across the fins is approximately 1 psi, it is possible, by introducing a resistance in the flow through the fins, to have the same pressure drop in the cooling and combustion air circuits. The design of the 6-cyl engine then becomes much simpler for there is only one compressor (saving one impeller and corresponding gearing) and one air path. It is therefore possible in the same engine frame to increase the cylinder diameter by 20%. In addition, the parasitic pressure drop between the cooling air compressor and the cooling fins is eliminated.

Since the change from double flow to uniflow cooling allowed a larger diameter, a cylinder was designed to reduce the pressure drop to a minimum. The bore diameter was increased from 4.25 to 4.75 in. (that is, 25% increase in the piston area). This permitted the design of much larger ports as pictured in Figs. 29 and 30. Based on previous tests the pressure drop of the combustion air across this cylinder should have been 1.3 psi (at a flow of 0.53-lb per sec air) with pear-shaped pistons or 0.8 psi with flat pistons. This cylinder was ready for testing early in March, 1955, when the decision was made to stop all long-range programs. A special feature of this 4.75-in. bore cylinder is that part of the combustion pressure is carried by the fins to the muff around them acting like ribs to carry the firing load from the thin cylinder to the stronger muff. This was done to keep the liner thickness to a minimum and to obtain lower cylinder temperatures.

It should be mentioned, in all fairness to the many scavenge port models made and flow-tested, that the final cylinder design looked like nothing that had ever been tested before.

Piston Design

The piston history followed pretty much the same development pattern as the cylinders. In this discussion each piston will be identified as a part number of Fig. 31. To match the original double-swirl cylinder, a flat-top piston was made and is shown in part 1. A double-swirl pattern on one of these piston crowns is shown in Fig. 7. This piston was an all-steel, copper-brazed construction with an Al-Fin muff on the skirt. All of the Orion pistons were oil cooled using the "cocktail-shaker" principle. While the fuel consumption with this combustion-chamber shape was poor, the piston ran very well in regards to life and strength. It was, however, soon replaced to improve the specific fuel consumption.

The next piston design was of the same general style except that the crown was a "Mexican hat," shown as part 2, and made for concentrating the air closer about the fuel nozzle spray. This crown was used with the single-swirl cylinders and was somewhat better in specific fuel consumption but in turn was also replaced.

The final piston design, after several modifications, was the "pear-shaped" combustion chamber. It should be pointed out that while these various pistons are being discussed as entities they were only a part of a general development program consisting of work on

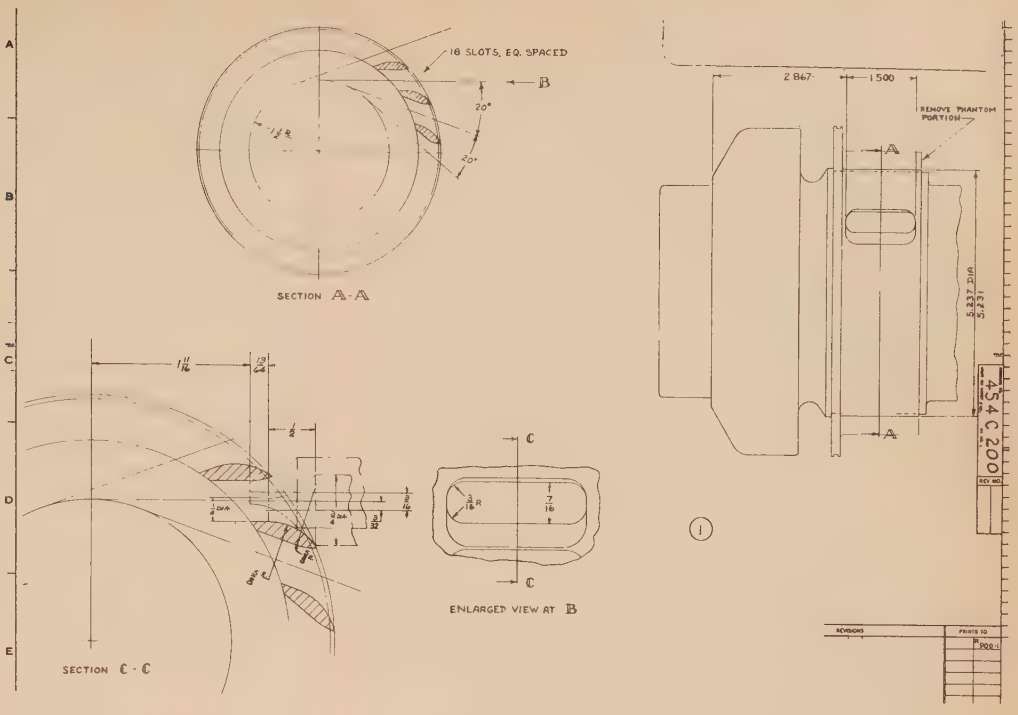


Fig. 28—Intake port belt drilling which gives decreased pressure drop

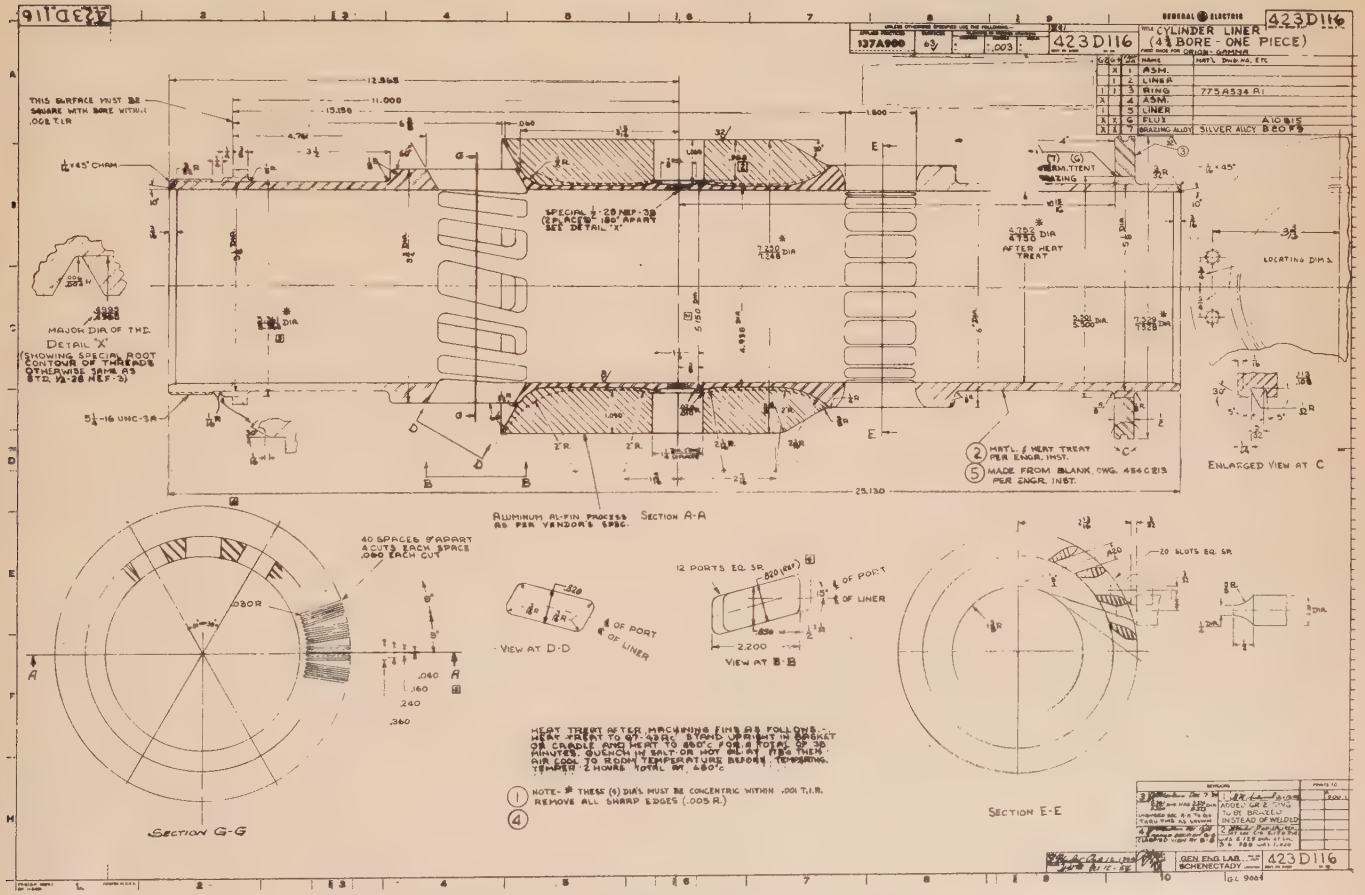


Fig. 29—Cylinder-liner design permitting much larger ports

cylinder ports and fuel-injection systems as well as the combustion-chamber shape. This piston is shown as part 3. The geometry of this "pear shape" could be formed by a ball end milling cutter feeding in at about 35 deg to the plane of a diameter. A photograph of this piston crown is shown in Fig. 32. Prior to the "pear-shaped" piston a separate aluminum piston-skirt design had been completed and was in use. This skirt permitted interchangeability with other crown designs as well as with new wristpin holders. An exploded assembly is shown in Fig. 33.

The "pear-shaped" combustion chamber was a marked success from the very beginning. It accomplished several things; first, it permitted the use of one injection nozzle, which had been bench tested and developed concurrently with the pistons and cylinder; second, it tended to reduce the heat rejection through the cylinder wall by covering up the exposed wall; third, it concentrated the air around the fuel spray and made it theoretically feasible to try to increase the equivalent orifice area through the cylinder and added "squish" to the combustion air. By properly designing for "squish" swirl, it was felt that it would be theoretically feasible to increase the equivalent orifice area through the cylinder by using radial ports, rather than the above-mentioned tangential ports. This phase of the development will be discussed later.

With the success of the "pear-shaped" piston, the next revision was a modification converting the "pear" to a "spherical" cavity for reasons of manufacturing simplicity. The "sphere" could be turned on a lathe. This piston assembly is shown as Fig. 31, part 4. These piston crowns were used in cylinders having a very strong tangential swirl as is evidenced in Fig. 8. All of these crowns were machined from low-carbon steel and consequently were not corrosion or high-temperature resistant. With engine performance improving, running time was getting longer, so that burning of the pistons' "pear" and "sphere" edges at the nozzle openings was appearing, mostly on the exhaust piston. Rather than make stainless steel crowns, two ways were tried to reduce the burning; one was to chrome plate the crown above the ring belt, and the other was to chromallize the surface above the top piston ring. While chrome plating was easy and cheap to apply and helped to demonstrate

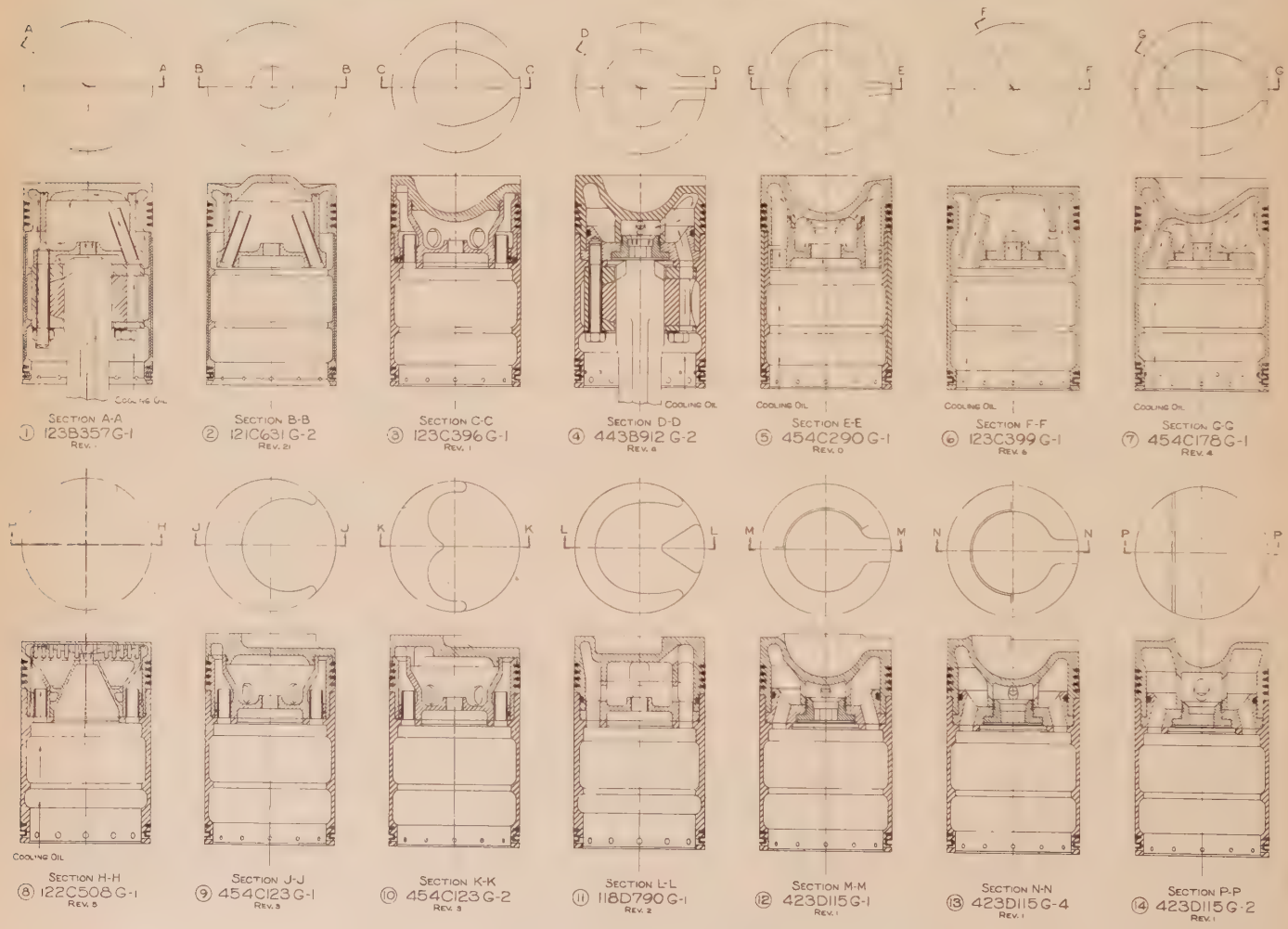


Fig. 31—Piston design types

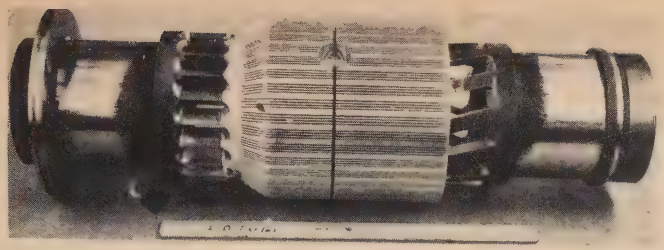


Fig. 30—Cylinder-liner assembly permitting much larger port design

Fig. 32—Pear-shape piston crown design

Fig. 33—Exploded assembly showing interchangeable piston-skirt design

Fig. 34—Chromallized piston crowns

a cure, it would not stand up under the high-temperature gradients. The bond strength would fail and allow the plating to peel off. The chromallized process is a metallurgical addition of chrome to the skin for a depth of about 0.0015 in. This proved to be a cure for the burning as demonstrated by over 100 hr of operation. Incidentally, the carbon could be wiped off easily, leaving a smooth brownish appearance to the crown. Fig. 34 shows a pair of chromallized piston crowns.

As stated above, one area of burning started at the channel for the nozzle spray. It was reasoned that since the top edge of the crown opened the exhaust ports for blow-down, the channel for the nozzle spray of the "pear," being nearly as wide as the port land would preopen the port allowing a much longer time for the hot gas to blow across the channel edge. It would at least contribute to the burning rather than the opposite effect. In order to reduce this preopening effect, a set of pistons was made without a channel and tested. The results were poor, but by progressively hand filing a channel in the crown it was possible to demonstrate that only about one half the opening originally provided was required. This small channelled crown is shown as part 5 of Fig. 31.

The engine's performance was now an accomplished fact, and time could be spent on thinking up new schemes for improvement of the cycle. As mentioned above, concurrent developments were going on for cylinders and their ports. Also, it was recognized that the reduction of scavenge port drop might be accomplished if it were possible to develop a "mechanical squish-swirl" piston that would create the necessary turbulence as the pistons approached inner volume dead-center. Accordingly, two basic crown designs were constructed. One was a spherical cavity crown with the swirl lips oriented to produce a swirl about the axis of the fuel nozzle spray. This piston is shown in Fig. 35. The second piston crown was a "cylindrical" cavity with the swirl lips also producing a swirl velocity about the axis of the nozzle (two nozzles could be used with this crown). This piston is shown in Fig. 36. Both of these piston crowns and swirl lips were of the same orientation so that two of the same part number made a complete combustion cavity. The third modification was similar to the first

Fig. 35—Piston with spherical cavity crown

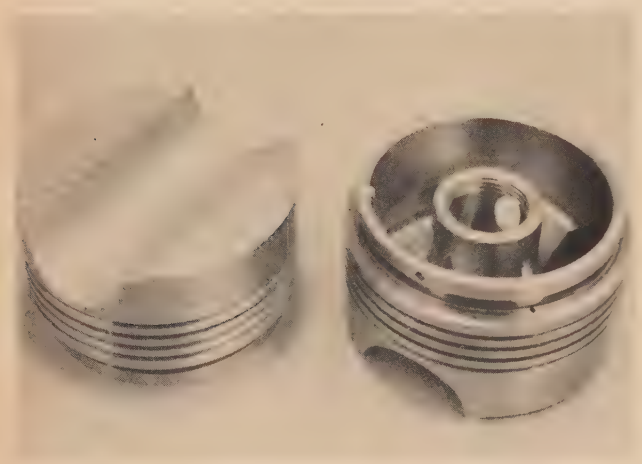


Fig. 36—Piston crown with cylindrical cavity

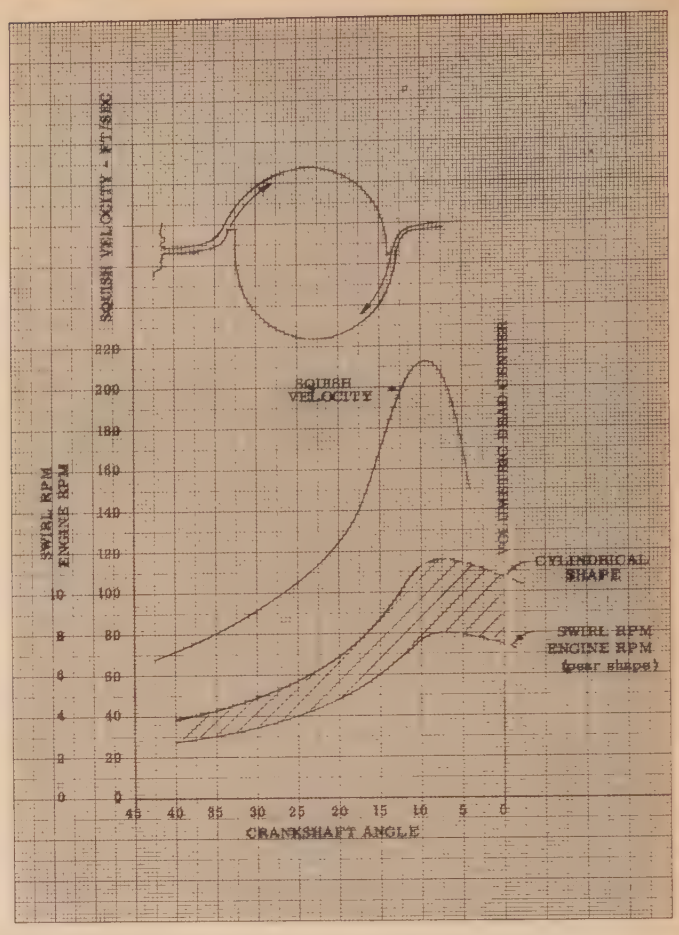


Fig. 37—Squish velocity and swirl rpm per engine rpm ratio

spherical cavity crown except that the swirl lips were turned 90 deg to enable the swirl to cross the fuel spray. These two crowns were not identical as is shown in Fig. 31, part 13.

Of these pistons, only the first spherical cavity was ever used in an engine. The swirl lips were only about half the height shown, and while there was an improvement in the specific fuel consumption when used in a tangential ported cylinder, it was marginal. The lips were doubled in height by welding, but the project ended before they could be retested in the single-cylinder engines.

The calculated swirl and approach velocities for these "mechanical squish-swirl" piston crowns are given in Figs. 37 and 38.

While the success of these pistons in a radial ported cylinder was never satisfactorily demonstrated, it is hoped that someone can continue the development of such a combination in either the opposed piston or a conventional single-cylinder and single-piston engine, particularly in connection with a compound engine development.

With radial scavenge ports it has been computed that it would be possible to obtain an equivalent orifice area through the cylinder equal to 5.25% of the bore area. One analytical investigation made on this powerplant showed that the gain in turbine power above 4½% orifice area was at diminishing returns. The "mechanical squish-swirl" piston with a radial scavenge ported cylinder would still be an overall advantage to a compound powerplant.

In a compound engine, the heat in the hot gas at a pressure consistent with the other elements of the cycle represents the bloodstream of the powerplant. Therefore, anything that can be done to reduce the heat rejection from the cylinder walls or piston crowns would certainly mean that that much additional heat would remain in the hot gas increasing the energy available to the turbine. It has been computed that the heat rejection through the piston crowns in about 4% of the heat of the fuel which in itself is not too bad, but when it has to be cooled out of the lube oil system it is then equivalent to more than 4% of the fuel heat.

For this reason a "hot-piston" crown development was initiated to study and design a "hot" piston. For testing these hot-crown pistons a 2-cyl, 2-cycle, GM 71 series engine was procured so that it would not tie

up our single-cylinder engines which were more urgently needed in other work.

The first "hot" flat-crown piston is shown as part 1 of Fig. 39. The crown is floating and held to the piston skirt by a snap ring. The assumption was made that the gas pressure would hold the crown down, on the basis that the edges of the crown would seal the gas pressure, keeping the pressure under the crown low. This, however, was not the case. After 2¾ hr of running it was found that because of the very high thermal gradients in the hot crown proper, it had distorted allowing the gas pressure to equalize on both sides and thereby making the snap ring groove take the inertia load of the crown at top dead-center. The ring groove was also cracked in the corner of the crown.

In order to maintain some semblance of a pressure seal between the crown and piston skirt, a piston ring was added to the outer edge of the crown. In addition, the strength of the attachment of the crown to the piston proper was increased by using an artillery type of interrupted jaw clutch. This also permitted easy removal of the crown. This design is shown in Fig. 40.

Tempil pellets were imbedded in the underside of the crown to measure the temperatures. This crown was operated for 15½ hr on the GM engine with good results as far as distortion was concerned except that the piston ring was stuck in the groove and was annealed. However, the test did prove that a gas force is necessary to balance the inertial force and that the ring must be in a cooler place.

This test resulted in a modification on the "artillery clutch." One was to add an overhanging lip on the outside diameter of the crown which was sealed by a piston ring in a grove in the ring carrier. This design is shown as part 2 of Fig. 39. The second modification was the spring-loaded crown made from a "clutch" ring carrier shown as part 5. Both of these pistons were run for 17 hr in the GM test engine with good results with the exception that the ring had lost some tension. Its cooling would have to improved. When this piston was put in the Beta single-cylinder engine and had run for 4½ hr, one of the jaws broke off. The material in all of these crowns was 347 stainless and would not be good enough for "long-time" running. Inconel X would have to be used in a final design. The spring-loaded crown was also run for 4½ hr in Beta. It did not show any distress.

The engine proved to be unable to stand the additional thermal load that was necessary to test the pistons. This resulted in dispensing with this engine, and the tests continued on the single-cylinder engines.

After the last tests on these two hot crowns, they were again redesigned. The "clutch" was moved from the inner to the outer diameter. These are shown as part 7 of Fig. 39 and in Fig. 41.

The spring-loaded crown was redesigned to include Bellville spring washers and the outer overhanging lip with the outside piston ring as shown in Fig. 42 and as part 6 of Fig. 39.

All of these hot crowns were flat except some modifications that were referred to as "Mexican hat" and "half moon." These are shown as part 3, part 4 of Fig. 39 and in Fig. 43.

Once the "pear-shaped" piston was a success, further work on hot-crown pistons was held in abeyance because of the need of other and more urgent problems. The project was completed before returning to complete the hot-crown developments.

Fuel Injection

Concurrent with the development of cylinders and pistons, the fuel-injection system was undergoing a rather drastic shakedown. The fuel-injection requirements that were specified in the original order for the fuel pumps stated that the pressure was to be about 10,000 psi at 2400 rpm with about 105 cu mm of fuel injected per stroke in each of two nozzles per cylinder, and that the cam angle duration was to be about 11 deg. The pump supplied was a 4-barrel automotive-type unit but with a speed limitation of 1200 rpm (or half engine speed). The pump is shown in Fig. 44. With such a pump, each nozzle had to be fed by two barrels through a Y block. The camshaft

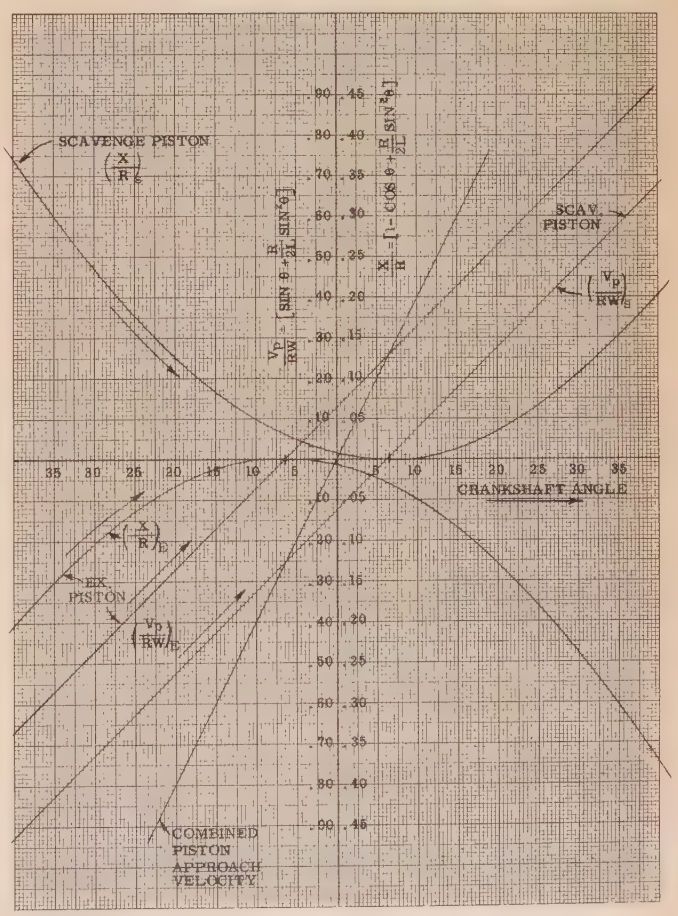


Fig. 38—Piston position and velocity for calculating swirl velocity

for this pump had two sets of tangential cams at 180 deg. The fuel pump, line, and nozzles are shown in the schematic drawing Fig. 45. This arrangement was first and only used on the Beta engine. The first fuel-injection nozzle was patterned after the open-hole Junkers fuel nozzle, shown in Fig. 45. The duration of injection for such a system was entirely too long (nearly 300 deg), and it was thought that the extra volume of the Y block and the other pump were partly responsible. The pump check valves were removed and filler pieces inserted in the unused volume. This reduced the injection duration to about 100 deg. In order to further alleviate this situation, a flexible ring check valve was installed in the Y block to isolate the inactive pump. This check valve is shown schematically in Fig. 47.

Because the check valve was difficult to keep tight, this device was dispensed with and the Y blocks mounted directly to the pump discharge fitting. The fuel pump was then supplied with an eccentric cam

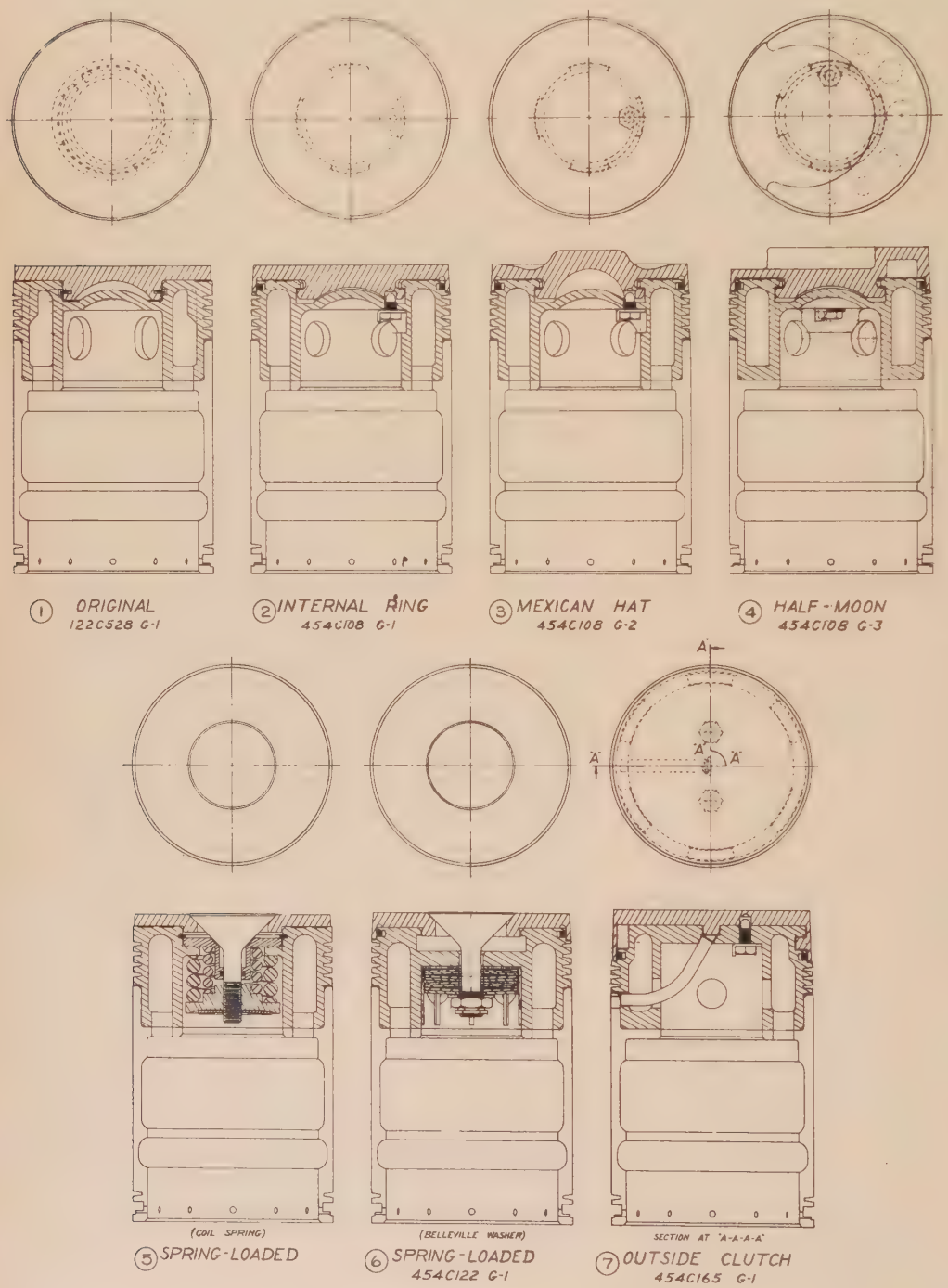


Fig. 39—Hot-crown piston-design types

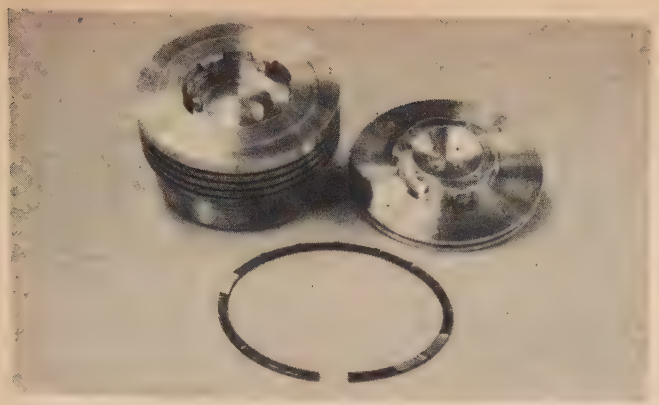


Fig. 40—Design of piston ring to maintain pressure seal between crown and piston skirt

Fig. 41—Redesign of hot crowns, moving clutch to outer diameter

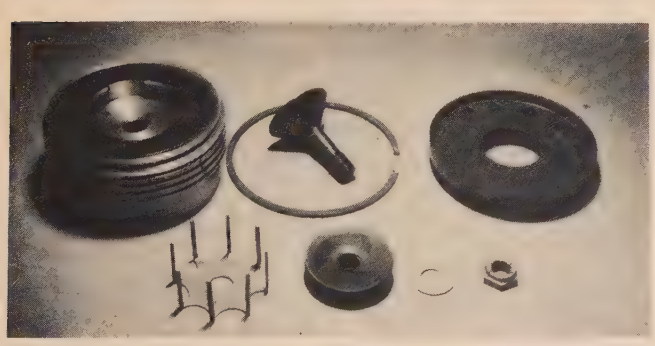


Fig. 42—Redesign of spring-loaded crowns to include Bellville spring washers

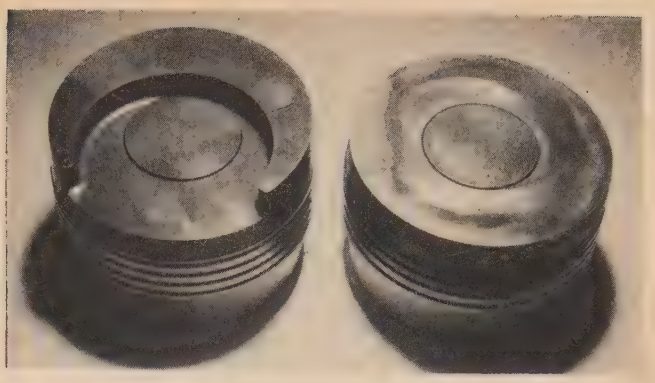


Fig. 43—Mexican hat and half-moon hot-crown piston designs

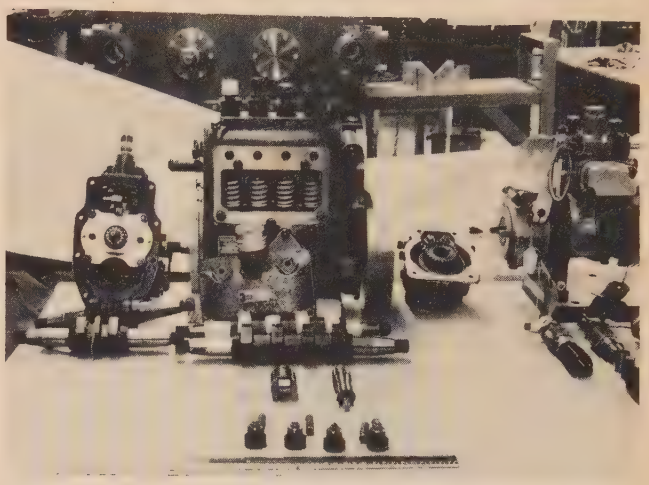


Fig. 44—Fuel-injection pump designed according to specifications

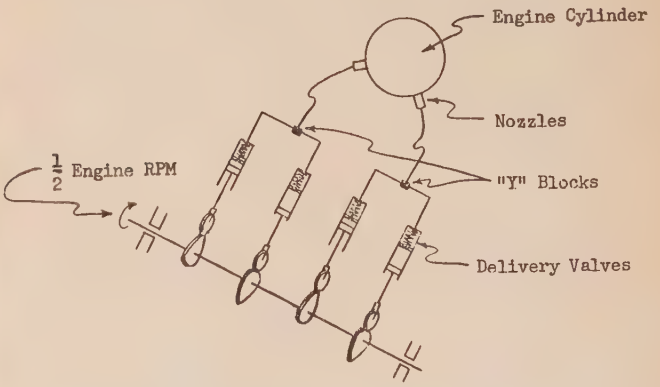


Fig. 45—Schematic drawing of fuel pump, line, and nozzles

that could rotate at full engine speed, thereby having two plungers feeding the same nozzle. This should have reduced the theoretical duration by one half. On testing, the results were not significantly changed. Because it did not seem possible to obtain an automotive-type pump small enough to fit into the space that was allocated to it in the Rigel engine design, it was deemed necessary to design one that would fit using as many standard parts as possible. For this reason a single-cylinder test pump was designed and put on test. This pump is shown in Fig. 48. At about this same time, failures of the roller tappets were becoming serious and from this was developed the flat tappet with a highly polished carboloy button brazed to the tappet guide face. Fig. 49 shows the button as the working surface. These buttons have been both bench

Fig. 46—Earliest design of Orion fuel-injection nozzle

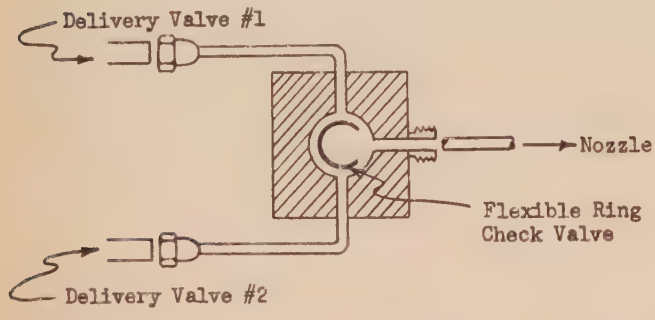


Fig. 47—Schematic diagram of flexible ring check valve

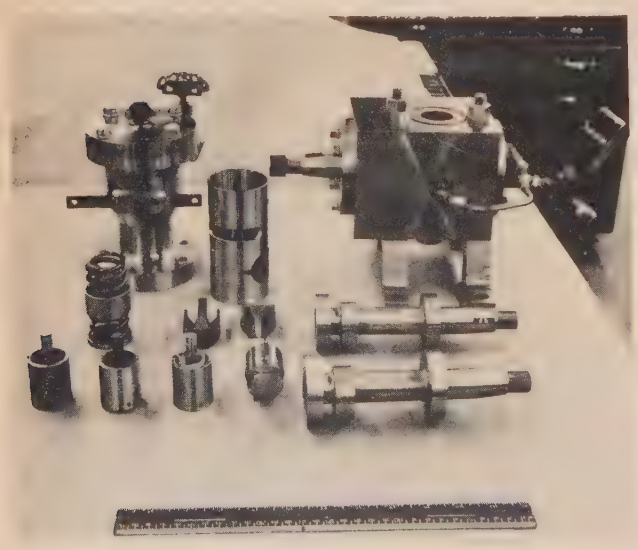


Fig. 48—Single-cylinder test pump

Fig. 49—Carboloy button as working surface

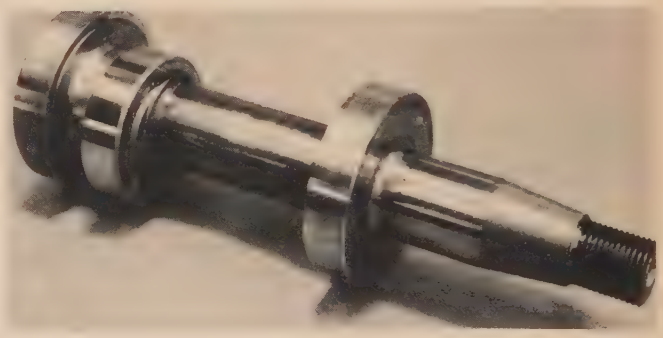


Fig. 50—Needle bearing on eccentric cam with flat tappet

Fig. 51—Slipper shoe

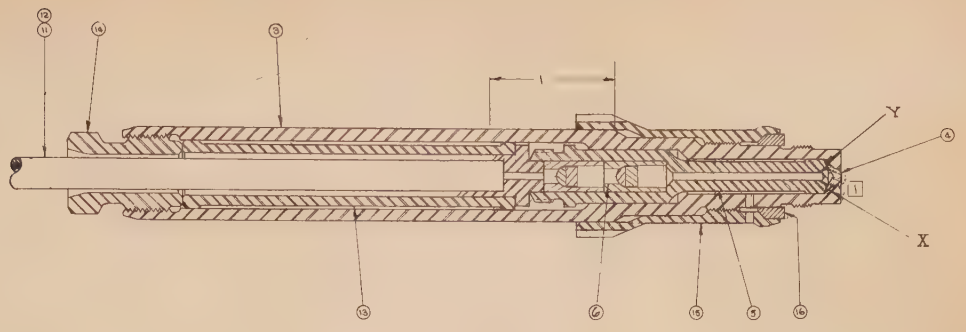


Fig. 52—Fuel injector with ball check valves in nozzle and removable tip for changing spray pattern

(1) (ASSEM. USING $\frac{1}{16}$ I.D. DISCHARGE TUBING)

(2) (ASSEM. USING $\frac{3}{32}$ I.D. DISCHARGE TUBING)

NOTE-

LAP MATING SURFACE "X" OF PTS. 3 & 4 AND MATING SURFACE "Y" OF PTS 4 & 5 AS FOLLOWS.-

1. CAREFULLY APPLY A FINE ABRASIVE COMPOUND TO SURFACE "X" (OR "Y")
2. INSERT NOZZLE INTO NOZZLE ADAPTER, USING LAPPING TOOL WHICH THREADS ONTO TOP OF NOZZLE, AND PROCEED TO LAP USING AN OSCILLATING ROTARY MOTION AND FREQUENTLY LIFTING & TURNING NOZZLE TO CHANGE RELATIVE INDEXING OF SURFACES BEING LAPPED. CONTINUE UNTIL SMOOTH & EVENLY GRAY SURFACES ARE OBTAIND AT "X", REMOVE PIN 10 AND LAP PTS 4 & 5 TOGETHER BY A SIMILAR PROCESS
3. DISASSEMBLE, THROUGHLY CLEAN AND MATCH MARK PARTS AT LOCATIONS INDICATED

CALIBRATE COMPLETED FUEL INJECTOR ASSEM. USE WATER AT 300 P.S.I.G. PRESSURE AND MEASURE & RECORD THE RESULTING FLOW. (SPRAY PATTERN MUST BE STEADY AND EVEN.)

tested and engine tested for several hundred hours with good results.

Before the troubles were licked several other designs were tested which worked very well. The use of a needle bearing on the eccentric cam with a flat tappet worked very well. This is shown in Fig. 50. Also tested with success was a slipper shoe shown in Fig. 51.

With the tappet problems solved, nozzle duration continued to be too long, and effort was concentrated on this problem. Pump barrels then proved to be the next weak link and newer special heavy walled barrels were obtained. Continuing with the pump experiments, ball check valves were next added both to the pump as well as in the nozzle. Ball check valves put in the nozzle are shown in Fig. 52. The check valves were inserted in the space formerly used by the edge filter. In addition, a removable tip for changing the spray pattern is shown in Fig. 52. When these check valves were installed, the duration was about 45 deg with a fairly sharp cut-off. Their life expectancy in an engine was only a few hours when the check valve springs would fail. The shortest duration obtained was about 35 deg after much delicate hand fitting and sculpturing of parts.

Excello-type nozzles shown in Fig. 53 were also used, and while their duration was the shortest (20 to 30 deg) of any nozzle tested up to this time, the life expectancy was short because the pintles would break off. Not only would the engine's performance falter, but some cylinder scoring was evident after pintles had broken.

Bosch nozzles were next tried, and their performance was very good except for sticking needles. This

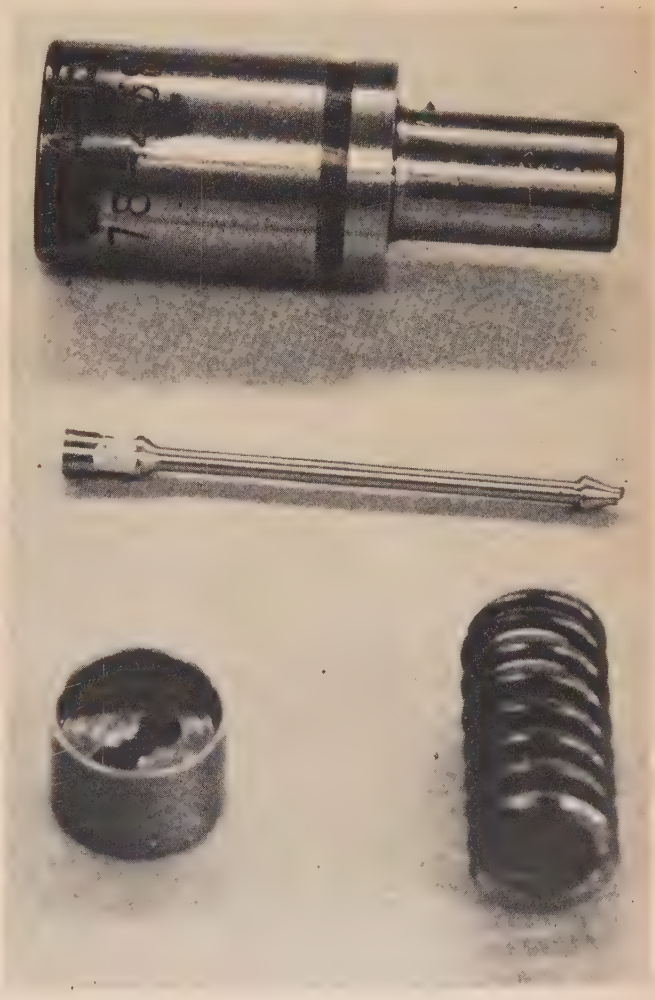


Fig. 53—Excello-type nozzles which proved unsuccessful

nozzle is shown in Fig. 54. Incidentally, this was one of the first nozzles used with the "pear-shaped" pistons and gave a specific fuel consumption of about 0.39. While the engine performance was good, the nozzle needle seemed to be too delicately fitted to be foolproof, and in addition required a fuel leak-off line which added to the plumbing.

Because the Excello-type nozzle seemed to have a certain merit, an oversized version, shown in Fig. 55, was designed and tested. This nozzle when bench-tested duplicated within 5% the performance of the Bosch nozzle in regards to flow and duration. The engine tests were short-lived because, both times the tests began a fuel leak developed and had to be stopped. At this time an entirely new fuel nozzle had been both bench tested and engine tested. So many other problems were solved that further work on other nozzle designs was stopped.

All fuel-injection systems up to the time of adopting the "accumulator" principle were of the "jerk" pump type of injection. Because the speeds were high, and the fuel flow per injection was beyond normal experience, the impact loading on the driving system were rapidly beating such things as gears, bearings, and splines to death. It was a combination of reasons that caused the author to resurrect the accumulator nozzle from the "free-piston" injection system he had used with success about 15 years previously on a project for one of the large aircraft engine companies.

During the initial developmental testing of "Orion," it became evident that a broad investigation of fuel-injection systems was warranted to attempt to establish high engine performance with a single fuel-injection nozzle. This discussion constitutes a memorandum of the design of one of the most satisfactory systems derived, the accumulator nozzle system.

The objectives of this design study may be listed as follows:

1. To reduce the duration of fuel injection into the cylinder for a given quantity of fuel.
2. To isolate from the injection process the influence of the volume of tube between pump and nozzle.
3. To increase the fuel pressure, on which control of injection is dependent, without overloading the pump-drive mechanism.
4. To increase initial fuel-flow rate so that a greater per cent of fuel might be burned at a constant combustion-chamber volume.
5. Eliminate pump delivery valves.
6. Use an eccentric cam.
7. Extend the pumping over a larger camshaft angle.
8. Eliminate the need for a leak-off line.
9. Use somewhat smaller injection tubes since the charging rate is much reduced.

The principle of the accumulator system is that the stored energy in the compressed fuel provides the potential energy for injecting the fuel into the combustion chamber. The method of operation can be deduced from Fig. 56. As the pump plunger reciprocates through the total stroke as indicated, the helix at the top of the plunger will cover the filling-line hole, trapping a volume of fuel. The trapped volume of fuel is a function of the angular position of the helix. As the plunger continues to lift, the fuel is pumped through the injection line into the nozzle body. Pumping continues until the spill edge at the bottom of the plunger matches the spill hole as shown, at which time the pressure is relieved in the plunger-barrel volume and the injection line.

Initially, spring S holds the needle on its seat at the nozzle tip, and the cuff on its seat at the nozzle top. When pumping starts, the cuff is lifted from its seat, and fuel flows into the nozzle body; in accordance with the compressibility, the pressure of the fuel within the nozzle is increased. When fuel delivery is interrupted by spilling of the injection line, the cuff returns to its seat retaining the pressure within the nozzle. Since the fuel pressure (P_n) times the area ($A_2 - A_3$) is greater than the spring force (S), the needle is thereby forced upward opening the spray orifice for the injection of the fuel.

Flow continues until the pressure within the nozzle body is less than that required to overcome the spring force, at which time the needle closes. The cycle is then complete and ready to be repeated.

While the quantity per injection is a function of the plunger angular position as stated above, the beginning of injection is controlled by the position of the "spill edge" on the plunger. An exploded view of the accumulator nozzle is shown in Fig. 57.

The objectives listed previously were real, and each was obtained. There is only one critical dimension that has to be closely controlled—the lift of the needle. The peak injection pressure used was about 14,000 psi, but pressures well over 20,000 psi have been tested. The pressure-time diagram of the nozzle and engine cycle are shown in Fig. 58. When the spill edge of the plunger dumps the line, a "sledge hammer" blow accelerates the needle opening to the extent that, if the lift is too large, the resulting velocity of the needle will hammer it to pieces when it strikes the stop. To prevent this, the lift was limited to 0.009 in. on the subject design.

The fit between the upper needle diameter and the cuff were at first of very high quality, approaching that of the Bosch-type needle fit. This was considered necessary to assure good results on the first attempt. As the development continued, the needle-cuff fit was progressively opened up in increments of 0.0002 in. and tested after each step. At the speeds of our operation this clearance could be opened up to 0.0006 in. without affecting the nozzle's performance. Therefore, it can be stated that this nozzle design does not require the preciseness of quality control that other nozzles need. These nozzles have been operated on all 14 cyl of the Orion project for several hundreds of nozzle hours.

Before leaving the discussion on fuel nozzles it should be mentioned that it was felt there was a need

for a high-speed precision drill press in order to do justice to this program. At first the nozzle holes were very carefully placed and controlled, but with the final success of the combination of cylinder, "pear" or "spherical" shaped combustion chamber, and the accumulator nozzle, the nozzle holes became fire hoses in effect. The nozzle now has one 0.043-in. diameter hole, and it can be drilled on most any machine tool. This philosophy has been shown to be correct many times; when the proper answers are known then most anything will work. The Orion combustion system definitely does work.

Turbine and Air Compressors

The turbine and the air compressors were both of conventional design and will only be very lightly discussed. The turbine design is shown in Fig. 59 and consisted of a single-stage wheel with 86 buckets.

The turbine was designed for 600 hp at a peak

Fig. 54—Bosch nozzles

Fig. 55—Oversize version of Excello nozzle

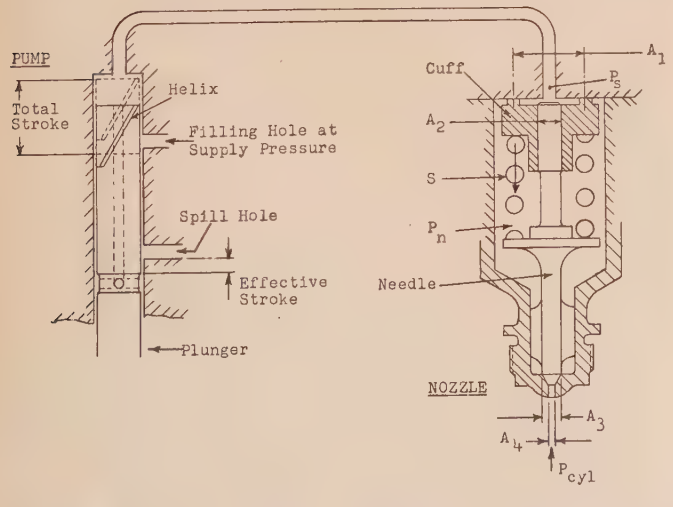


Fig. 56—Schematic diagram showing operating method of accumulator system

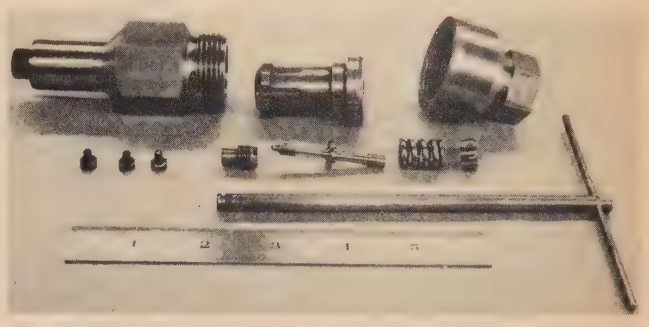


Fig. 57—Exploded view of accumulator nozzle

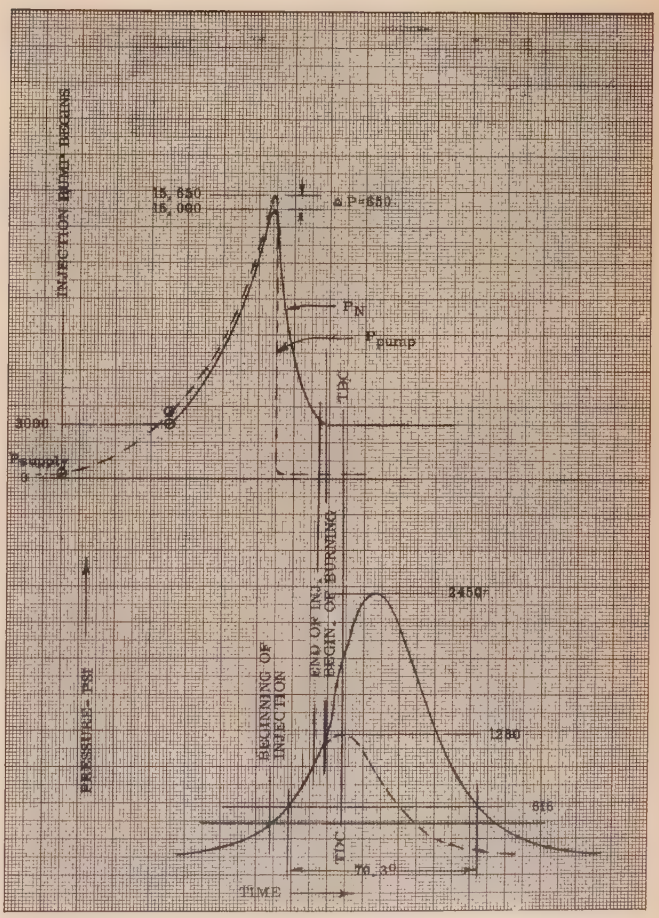


Fig. 58—Pressure-time relation between injection nozzle and power cylinder

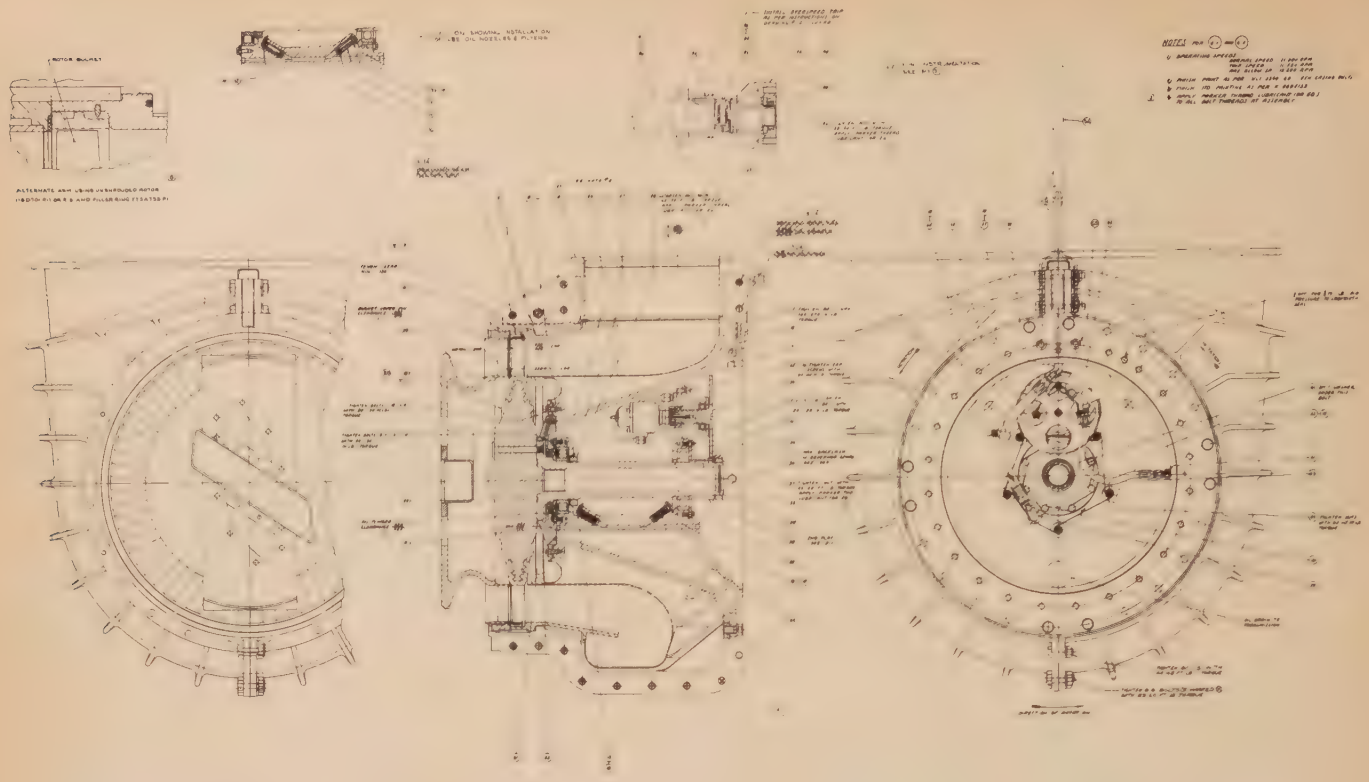


Fig. 59—Turbine design for Orion project

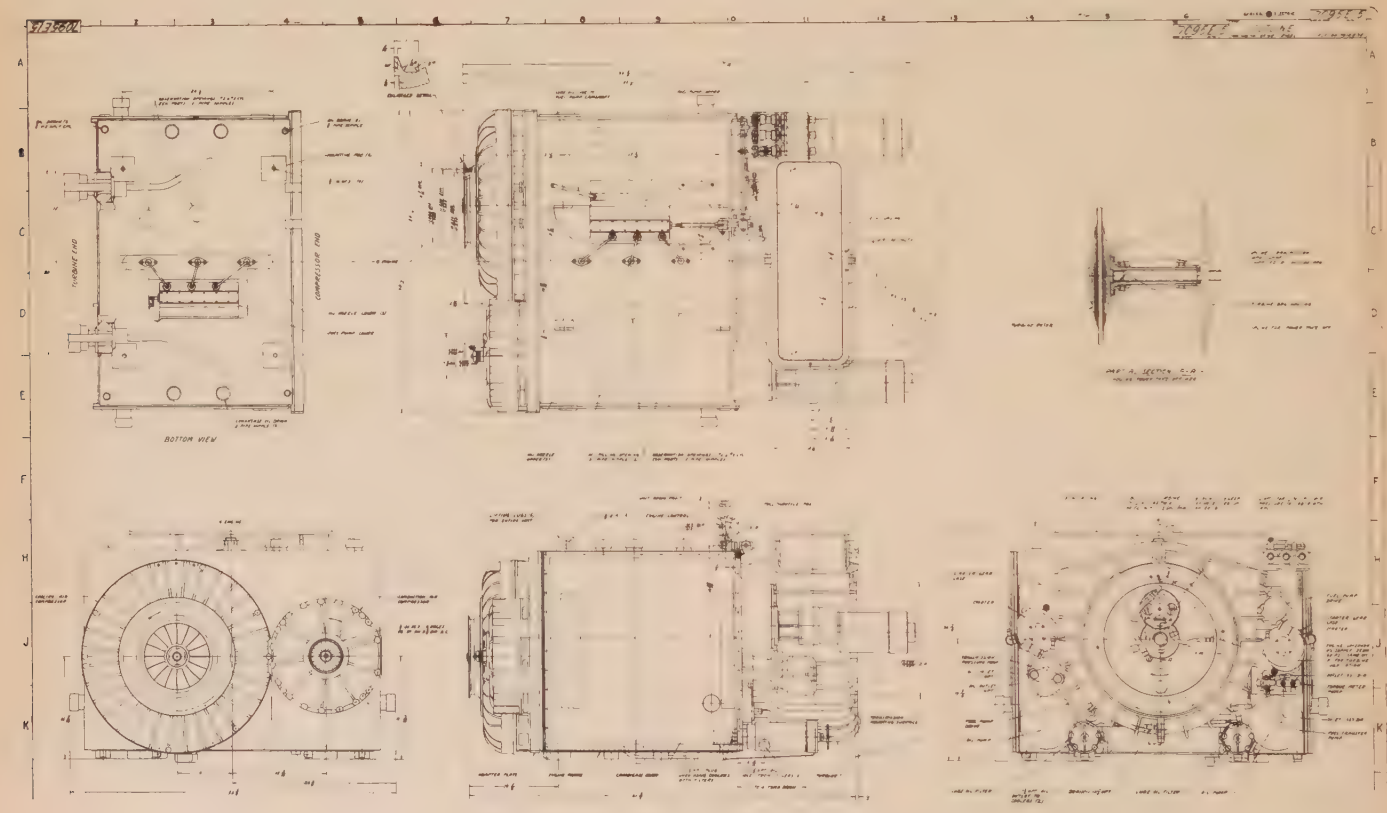


Fig. 60—Turbine assembly on Rigel engine

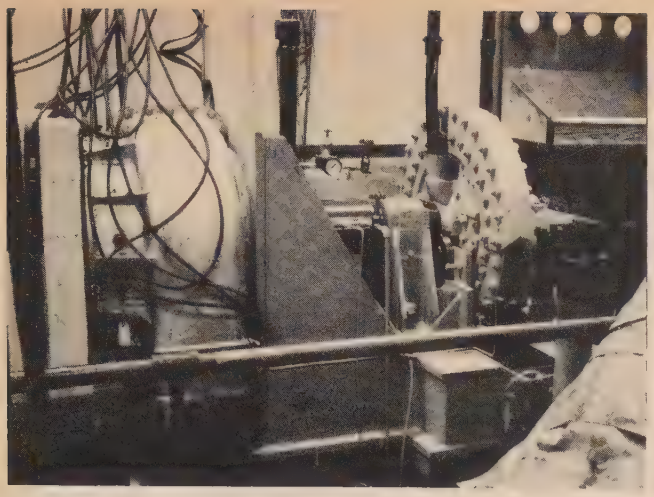


Fig. 61—Turbine steam test installation

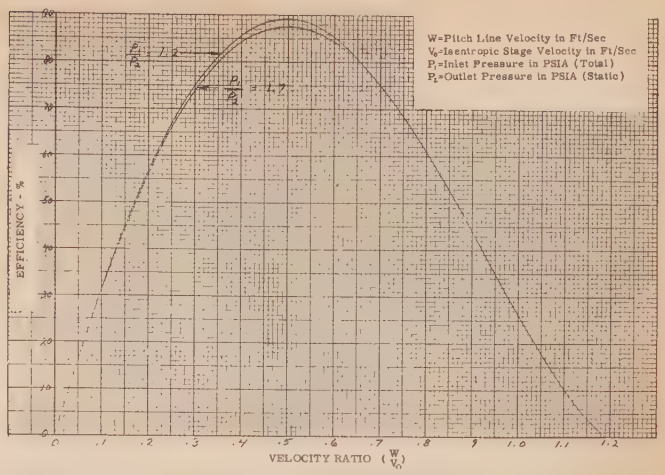


Fig. 62—Turbine steam test efficiency versus velocity ratio

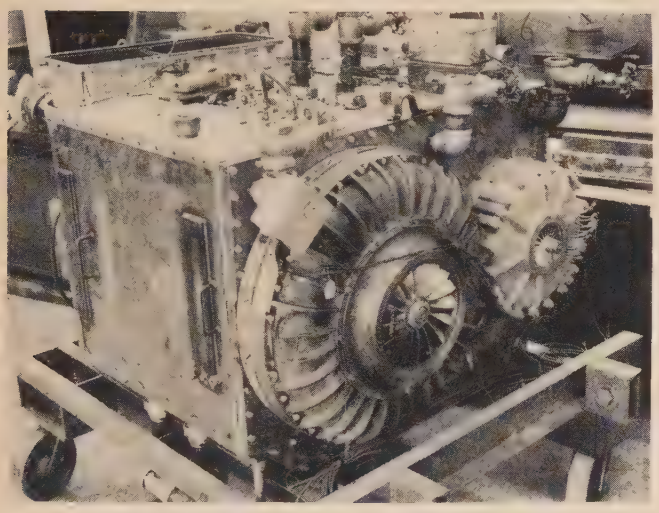


Fig. 63—Engine showing attachment of two air compressors

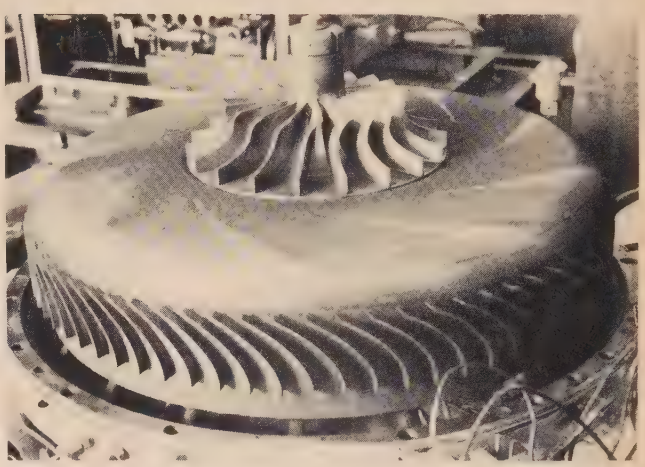


Fig. 64—Dewhirler vane design on compressor

turbine efficiency of 85% with gas conditions as follows: (1) Flow = 15.1 lb per sec, (2) Inlet pressure = 27.0 psia, (3) Inlet temperature = 474 F, and (4) Exhaust pressure = 15.0 psia.

The layout speed was 9310 rpm. Since the turbine was designed for operation with a transmission having a number of shift stages, it was to operate from stall to some speed beyond peak efficiency. The maximum speed before shifting was established at 11,000 rpm. A maximum speed of 13,200 rpm was established as the allowable overspeed.

These buckets were designed for vortex distribution of the tangential component of the nozzle velocity, and the bucket section was designed on laminar flow principles. The bucket was attached to the wheel by a standard GE 3-hook, circumferential type which was used because tooling was available to produce it at low cost. It was adequate for the job because the stress and temperature were low, and it had good mechanical damping.

The turbine assembly was steam flow-tested prior to its assembly on the Rigel engine. The turbine assembly is shown on the Rigel engine in Fig. 60. The turbine steam test installation is shown in Fig. 61.

Fig. 62 shows the turbine efficiency based on steam test results. The curve gives stage efficiency versus velocity ratio for various pressure ratios from nozzle inlet to bucket outlet annulus.

The difference in performance between operation with steam and with gas from a gas generator should basically be very little because the conditions under which the turbine ran in each case resulted in almost the same Mach number and within 50% of the same Reynolds number.

These results were verified or bettered when the turbine was tested on the Rigel engine. These test results are discussed later under Rigel Engine Test.

The attachment of two air compressors to the engine is shown in Fig. 63 and they are driven by the synchronizing gear train as shown in Fig. 10.

Since the General Electric Co. had built a supercharger with about the right airflow and pressure ratio

versus speed characteristics, identified as CHM 2D, it was decided to modify it rather than design a new impeller, thereby saving considerable time and expense. There are two design variants on both of the compressors, one was the dewhirler vanes and the other was the coned impeller to shaft, seating surface. However, it was unnecessary to use the coned impeller on the cooling air compressor. These vanes reduced the tangential component of the air velocity to an axial component so that the vortex velocity would not increase as the air flowed radially inward to the discharge opening of the compressor. These vanes are shown on Fig. 64 and are similar for both the cooling air and the combustion air compressors. The coned seating surface was added when it was found that, due to radial expansion of the impeller back face, an eccentricity developed resulting in enough unbalance to cause a rub. In order to maintain an axial force by elastic means, a stack of Bellville washers was compressed to an initial deflection. This more than compensated for the reduction in axial length as the impeller climbed up the coned ramp.

These Bellville washers are shown in Fig. 65 and are similar for the two impellers.

The combustion air compressor had to be designed, as there was no compressor on the shelf that could be used by revision.

The combustion air-compressor impeller was an investment type of aluminum casting including a shroud as shown in Fig. 66.

This compressor was designed for a flow to satisfy the additional requirement of exhaust port bridge cooling air. When this cooling airflow was no longer necessary, the compressor was reduced in capacity by turning off the shroud as shown in Fig. 64.

The cooling air compressor performance is shown in Fig. 67, but a complete map was not obtained for the final configuration. A complete map of the combustion air compressor is shown in Fig. 68. The compressor specifications are given in Table 4.

Rigel Engine Test

Rigel I was groomed for installation in the test cell as shown in Fig. 63. The installation was first made without the turbine wheel and water brake (which was still in use for steam-testing the turbine). The turbine nozzle was the resistance against which the gas generator was running. The engine was first started on Saturday afternoon, Jan. 30, 1954. The turbine wheel was added in September, 1954. The idea was later conceived that some information would be obtained of the joint turbine and gas-generator operation by measuring the stall torque. The rotor was locked using a Hagan Thrus-Torq. The stall-torque readings could give some indication of the turbine performance were it allowed to drive a load.

After the turbine steam tests were completed, the water brake was installed in May 1955 and the turbine power was measured for the first time as a part of a complete gas generator—gas turbine powerplant. The installation is seen in Figs. 69 and 70. During this period of time the entire program of cylinders,

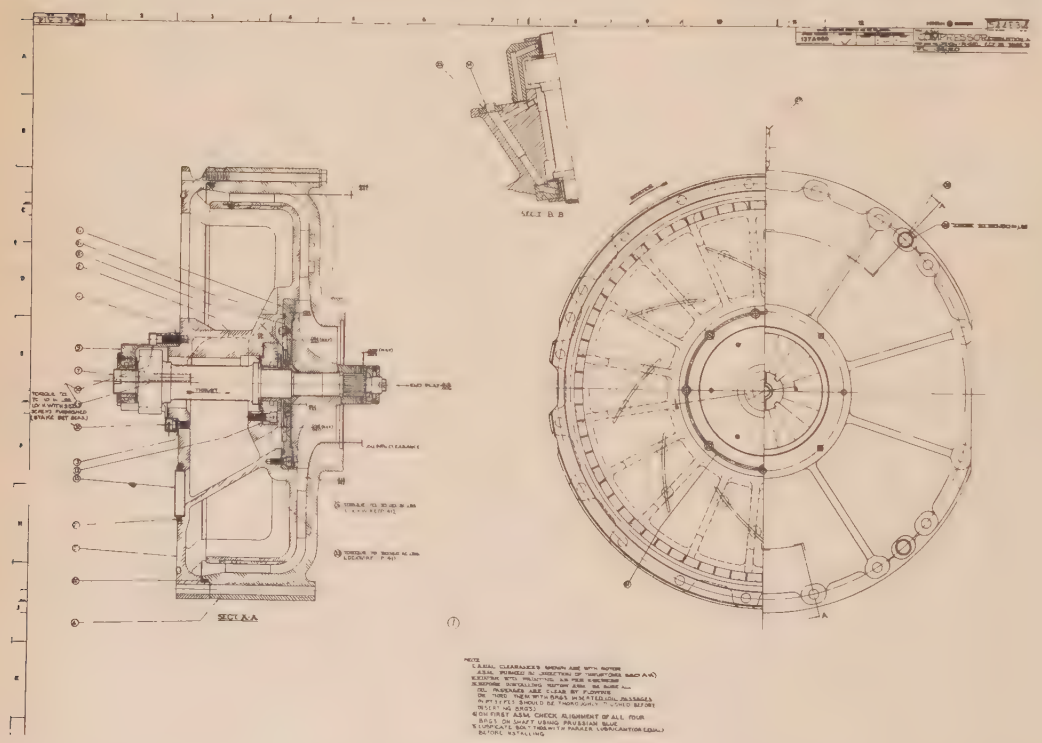


Fig. 65—Design of Bellville washers

pistons, injection system, and the turbine and air compressors were still being tested and revised. Consequently, the operation became quite involved at times.

In June, 1955, the Rigel II engine was revised to include all the latest development parts, and its tests began to click in practically every respect.

This engine assembly was able to operate at full gas-generator power and ran continuously at full load of 2340 rpm for 6½ hr in one day. The next day running was continued until a total of 10 hr 13 min were logged. From this time on it could be said that testing had started. The statement can now be made, that to develop a piece of machinery it must first be able to run mechanically before the thermodynamic cycle, the life, or other improvements can be made. It is still a proved fact that things must be measured before they can be understood. During the operations of the Rigel engines, most every conceivable failure occurred and some happened that couldn't happen. For instance, the first time the Rigel engine was started these were, main bearing failures, fuel pump drive bearings that failed, electric starter gear teeth

Table 4—Compressor Specifications

	Combustion Air Compressor	Cooling Air Compressor
Weight Flow, lbs per sec	3.15	12.77
Pressure Ratio	2.34	2.11
Static Efficiency %	80%	81%
Inlet Pressure, psi	14.1	14.1
Inlet Temperature, F	80	80
Outside Diameter of Impeller, in.	7.15	14.6
Rpm	37,900	18,050

Fig. 66—Combustion air compressor impeller

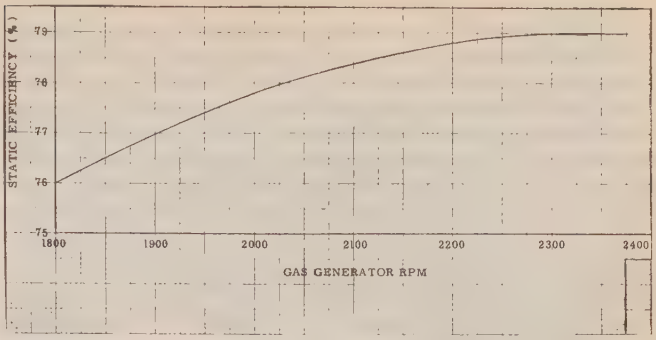


Fig. 67—Cooling air compressor static efficiency versus gas generator rpm

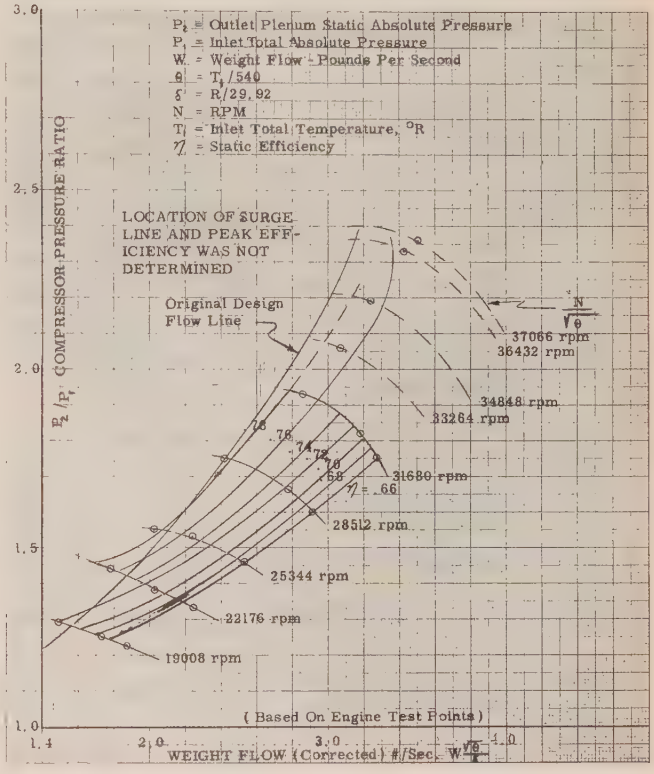


Fig. 68—Combustion air compressor with cut-down impeller

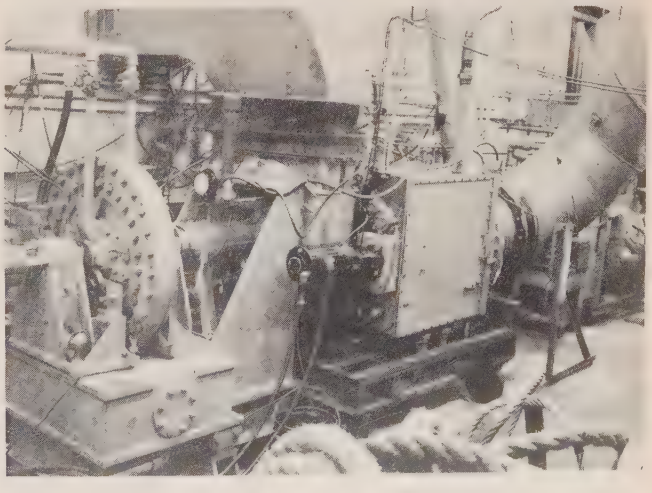


Fig. 69—Installation of water brake

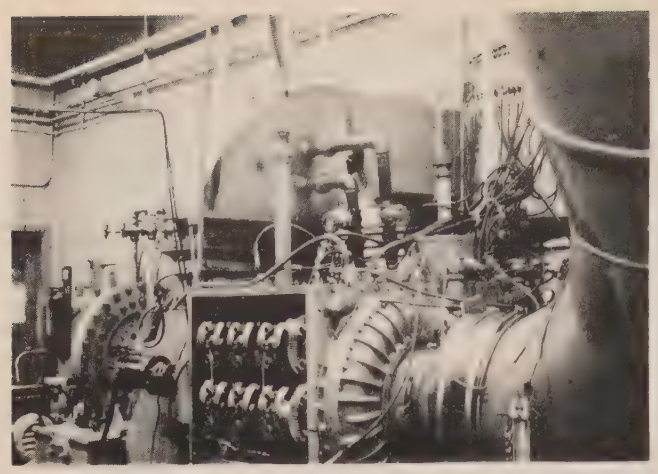


Fig. 70—Engine from turbine end showing installation of water brake

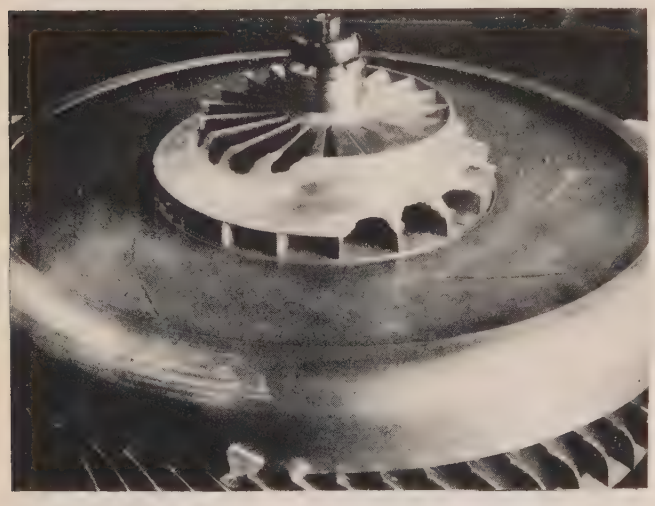


Fig. 71—Failure of compressor impeller rubs

which chipped, chips in the oil lines, splined connections which fretted, and many other nuisance and time-consuming accidents. All of these were quite easily repaired.

Such failures as compressor impeller rubs would load the engine with aluminum dust requiring that it can be entirely torn down and cleaned. One such rub is shown in Fig. 71.

A lube oil pump gear scored and the driveshaft sheared off. This failure is shown in Figs. 72 and 73.

Several aluminum piston skirts seized with the results as shown in Fig. 74. This also filled the engine with aluminum dust requiring a complete teardown for cleaning.

Another cause of an engine shutdown was the first and only failure of the center fin section as shown in Fig. 75. This fin section was a revised standard steel cylinder center fin section which had been bored out to take the cast ductile iron cylinder. The outside diameter of the fin had been reinforced with a shrink band to take the load, but the high stress concentration of the nozzle hole was just too much for the material.

Fig. 72—Scored lube oil pump gear

A most unusual failure occurred when the nut locking the cylinder to the frame backed-off. At first this spanner nut was wired with 0.035 soft steel wire; the wire failed, and it was increased to 0.070 in. This wire also failed, and a tab lock as shown in Fig. 76 was finally added. The cross-section of the tab lock shown at the arrow was about ½ in. wide x $\frac{3}{16}$ in. thick. This tab lock was broken, and the nut backed off allowing the cylinder to move endwise about ¼ in. There was a lot of smoke at this time, but not too much damage as the exhaust gases would blow into the crankcase showing up as excessive blow-by. This failure was never explained nor completely solved when the project was completed. All of this happened on one cylinder.

The results of the final tests on the Rigel engines of the "Orion" project are shown on the curves that follow.

The turbine horsepower versus turbine speed is given in Fig. 77. The test points are indicated by the small circles.

The transmission characteristic was supplied by the Arsenal and is a 3-speed transmission with a splitter gear, making six speeds in all. A map based on engine test data and transmission torque multiplication characteristics is shown in Fig. 78.

To allay the concern about cylinder wear versus wall temperature, a controlled test using six new cast ductile iron cylinders was run with the results shown in Fig. 79.

A complete set of curves, showing the calculated test results of the component performance of the Rigel engine are as follows:

The curves of efficiencies, Fig. 80, show how well the compressor static efficiencies have held up as speed

Fig. 73—Failure showing driveshaft which sheared off

Fig. 74—Failure of aluminum piston skirts

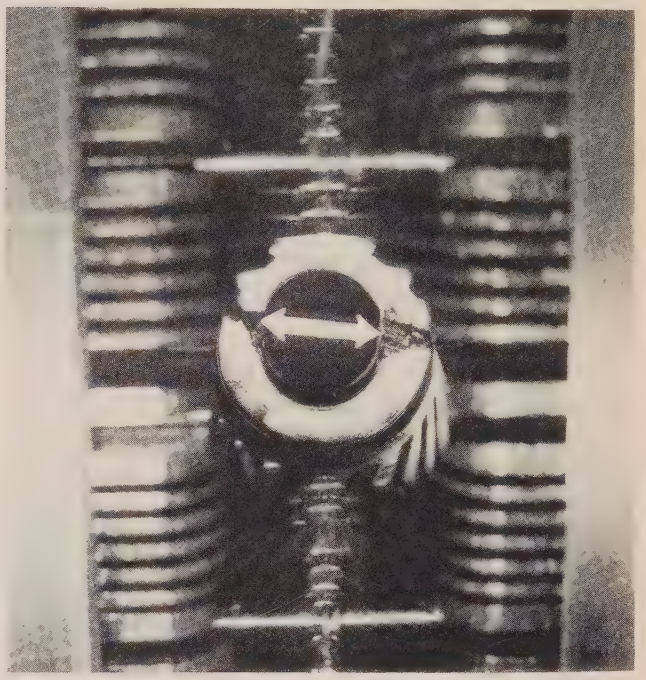


Fig. 75—Failure of center fin section due to materials failure

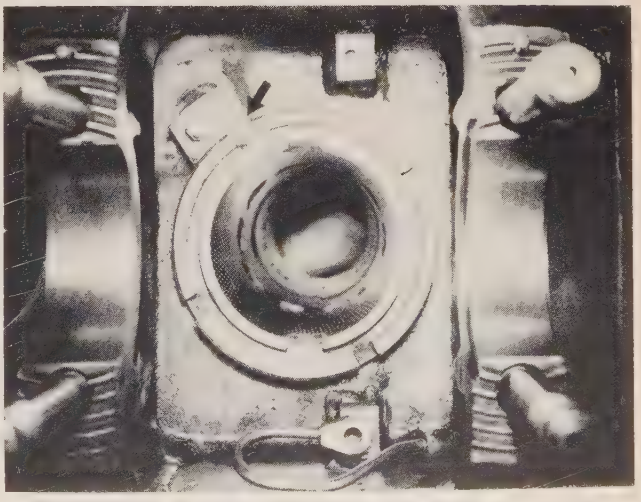


Fig. 76—Broken tab lock which resulted in another type failure

was increased. The combustion air compressor efficiency, η_a is low because the compressor was operating at a flow higher than design. This was due to the cylinder flow area being larger than that planned for a subsequent cylinder modification designed to trap more air in the cylinder for combustion. While the turbine efficiency is good at about 2100 rpm, the peak point should be shifted to the right by additional trimming and elimination of fouling.

The curve of turbine specific fuel consumption is shown to be decreasing much in the same manner as in a normal engine. The availability of energy in the working fluid is increasing. The fuel consumption, based on the compressor power remains practically constant reflecting the constancy of the product of the efficiencies (η_d, η_m, η_c).

In Fig. 81, the increase in turbine inlet temperature with load is shown. The two lower curves show the increase in the "equivalent flow area" through the system which is a most complicated phenomenon and is influenced by temperature, pressure ratio, timing effects, and to some extent by the change in the constituents of working fluid.[1]

The turbine inlet pressure and airflow, Fig. 82, are increasing functions of load and follow a quite normal path.

Fig. 83 shows the division of work between the two air compressors and the turbine output. It is ap-

1 "Steam Turbines and Their Cycles," by J. K. Salisbury, pp. 212–218. Pub. by John Wiley and Sons, Inc., New York, 1950.

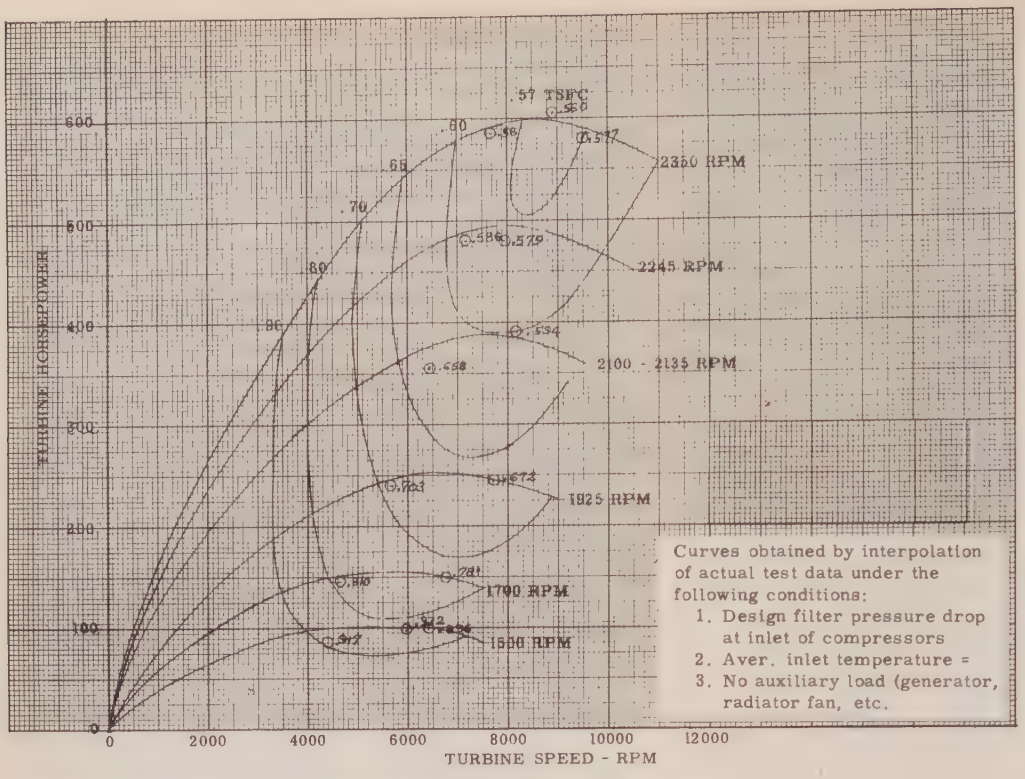


Fig. 77—Turbine horsepower versus turbine speeds at various gas-generator speeds

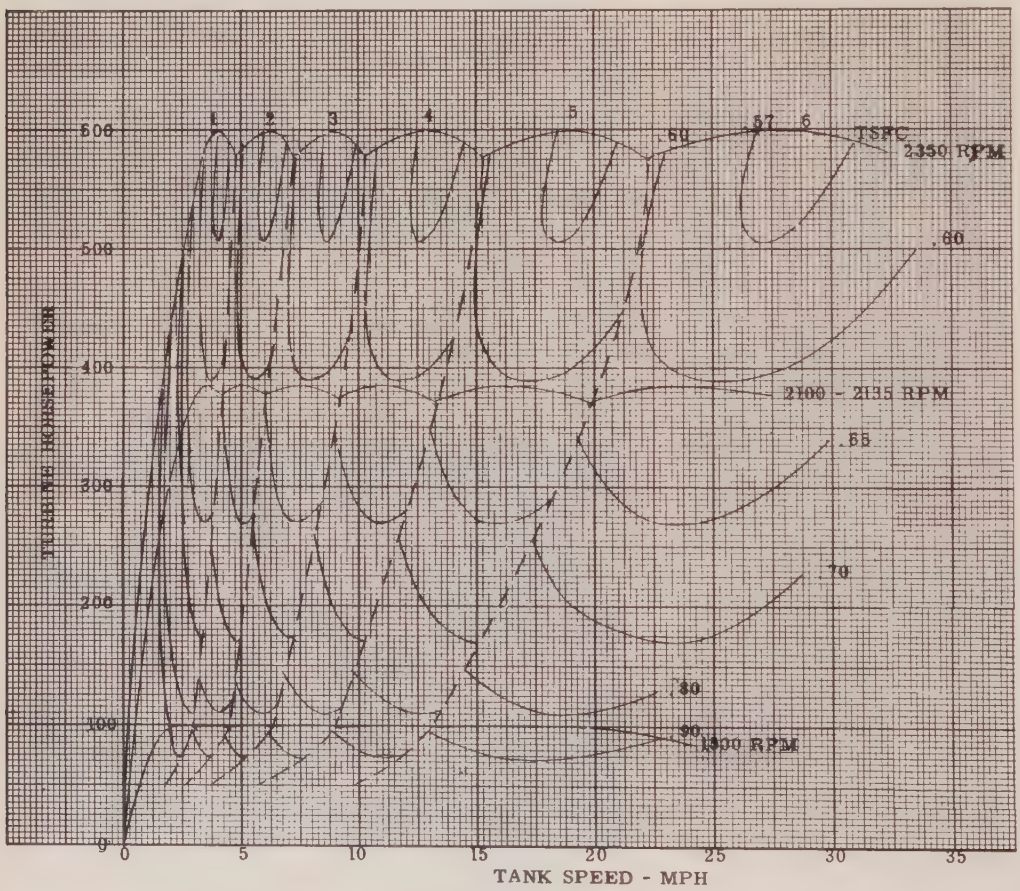


Fig. 78—Rigel performance for XTF-1400 transmission. Graph shows horsepower input to transmission versus tank speed with lines of constant specific fuel consumption

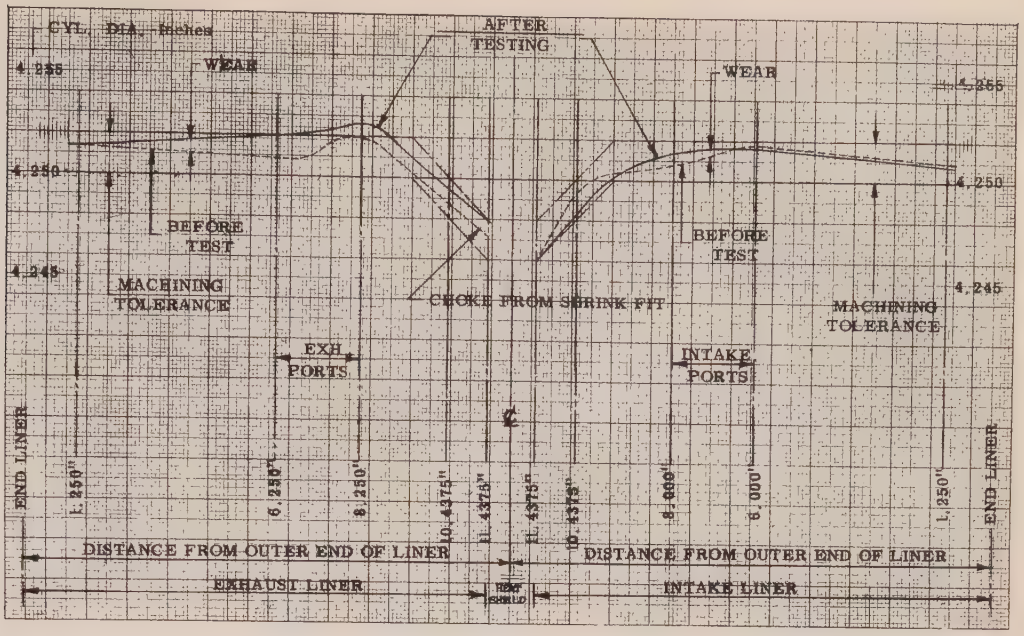


Fig. 79—Average wear on six cast ductile cylinder liners of Rigel II engine after 75 hr of operation

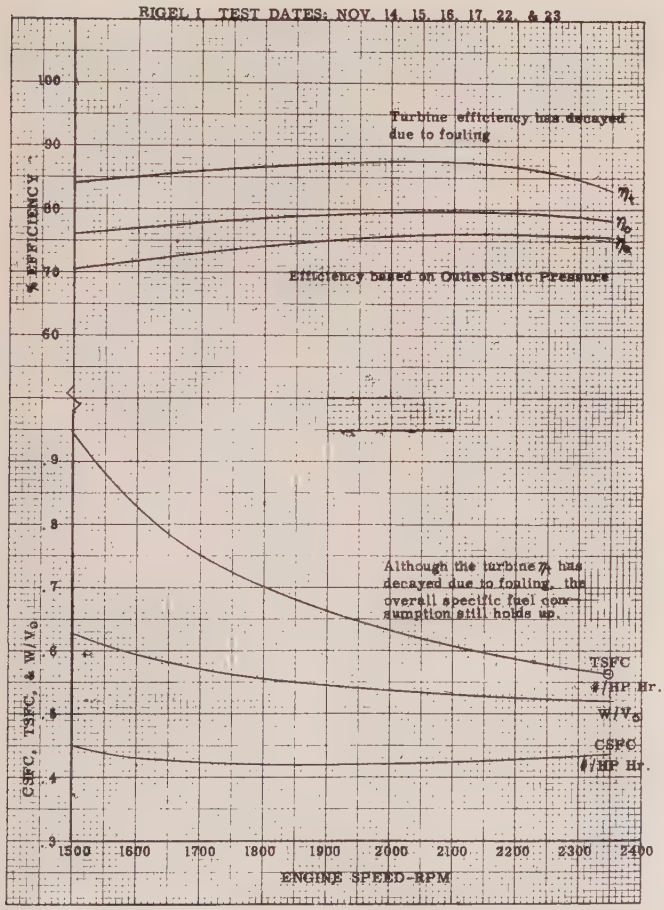


Fig. 80—Curves of turbine efficiencies and specific fuel consumption

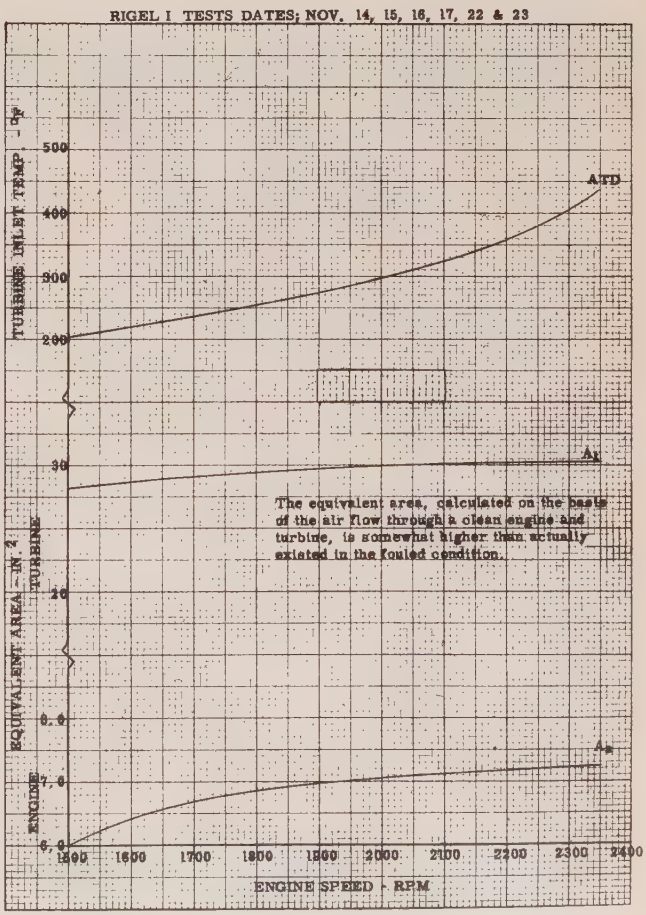


Fig. 81—Increase of turbine inlet temperature and equivalent flow area with increasing engine speed

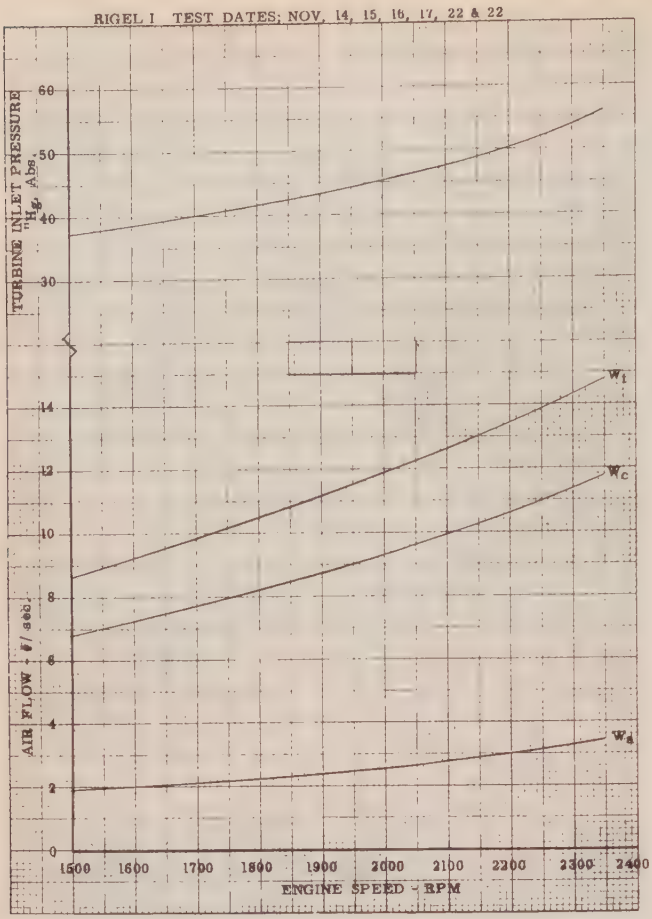


Fig. 82—Increase of turbine inlet pressure and airflow with increasing load

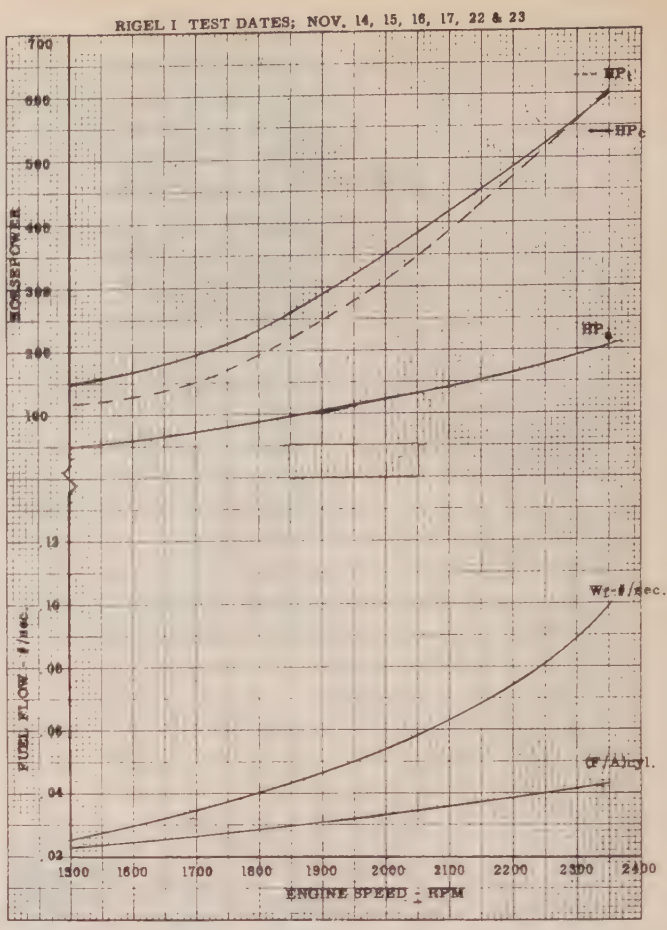


Fig. 83—Graph showing improved turbine performance due to increase of available energy

parent that as the load increases the turbine performance improves due to the increase of available energy of the working fluid as a function of the increased turbine pressure ratio.

The 'Heat Balance,' Fig. 84, shows the improvement in heat to turbine work as the load increases, due to the increase in availability.

Successful operation of the Orion airflow system is predicated upon being able to balance the flow of two independent airstreams. The flow path of each stream, one of combustion air and the other of cooling air, meet at the turbine inlet.

Fig. 85 shows a graphical presentation of the complete "Orion" cycle as it was calculated for one condition of the Rigel engines. In the central part of the chart is shown a simplified flow diagram. The pressure at point 5 is common to both streams otherwise there will be an irreversible energy loss.

The curve 1 is the combustion air compressor characteristic of pressure versus weight flow at the speed of 37,000 rpm. Curve 2 is the pressure P_5 common to both air streams at point 5. Curve 14 is the pressure at the entrance to the engine duct. Curves 3 and 4 are the air compressor horsepower and discharge temperature, respectively. The pressure drop from curve 1 to curve 2 is the summation of all the individual losses in the combustion air circuit.

In the cooling air circuit, the curve 5 is the cooling air compressor characteristic of pressure versus weight flow at the speed of 16,000 rpm. Curve 6 is the pressure P_5. The pressure drop from curve 5 to curve 6 is again the summation of all the component losses in the cooling air circuit. Curves 7 and 8 are the discharge temperature and the compressor horsepower respectively.

The total flow through the turbine is the sum of both airstreams; therefore, curve 9 is the combined pressure versus weight flow characteristic of the Rigel system at point 5. The curves 11 and 12 are the total combined compressor horsepower, and the combined mixture temperature of the two airstreams. Curve 16 is the turbine inlet pressure, after correction, for the duct loss from point 5 to point 7. Curve 13 is the turbine flow versus pressure characteristic.

Acknowledgment

I want to thank the General Electric Co., the engineers, laboratory technicians, and the supporting machine-shop personnel who made it possible to work

on and solve the many problems that arose on the Orion project.

APPENDIX

Cycle Analysis—Power Balance

Cooling the engine cylinder and exhaust port bridges was one of the recognized problems that would have to be given close attention. Study of this problem did achieve several advantages that were incorporated in the Rigel engine design and at the same time satisfied the specifications in a most elegant manner. This analysis concerns the division of power for pumping cooling air, as a function of the heat rejection and the balance of power left for supercharging the engine.

First, by pressurizing the cooling system, it was possible to depress the bmep, down to a value where it was believed that life expectancy of the thermally loaded parts of the engine would be increased. Balancing the engine power with cooling air compressor load at a relatively low supercharge pressure level permits the advantage of the independent gas generator at a low engine bmep. Other gas-generator cycles which have been investigated require a much manifold pressure and, consequently, a higher imep. Secondly, the heat lost from the cylinder walls and absorbed by the cooling air could be made to do work on the turbine, thereby reclaiming a certain return that would otherwise be lost as in the case of a fan-cooled cylinder. This reclaimed heat did in fact about balance out the pressure drop through the fins, and the resulting package surrounding the engine become a pressure vessel impervious to its surroundings and could be submerged without damage provided the air induction and exhaust gas systems were provided with "snorkels."

Fig. 86 is a rather interesting curve and from it can be estimated what the limitations are when air-cooling an engine.

For project Orion, the 2-cycle cylinder was not only supercharged, but was also aircooled. While air-cooling had important advantages for tank application, it imposed a greater limitation to cylinder temperature than would have been encountered with a liquid-cooled cylinder. If liquid cooling could be used in future 2-cycle engines, they could be supercharged to a higher level. This would achieve a considerable gain in the overall thermal efficiency. This is shown in the curve of "heat to turbine work," Fig. 84.

By supercharging the cylinder, power was increased

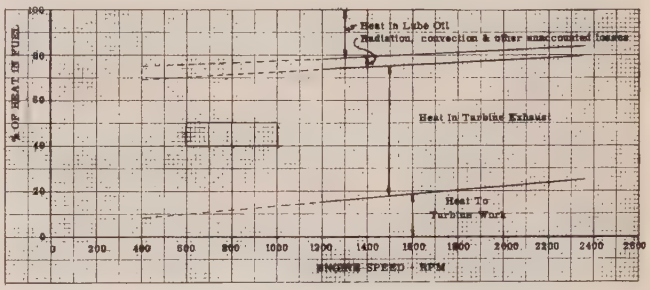


Fig. 84—Heat balance showing improvement in heat to turbine work with increasing load

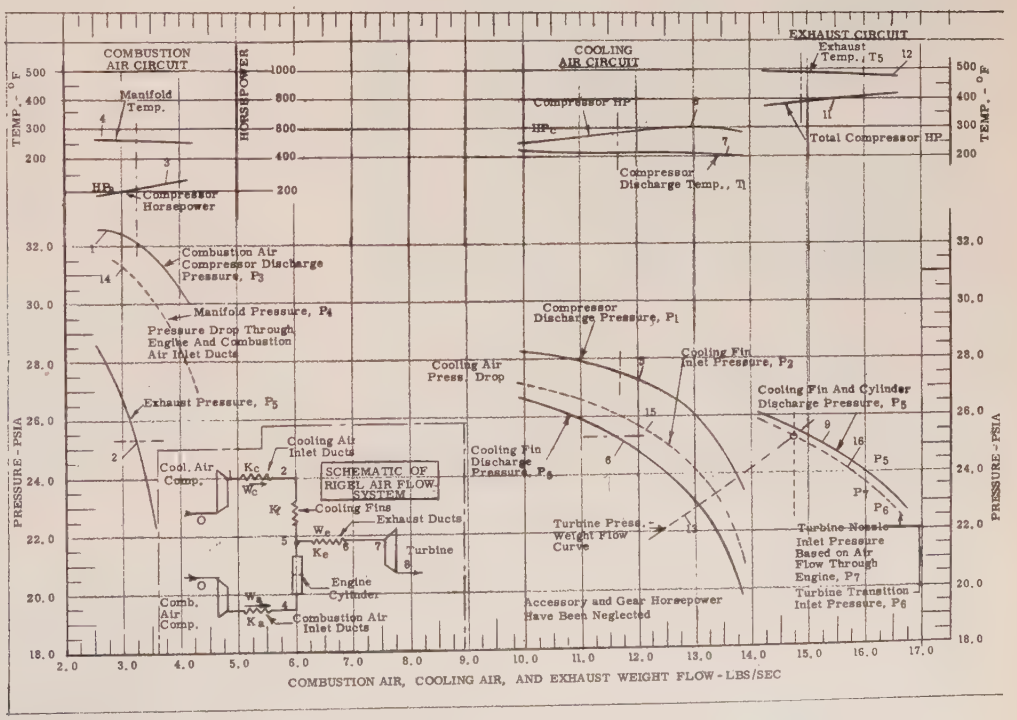


Fig. 85—Graphic representation of complete Orion Rigel cycle

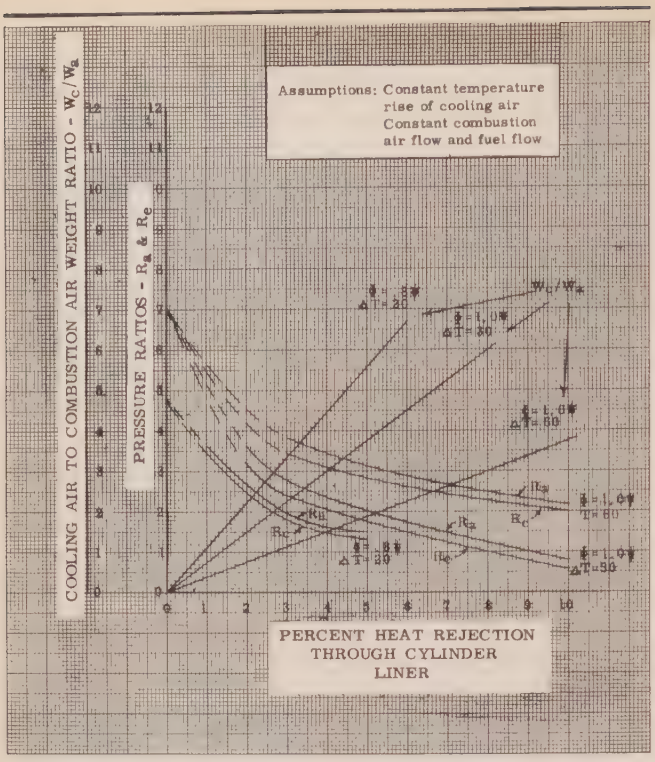


Fig. 86—Pressure and airflow ratios as function of cylinder heat loss for fixed temperature rise of cooling air

by a ratio of 2 to 1 over the unsupercharged engine. For purposes of design, an arbitrary temperature limit of 500 F at the upper piston ring was established. This was determined to be near the limit of wall temperature which could be maintained with pressurized air cooling. On the valve-in-head engine with atmospheric air cooling, the allowable supercharge level and the resulting mep's were even higher.

With a liquid-cooled engine, assuming that only the pumping losses require power, it could be supercharged in about the fashion as indicated by the 0% heat rejection on a curve about midway between the $\Phi = 1$ and $\Phi = 0.8$ curve.

The schematic diagram, Fig. 87, illustrates the different components and their relation to the other parts of the powerplant for analysis of the cycle.

The derivation of the cycle is as follows: (See Nomenclature for definition of terms.)

Total heat input to the system is

$$H_f = W_f h_f \tag{1}$$

The fuel flow can also be written in terms of the net or engine bhp, then

$$W_f = \frac{\text{bhp} \times \text{SFC}}{3600}$$

The engine output in enthalpy units is

$$H_e = W_f H \eta_a \eta_m \tag{2}$$

This output in turn is equal to the input to the two compressors and the auxiliary equipment.

Then

$$H_e = W_c \frac{k}{k-1} \frac{RT_o}{778\eta_c} \left[R_c^{\frac{k-1}{k}} - 1 \right] + W_a \frac{k}{k-1} \frac{RT_o}{778\eta_a} \left[R_a^{\frac{k-1}{k}} - 1 \right] + E \tag{3}$$

The heat energy going to the auxilliary equipment is defined as being equal to "y" % of the fuel input heat energy, then

$$E = W_f h_f \eta_a \eta_m y \tag{4}$$

Equation 3 can then be written as

$$H_e = W_c \frac{k}{k-1} \frac{RT_o}{778\eta_c} \left[R_c^{\frac{k-1}{k}} - 1 \right] + W_a \frac{k}{k-1} \frac{RT_o}{778\eta_a} \left[R_a^{\frac{k-1}{k}} - 1 \right] + W_f h_f \eta_a \eta_m y \tag{3a}$$

The combustion airflow consists of two components: one is the air trapped by the piston when it closes the ports, and the second is the scavenge air or that portion blown through the cylinder for purging the burnt products of combustion. The cooling airflow is that used to cool the engine cylinders and is based entirely on the heat rejection through the cylinder walls and the allowable temperature rise of both the wall and cooling air.

The trapped air for combustion without regard to engine geometry is

$$W_t = W_f \left(\frac{a}{f} \right) \lambda \tag{5}$$

and the scavenge airflow as a function of the air for combustion also without regard to engine geometry is

$$W_s = W_t \lambda_s \tag{6}$$

The total combustion airflow is then

$$W_a = W_t + W_s = W_f \left(\frac{a}{f} \right) \lambda + W_f \left(\frac{a}{f} \right) \lambda \lambda_s$$

Finally

$$W_a = W_f \left(\frac{a}{f} \right) \lambda \, (1 + \lambda_s) \tag{7}$$

The cooling airflow demand is a function of the permissible cylinder-wall temperature and the corresponding temperature rise of the cooling air across or through the cylinder cooling fins. The heat rejection through the cylinder wall is

$$H_r = W_f h_f x = W_c C_p \Delta T_{cc}$$

Then

$$W_c = \frac{W_f h_f x}{C_p \Delta T_{cc}} \tag{8}$$

The temperature of the cooling air compressor discharge is

$$T_c = T_o \left(\frac{R_c^{\frac{k-1}{k}} - 1}{\eta_c} + 1 \right) \tag{9}$$

and for the combustion air compressor

$$T_a = T_o\left(\frac{R_a^{\frac{k-1}{k}} - 1}{\eta_a} + 1\right) \qquad (10)$$

Note that both pressure ratios R_c and R_a must be corrected for a filter or duct drop or other losses that are peculiar to the particular installation. Then

$$R_c = \frac{P_c}{(P_o - \Delta P_{cF})}$$
$$P_c = P_t + \Delta P_{cc} \qquad (11)$$
$$\Delta P_{cc} = F(W_c^2)$$

and

$$R_a = \frac{P_a}{(P_o - \Delta P_{aF})}$$
$$P_a = P_t + \Delta P_{ac} \qquad (12)$$
$$\Delta P_{ac} = F(W_a^2)$$

Substituting equations 4, 5, 6, 7 and 8 into equation 3a will give the energy balance between the engine, the compressors, and the auxiliary load as a function of the airflow, pressure ratios, and the heat rejection. Also assume that both compressors have the same efficiency, then

$$W_f H_f \eta_d \eta_m (1 - y) = \frac{W_f R T_o k}{778(k-1)\eta_c}\left\{\frac{x h_f}{C_p \Delta T}\left[R_c^{\frac{k-1}{k}} - 1\right] + \left(\frac{a}{f}\right)\lambda(1 + \lambda_s)\left[R_a^{\frac{k-1}{k}} - 1\right]\right\} \qquad (13)$$

By consolidating all the constants, the balance equation between the compressors and the engine in terms of a parameter will be as follows:

$$\Phi = 778 h_f \eta_m (1 - y)\frac{(k-1)\eta_c}{kRT_o}$$

This can also be written in terms of bhp and ahp, where the ahp is the power to drive the auxiliary load. Then

$$\Phi = \frac{(bhp - ahp)}{bhp}\,\frac{k-1}{k}\,\frac{\eta_c \eta_m}{RT_o}\,\frac{3600 \times 550}{\text{SFC}} \qquad (14)$$

Finally

$$\Phi = \frac{(bhp - ahp)}{bhp}\,\psi$$

By equating Φ to the right-hand bracketed term of equation 13, a relation will be obtained for computing the pressure ratios as a function of the necessary cooling and combustion airflow.

$$\Phi = \left\{\frac{x h_f}{C_p \Delta T}\left[R_c^{\frac{k-1}{k}} - 1\right] + \left(\frac{a}{f}\right)\lambda(1 + \lambda_s)\left[R_a^{\frac{k-1}{k}} - 1\right]\right\} \qquad (15)$$

The ratio between the cooling airflow and the combustion airflow as a function of the heat rejection and the allowable temperature rise can be derived

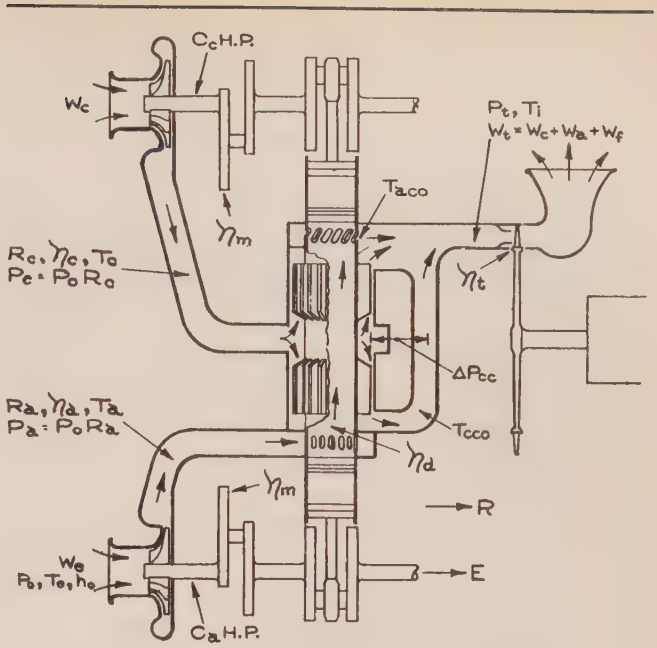


Fig. 87—Diagram of Orion thermal cycle

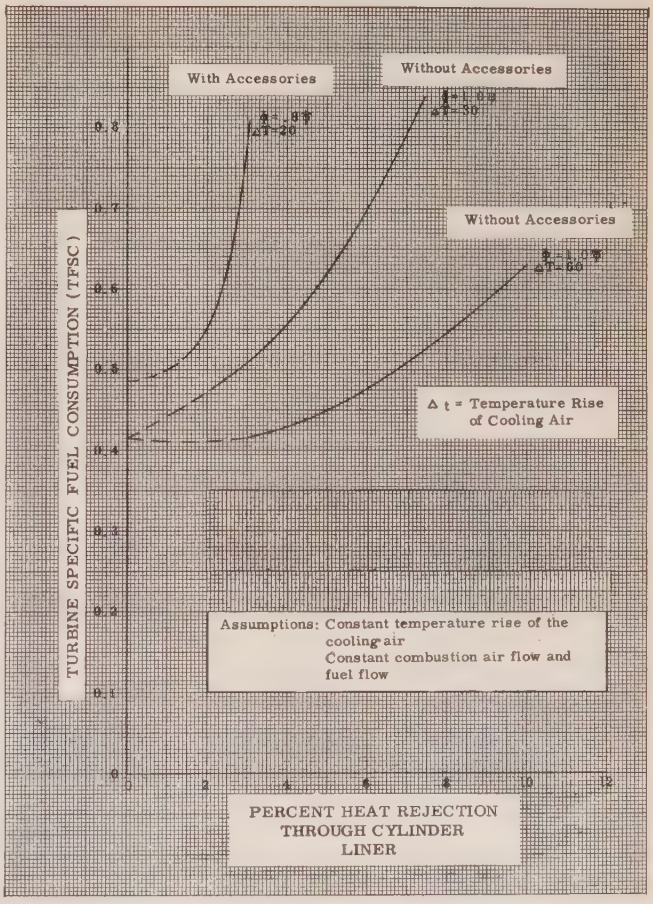


Fig. 88—Effect of cylinder-liner heat loss on specific fuel consumption as a function of fixed temperature rise of cooling air (through change in turbine temperature)

from equations 7 and 8. This is the factor least understood by those that have not studied this cycle.

$$\frac{W_c}{W_a} = \frac{h_f}{C_p\left(\frac{a}{f}\right)\lambda(1+\lambda_s)}\left(\frac{x}{\Delta T_{cc}}\right) \qquad (16)$$

On the curves of Fig. 86 is shown the plot of both equations 15 and 16.

The two airstreams, one the combustion air and the other the cooling air, combine in a plenum chamber upstream of the turbine at a pressure P_t which is common to both streams by design to minimize the mixing energy loss. The mixture temperature is a weighed mean of the heat content of both streams.

The cooling air, as it enters this plenum chamber, has an enthalpy content as follows:

$$H_{cco} = W_c C_p T_{cco} \qquad (17)$$

By substituting equations 8 and 9, the enthalpy content of the cooling airstream is

$$H_{cco} = \frac{W_f h_f x}{C_p \Delta T} C_p \left\{ T_o \left(\frac{\left[R_c^{\frac{k-1}{k}} - 1 \right]}{\eta_c} + 1 \right) + \Delta T_{cc} \right\} \qquad (18)$$

The enthalpy content of the combustion airstream is

$$H_{aco} = W_a C_p T_{aco} \qquad (19)$$

By substituting equations 7 and 10 and adding the heat energy in the exhaust, the enthalpy content of the combustion airstream is

$$H_{aco} = W_f\left(\frac{a}{f}\right)\lambda(1+\lambda_s)C_p T_o \left[\left(\frac{R_a^{\frac{k-1}{k}} - 1}{\eta_a} \right) + 1 \right] + W_f h_f (1 - \eta_d - x - Z) \qquad (20)$$

The mixture temperature in the plenum chamber is

$$T_t = \frac{H_{cco} + H_{aco}}{W_t C_p} \qquad (21)$$

Where W_t is the sum of the airflows of equations 7 and 8 plus the fuel flow W_f.

With the determintaion of the power gas conditions at the nozzle section the turbine output can then be computed

$$\text{thp} = \frac{W_t \eta_t}{550}\left(\frac{k}{k-1}\right) R T_t \left[1 - \frac{1}{R_t^{\frac{k-1}{k}}} \right] \qquad (22)$$

Fig. 88 shows the results of a turbine specific fuel consumption calculation both as a function of the cylinder heat rejection and the auxiliary power for a turbine of 600 shaft hp. It is apparent that too much auxiliary power reduces the net power to produce power gas with a very detrimental reaction to the turbine specific fuel consumption.

It is also important to notice the prominent part cylinder heat rejection has in increasing the SFC, and for this reason much more work should be done on both "hot-crown" pistons and heat-shielded cylinders. The "hot-crown" piston program, started rather early in the project, was later discontinued due to more urgent problems.

Nomenclature

$\left(\frac{a}{f}\right)$ = Stoichiometric air/fuel ratio = 14.2 lb air per lb fuel (approximately) and depends on the C — H ratio of the particular fuel being used, dimensionless

E = Heat to auxiliary equipment as $y\%$ of total fuel heat, Btu per sec = $W_s h \eta_a \eta_m y$

h_f = Heat content of fuel, 18,300 Btu per lb (LHV)

h_o = Enthalpy of ambient air, Btu per lb

H_e = Engine output in heat units, Btu per sec

H_f = Total heat input to system, Btu per sec

H_r = Heat rejection through the cylinder wall as $x\%$ of total fuel heat, Btu per sec = $W_f h x$

Heat to lube oil, piston cooling, radiation, and other unaccounted for losses as $Z\%$ of total fuel heat, dimensionless = $W_f h Z$

k = Ratio of specific heats, dimensionless

P_o = Ambient pressure, psia

P_t = Turbine inlet pressure, psia

ΔP_{ac} = Combustion air drop through cylinder, psi

ΔP_{cc} = Cooling air drop across cylinder, psi

ΔP_{aF} = Combustion air filter drop, psi

ΔP_{cF} = Cooling air filter drop, psi

R = Universal gas constant for air, 53.3 ft-lb per lb F

R_a = Combustion air compressor ratio, dimensionless

R_c = Cooling air compressor ratio, dimensionless

T_o = Ambient temperature, R

T_t = Turbine inlet temperature, R

ΔT_{cc} = Temperature rise of cooling air across cylinder fin, F

W_a = Combustion airflow, lb per sec

W_c = Cooling airflow, lb per sec

W_f = Fuel flow, lb per sec

W_s = Scavenge airflow, lb per sec

W_t = Trapped airflow, lb per sec

η_a = Combustion air compressor efficiency (static), dimensionless

η_c = Cooling air compressor efficiency (static), dimensionless

η_d = Diesel thermal efficiency, based on bhp at crankshaft flange, dimensionless

η_m = Mechanical efficiency for all external drives, dimensionless

η_t = Turbine thermal efficiency, dimensionless

λ = Combustion excess air ratio (1.5 approximately), dimensionless

λ_s = Excess scavenge air ratio, dimensionless

Φ = Engine parameter indicating the useful portion of the engine power available for compressing air, dimensionless

$$\Phi = \frac{k-1}{k}\frac{\eta_c \eta_m}{RT_o}\frac{3600 \times 550}{\text{SFC}}$$

CAVITATION CONTROL THROUGH DIESEL-ENGINE WATER TREATMENT

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W. C. Leith, Dominion Engineering Works, Ltd.

This paper was presented at the SAE Summer Meeting, Atlantic City, June 4, 1956.

IN recent years an increasing amount of attention has been focused on the effects of cavitation on diesel-engine cylinder liners. The design trend toward higher speeds and increased thermal and mechanical loading has in many cases made a serious problem of the erosive effects of this type of attack.

This phenomenon takes place only in specific areas of the liner. Fig. 1 shows a typical cylinder and indicates the regions where the strongest attack takes place. The severest pitting is always on the coolant side of the thrust surface of the liner, the side on which the piston bears during the power stroke. Less severe damage takes place on the water side of the free surface, and even more minor effects are noted on the axis parallel to the crankshaft. In almost all cases the areas along the 45-deg axis are completely free of any type of erosion, which tends to indicate that the cavitation attack is due primarily to the vibratory nature of the engine service. It appears that the liner is distorted into an oval shape by piston action and then rebounds into an oval shape along the longitudinal axis.

* Now with General Electric Co.

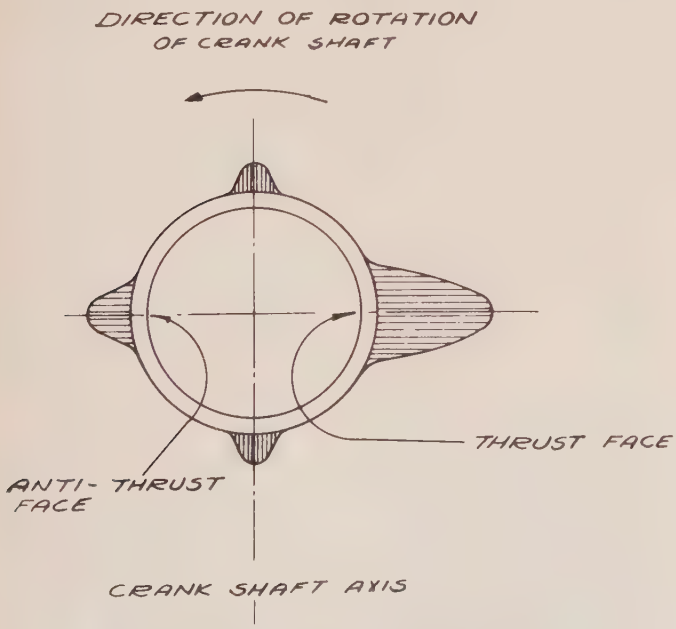


Fig. 1—Diagrammatic representation showing relative intensity of cavitation erosion of cylinder liners, related to direction of crankshaft rotation

The magnitude of the attack is independent of the location of water-inlet and -outlet surfaces. This is particularly noticeable in V-type engines where the water connections are usually made on opposite sides of the jackets in the two banks. While it is possible that water velocity could be a significant factor in inducing cavitation, it has been determined that this velocity is usually so low in diesel-engine design as not to be relevant. This theory is further verified by the nonuniform distribution of erosion despite the uniform distribution of water flow.

The generally accepted theory of cavitation implies that pitting occurs due to the formation of vapor bubbles within the liquid and their subsequent violent collapse. The bubbles are formed when the local pressure in the liquid falls below the vapor pressure of that medium, which may occur at a discontinuity in a flowing stream or during the negative stroke of a vibratory cycle. When pressure in the liquid returns to normal or when the vapor travels to a region of higher pressure, it will collapse very violently, inducing a local stress which has been estimated at 60,000 psi. Repeated impingement of the shock waves associated with bubble collapse proceeds to erode rapidly the adjacent metallic surfaces.

The formation and collapse of the vapor bubbles is evidently dependent on both the temperature and pressure conditions present at the liner surface. As

IT has been found that by means of magnetostrictive cavitation testing, it is possible to duplicate the conditions that lead to deterioration of the water side of diesel-engine cylinder liners.

In order to obtain accurate results it is necessary that conditions of temperature and pressure present in the water system be duplicated on the laboratory facility. It has been shown that increasing water pressure will definitely inhibit the erosive action of cavitation. Changes in operating temperatures will have less predictable results.

The use of a chromate-type corrosion inhibitor is effective in decreasing cavitation within concentration limits of 1000 to 2000 ppm.

Soluble-oil water treatments are also effective in concentrations between ½ and 2%. Boron-nitrate type of corrosion inhibitors are ineffective in combating cavitation.

the water temperature increases the vapor pressure rises rapidly, resulting in more and larger bubbles being created by the same driving mechanism. However, the pressure differential which affects the violence of collapse is decreased due to the increased vapor pressure. The end condition of the temperature cycle is reached at the boiling point, where cavitation is theoretically not possible since the vapor pressure balances the liquid pressure, and violent collapse will not occur. Pressure conditions likewise have contradictory effects in that the greater pressure serves to inhibit bubble formations. However, when the driving force is sufficiently extreme to create vapor conditions these bubbles will collapse with greater violence due to the higher driving pressure.

In view of the fact that it is a vibratory effect which instigates the cavitation attack on cylinder liners, it was decided to investigate various aspects of this phenomenom making use of the magnetostrictive-type cavitation tester. Previous work has been done by Rheingans [1] and reported by Speller and LaQue [2] using this type of equipment to evaluate cylinder-liner materials. In testing water treatments it was felt to be of paramount importance that operating temperatures and pressures be duplicated, and the test facilities were modified accordingly. While additional work was done in testing materials and coatings, this paper shall be exclusively concerned with the evaluation of various types of water treatment, holding the material as a constant.

Magnetostriction Apparatus

History—The principle of magnetostriction, which is the change in length of nickel when subjected to a magnetic field, was discovered by Joule, although it was Pierce who first built a magnetostrictive oscillator for the transmission of sound. Gains [3] published a paper in 1932 which reported an oscillatory apparatus for producing longitudinal vibrations in a nickel tube by magnetostriction. This apparatus combined the advantages of a small input of energy and easy handling of test materials to produce accelerated damage.

The first comprehensive report on magnetostrictive-cavitation testing was written by Kerr [4] in 1937 and lists 80 materials tested both in sea water and fresh water. A comprehensive treatise on accelerated cavitation was given by Nowotny [5] in 1942 in which the effects of temperature, pressure, surface tension, viscosity, gas content, and chemical solutions were investigated.

Fig. 2 is a diagrammatic representation of the magnetostriction apparatus in the mechanical research laboratory of Dominion Engineering Works, Ltd. A

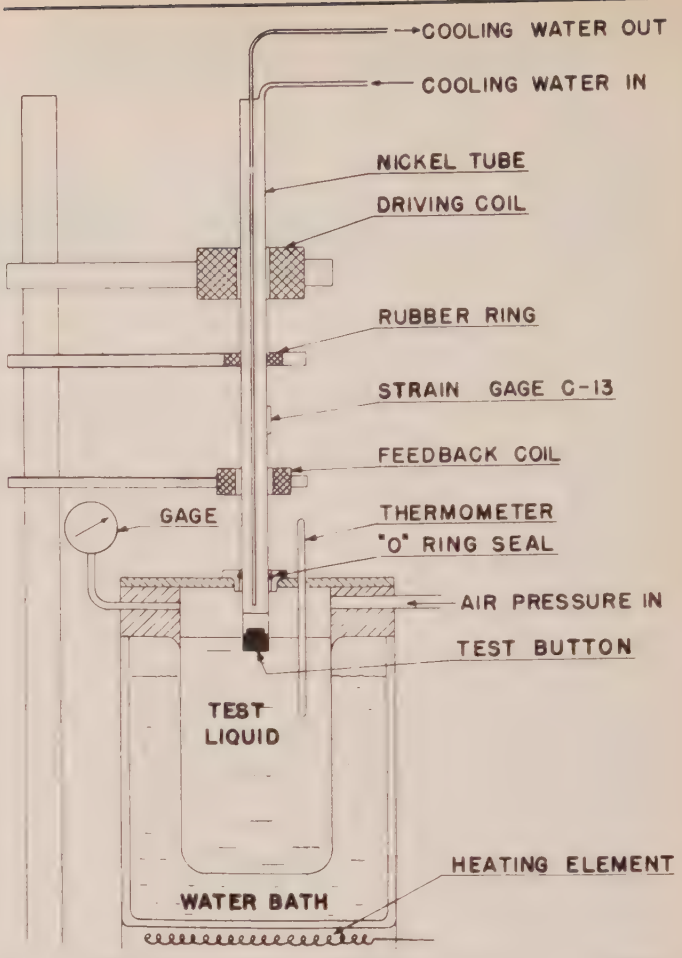


Fig. 2—Vibratory cavitation damage apparatus as modified for temperature and pressure control

[1] ASME Transactions, Vol. 72, 1950, pp. 705–719: "Accelerated Cavitation Research," by W. J. Rheingans.
[2] *Corrosion*, Vol. 6, July, 1950, pp. 209–215: "Water-Side Deterioration of Diesel-Engine Cylinder Liners," by F. N. Speller and F. L. LaQue.
[3] *Physics*, Vol. 3, May, 1932, pp. 209–229: "Magnetostriction Oscillator Producing Intense Audible Sound and Some Effects Obtained," by N. Gaines.
[4] ASME Transactions, Vol. 59, 1937, pp. 373–397: "Determination of Relative Resistance to Cavitation Erosion by Vibratory Method," by L. S. Kerr.
[5] "Destruction of Materials by Cavitation," by H. Nowotny. Pub. by VDI-Verlag GMBH, Berlin, 1942.

photograph of the test unit, including temperature and pressure modifications is shown in Fig. 3.

This equipment consists of a high-frequency oscillator which produces a forced vibration in the nickel tube as its natural frequency. Vibrations are induced at 6500 cps, and each end of the tube has an amplitude of 0.0035 in. The tube is 12 in. long, has ⅝-in. and 0.032-in. wall thickness, and is suspended by a tight rubber ring at the center. The test button is screwed in the lower end of the tube and immersed ¼ in. in the test liquid. Cooling is accomplished by an internal water spray. The amplitude of longitudinal vibration is determined by a resistance strain gage mounted on the nickel tube.

Relative performance of the apparatus is checked at frequent intervals using standard specimens, which are made of half hard rolled brass in accordance with specification ASTM B 16–52.

All tests are run for a period of 120 min.

Temperature and Pressure Control

Water-bath temperature is controlled to limits of ± 2 F within the test range of 50 to 190 F. A thermometer is used to check the test liquid while water-bath temperature is controlled by means of a heating coil and cold water.

A stainless-steel pressure vessel with neoprene "0" ring seals is used to contain pressed air above the test liquid. Control is maintained at ± 0.2 psi. The "0" ring seal at the nickel tube was soaked in vasoline and then wiped dry. Calibration tests were made at atmospheric pressure with the container in place, and excellent correlations were obtained as compared with completely free specimens.

Test Procedure

Specimen Preparation—Test buttons were machined from actual alloy cast-iron cylinder liners. Particular care was maintained to assure that the surface to be cavitated was part of the cooling surface. In this portion of the program it was felt to be desirable to eliminate material variables, and all test results described were made on buttons from two liners of similar hardness and microstructure. Chemical composition of the test material is given in Table 1.

The specimens are ⅝ in. in diameter and weigh approximately 13 g. A surface finish of 50 microin. is obtained by the use of metallographic polishing paper. The buttons are prepared for testing by first boiling in distilled water for 10 min. This treatment is used to saturate the specimens, since cast materials may absorb water into their porous structure.

After testing, all test buttons are photographed so

Table 1—Typical Properties of Material Tested

Chemical Composition, %			
Total Carbon	3.08	Chromium	0.33
Combined Carbon	0.52	Nickel	1.27
Manganese	0.78	Molybdenum	0.41
Silicon	2.13		
Hardness			
210 Brinell			
Microstructure			
Type A Graphite Fine Pearlite Matrix			

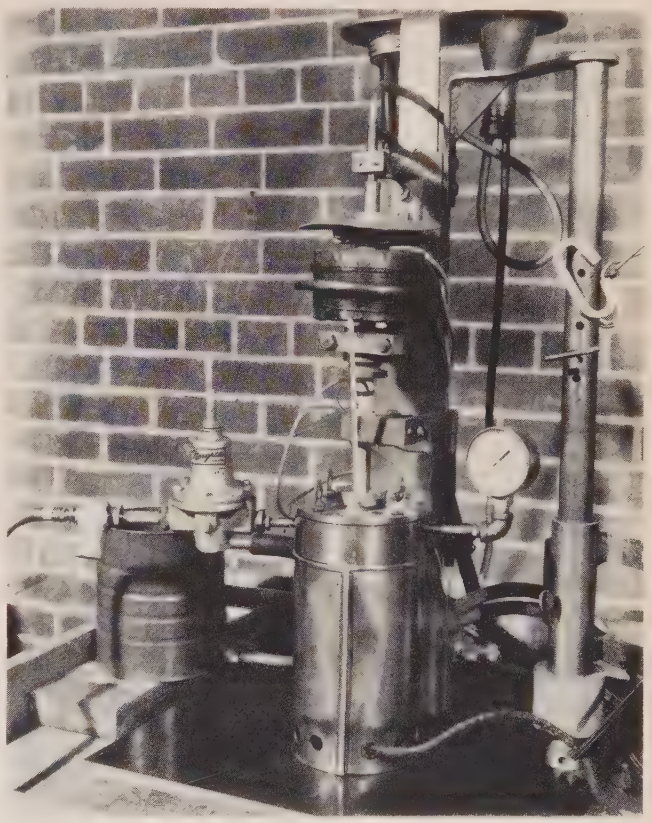


Fig. 3—Photograph of test unit used for magnetostrictive cavitation testing. Details are shown in Fig. 2

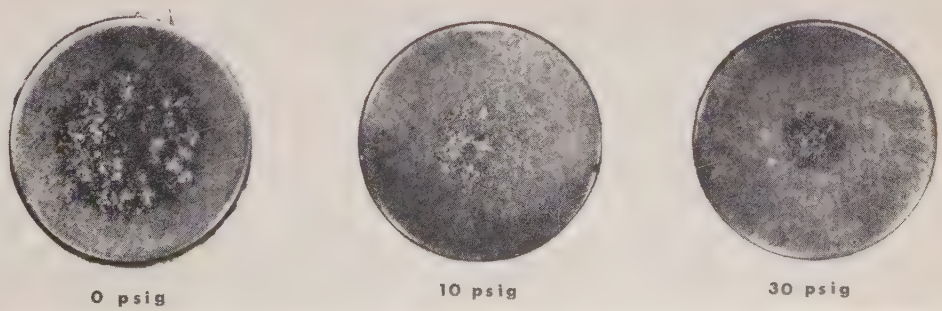


Fig. 4—Effect of pressure variations on cavitation. A marked decrease in area attacked is shown on specimens tested at various pressures in chromate solution at 170 F

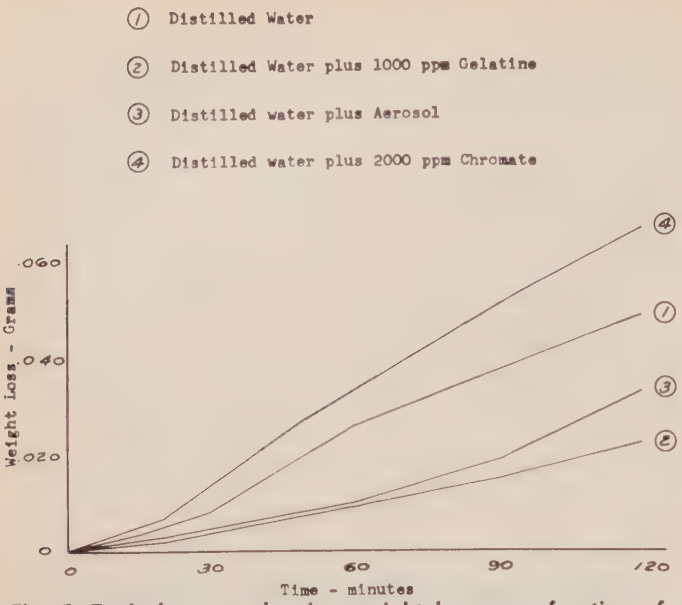


Fig. 5—Typical curves showing weight loss as a function of time. Data obtained at 80 F and atmospheric pressure

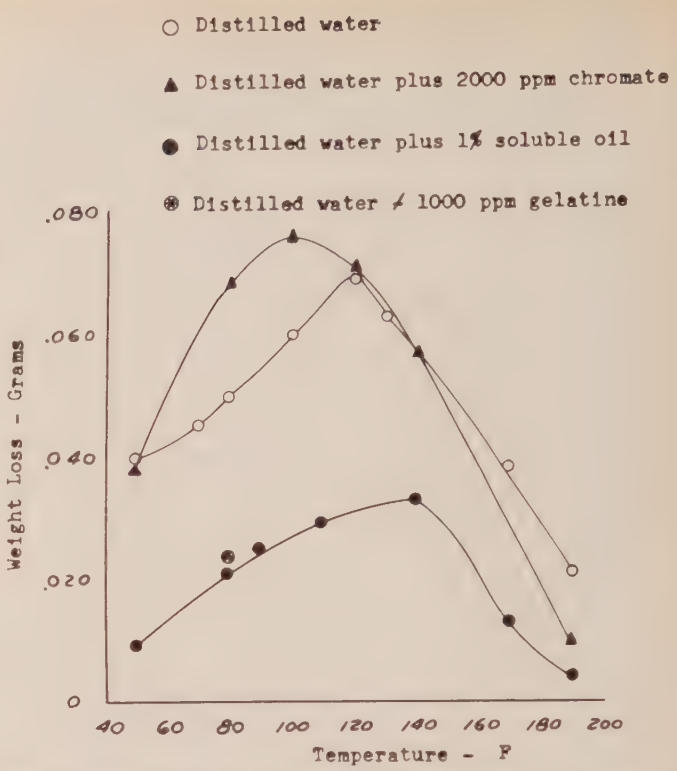


Fig. 6—Weight loss as a function of temperature at atmospheric pressure, in various liquids

that size and pattern of the damaged area can be correlated to the weight loss. Fig. 4 shows the photographic record made of a series of specimens.

Water-Treatment Compounds

In order to assure reproducibility of test liquids, distilled water has been selected as a standard medium. The additives evaluated were:

1. Chromate ($K_2Cr_2O_7$) type water treatment of proven potency in subduing cavitation damage.
2. Boron-nitrate type of inhibitor which was known to provide protection against corrosion while being ineffective in preventing cavitation.
3. Soluble-oil corrosion inhibitor.
4. Gelatine. This material was reported by Horton [6] as strongly inhibiting cavitation in test liquids. While of questionable value from the stability point of view, it was felt desirable to test this additive as representative of the amino acid wetting agents.
5. Aerosol. A chemical surface-tension depressant which was selected to increase wetability and adherence of the liquid to the liner.

Temperature and Pressure Variations

In order to obtain information on the potency of the various solutions over the engine-operating range, tests were conducted at the following temperatures: 50 F, 80 F (room temperature), 115 F, 140 F, 170 F. (operating temperature at liner surface), and 190 F.

To determine the effect of pressure, tests were run under atmospheric conditions, 10 psig, 20 psig (operating condition on the cylinder liner), and 30 psig. Complete pressure runs were made at room temperature and also 170 F.

Test Results

Fig. 5 shows typical curves of weight loss of the specimens in various test solutions. Weight loss in this case is plotted as a function of time and shows what is commonly regarded as an incubation period of a fatigue nature during the early portion of testing. Of particular interest is the curve shown for the chemical wetting-agent type additive. Originally, this additive shows a high potency in restraining cavitation. Subsequently, however, a breakdown of the treatment appears, and cavitation proceeds at a rapid rate. This loss in effectiveness is probably due to the severe abuse of vibratory testing. Based on these results it was decided to discontinue further testing on this type of surface active agent.

Complete temperature runs were made using distilled water, chromate solution, and soluble-oil water treatment. Fig. 6 shows curves obtained under atmospheric pressure conditions, while Fig. 7 indicates the results obtained under a pressure of 20-psi gage. The original data is given in Table 2. Of particular interest is the fact that under ambient pressure the chromate type of inhibitor could not be considered as being markedly more effective than the water alone, while the soluble oil was extremely potent.

When the system is pressurized to a condition simi-

[6] *Journal of the Acoustical Society of America*, Vol. 25, May, 1953, pp. 480–484: "Effect of Intermolecular Bond Strength on Onset of Cavitation," by J. P. Horton.

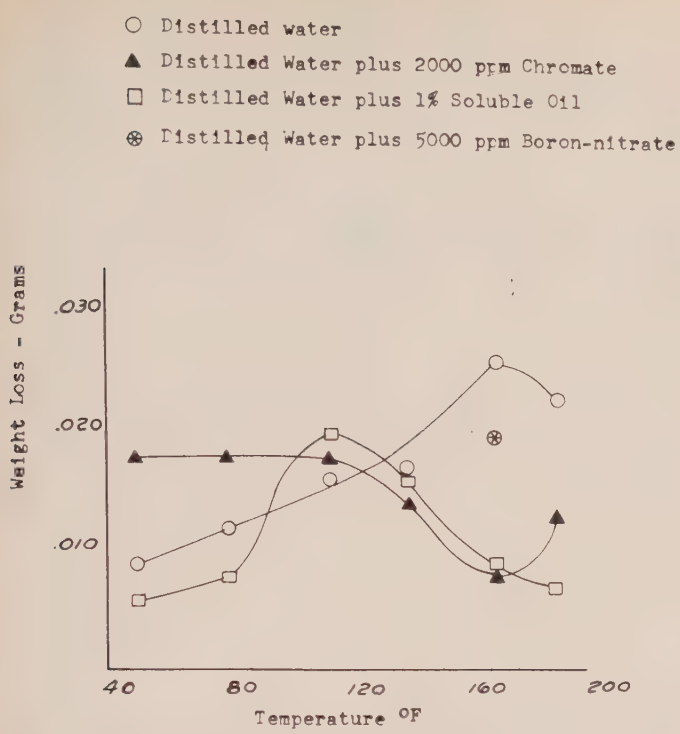


Fig. 7—Weight loss as a function of temperature at 20-psi gage, in various liquids

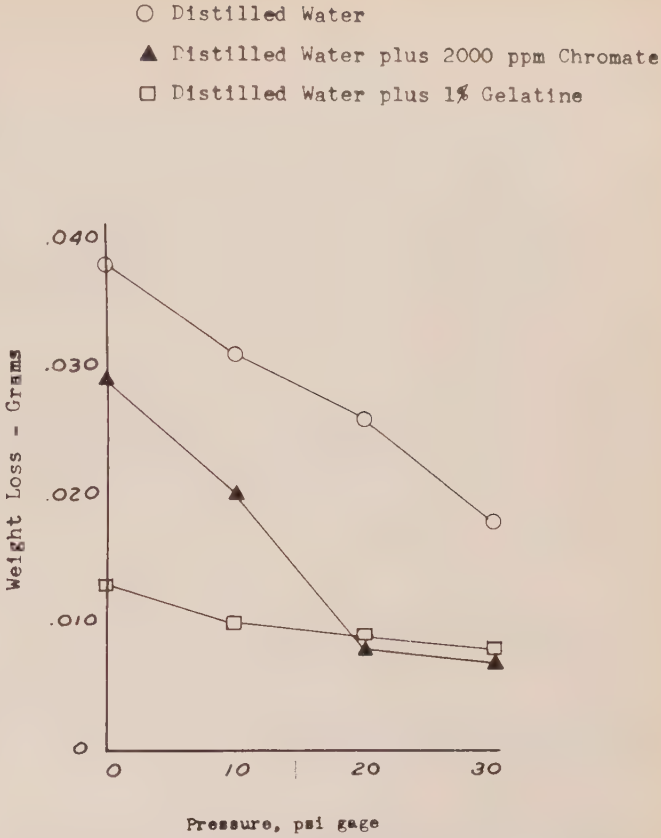


Fig. 8—Effect of pressure on cavitation damage. Data obtained at 170 F

lar to that encountered in diesel-engine operation the reason for the effectiveness of the chromate treatments becomes quite clear. The fact that maximum weight-loss differential between chromate and water takes place at the point of normal operating conditions would tend to indicate that damage occurs almost exclusively under full-load operation rather than during warmup or idling periods. The remarkable performance of the soluble oil under atmospheric pressure was not duplicated under the pressurized conditions, although the overall effectiveness appears to be equivalent to that of the chromate-type treatment.

The boron-nitrate-type corrosion inhibitor was tested only at operating conditions, and the result is plotted in Fig. 7. As was anticipated, this additive cannot be considered as effective in inhibiting cavitation. Extensive testing was not done with the gelatine treatment due to its general nature. The single point shown on Fig. 4 and 5 indicates that this additive could provide potent cavitation inhibition at room temperature and atmospheric pressure. Tests conducted at elevated temperatures and pressures showed virtually no benefit from this type of water treatment.

Fig. 8 and Table 3 show the results obtained while maintaining a constant temperature and varying the pressure. In all cases there was a decrease in weight loss with each incremental increase in pressure. While it would have been of considerable interest to determine the pressure levels which could completely stop cavitation from taking place, this was not possible with the equipment at hand.

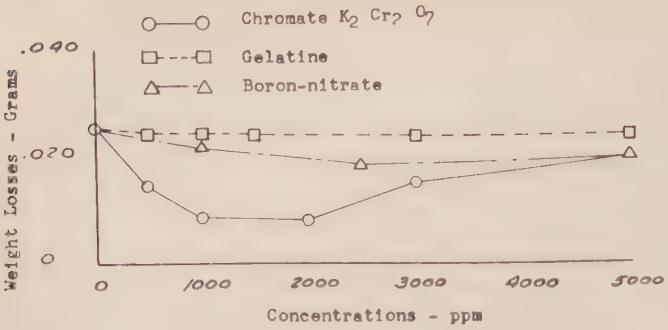
In order to determine the concentration limits

Table 2—Cavitation Rates of Standard Specimens under Varied Conditions of Temperatures, Pressure, and Water Treatment

(Cavitation Rate—g/120 min)

Temperature, F	Pressure, atmospheric			Pressure, 20-psi			
	Distilled Water	Chromate[a]	Soluble Oil	Distilled Water	Chromate[a]	Soluble Oil[b]	Boron-Nitrate[c]
50	0.040	0.39	0.008	0.009	0.018	0.006	
70	0.045						
80	0.050	0.069	0.021	0.012	0.018	0.008	
100	0.060	0.076					
115			0.029	0.016	0.018	0.020	
120	0.069	0.071					
130	0.063						
140	0.057	0.033		0.017	0.014	0.016	
170	0.038	0.029	0.013	0.026	0.008	0.009	0.020
190	0.022	0.010	0.004	0.023	0.013	0.007	

[a] Concentration 2000 ppm of $K_2Cr_2O_7$
[b] 1% concentration
[c] Concentration 5000 ppm



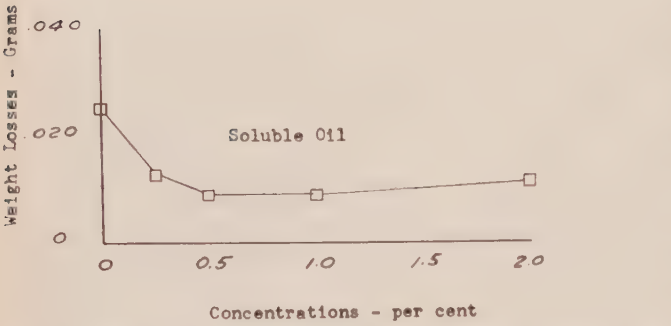


Fig. 9—Effect of concentration of various water treatments at 170 F and 20 psig

Table 3—Effect of Pressure on Cavitation Rate

At 170 F Cavitation Rate—g/120 min

Pressure psi gage	Distilled Water	Chromate 2000 ppm	Soluble Oil
0	0.038	0.029	0.013
10	0.031	0.030	0.010
20	0.026	0.008	0.009
30	0.018	0.007	0.008

Table 4—Effect of Water-Treatment Concentration on Cavitation Rate

At 170 F and 20 psig Cavitation Rate—g/120 min

Treatment Concentration, ppm	Chromate	Boron-Nitrate	Soluble Oil	Gelatine
0	0.026	0.026	0.026	0.026
500	0.015			0.025
1000	0.009	0.023		0.025
2000	0.008			0.024
2500		0.019	0.013	
3000	0.015			0.024
5000 (0.5%)	0.020	0.020	0.009	0.024
10,000 (1%)		0.016	0.009	
20,000 (2%)			0.011	

within which each additive was most effective, tests were run under the operating conditions of 170 F and 20 psig for the full 120-min cycle. Concentrations were varied from a minimum value to one considerably in excess of recommended usage. Results are shown in Fig. 9, and the data is given in Table 4. Based on these tests the soluble-oil type of inhibitor showed the desirable characteristic of a high degree of potency over a wide concentration range. The chromate solution was particularly effective in the range of 1000 to 2000 ppm, but it seems indicated that overdosage could completely destroy the cavitation-inhibiting properties of this additive. It is of interest to note that the recommended concentration of $K_2Cr_2O_7$ for corrosion-resisting purposes is on the order of 500 ppm, though previous field experience had indicated that dosages of 2000 ppm were necessary to provide adequate cavitation protection. The validity of these empirical conclusions appears to be borne out by the test results.

Conclusions

1. Based on the test work described above it is concluded that accurate duplication of diesel-engine cylinder-liner cavitation can be achieved by means of magnetostrictive testing. Where previous knowledge existed on the potency of various types of water treatment, the vibratory test duplicated these results with satisfying accuracy. Nevertheless, it must be maintained that this type of laboratory testing should be considered only as a method of screening, which serves to indicate fruitful areas for comprehensive field testing.

2. In order to obtain accurate results from the magnetostrictive test equipment it is essential that engine-operating pressures and temperatures be duplicated. Results obtained under other conditions can be completely misleading.

3. The soluble-oil type of water treatment may be considered as providing the same degree of overall cavitation protection as the chromate-type water treatment. In addition, the soluble oil has the added advantage of being relatively insensitive to concentration variations.

4. The chromate-type water treatment appears susceptible to overdosage. Care should be taken to maintain concentration limits between 1000 and 2000 ppm.

5. The boron-nitrate type of corrosion inhibitor does not provide protection against cavitation up to 5000-ppm concentration.

6. Gelatine additions appear effective in inhibiting cavitation under ambient conditions, but seem to be without merit at elevated temperatures and pressures. The surface-tension depressant used in this test broke down under cavitation abuse. It is thought possible that other types of surface active agents may show more desirable characteristics and should be investigated. Water treatments such as these do not of themselves provide corrosion protection and would be used in conjunction with some other type of additive.

7. It appears that pressurizing the water system of diesel engines will result in diminished cavitation effects regardless of water treatment used.

8. Changes in operating temperature will have less predictable effects upon cavitation. The net effect of such a change can be predicted by laboratory testing of the approved water treatments.

DISCUSSION

. . . of this paper, as well as of the following Joyner paper, is to be found on pp. 346–348.

REDUCTION OF CAVITATION PITTING OF DIESEL-ENGINE CYLINDER LINERS

J. A. Joyner, Studebaker-Packard Corp.

This paper was presented at the SAE Summer Meeting, Atlantic City, June 4, 1956.

THIS paper deals with cavitation pitting where vacuum bubbles exist. Experiments are described which attempt to correct this condition where plating of liners varied in composition and thickness. It was concluded that a nickel-chrome plating has very good resistance to cavitation-pitting attack.

Investigation of cylinder-wall vibration was made to find out about the cause of cavitation. Minimum liner wall vibration results with use of a 2-piece heavy wall liner and a piston with 20% less clearance. Endurance tests prove that on these pitting has been eliminated.

DAMAGE to the water side of cylinder liners, which until recently has been classified as chemical or electrolytic attack, is now recognized in many instances to be associated with mechanical action defined as cavitation pitting.

Many excellent papers have been written on cavitation pitting, but the majority of these cover laboratory work on materials or deal with the problem as encountered in pumps or hydraulic turbines. These papers are helpful to the diesel-engine designer, but they do not apply directly to his specific problem.

With the increase in demand for more horsepower per pound of engine, it is necessary to work all materials a little nearer to their endurance limit. Unfortunately, in the case of diesel-engine cylinder liners, the increase in deflection which results from the higher stresses can cause damaging cavitation pitting although the material is not being worked near its endurance limit.

There are at least two types of cavitation pitting. One type is associated with the formation and collapse of vacuum bubbles created at or near the surface of the material when the absolute pressure in the area drops below the vapor pressure of the liquid. The second type is caused by pressure waves being transmitted through the liquid. There are no vacuum bubbles associated with the second type; yet, the damaged surface has the same general appearance. This paper will deal with the type of cavitation pitting where vacuum bubbles do exist.

Brief Description of Engine

A problem was encountered, in the fall of 1951, in the Series 142 diesel engine designed and built by Studebaker-Packard. These engines are light-weight, high-speed, turbocharged engines operating on a 4-stroke cycle with a precombustion chamber. Each cylinder has a displacement of 142 cu in. and develops 50 hp at a rated speed of 2000 rpm. The engine was designed with a unit-head and cylinder-liner as-

sembly.[1] The original cylinder consisted of a nitralloy barrel (AMS 6470) quenched and tempered to 28–32 Rockwell C and welded to a 4130 cast-steel head. The welded joint was approximately 0.3 in. below the face of the head. The liner was nitrided on the inside diameter to a case depth of 0.007 to 0.010 in. to increase wear resistance and nickel-plated with 0.003 in. nickel on the outside diameter to prevent rusting. A photograph of the parts and cylinder assembly is shown in Fig. 1. The liner was supported in the crankcase just below the head and sealed at the bottom with three "O" rings as shown in Fig. 2. A forged AMS 4142 heat-treated aluminum piston was used. A photograph of the piston is shown in Fig. 3.

Description of Damage

Damage was noted at the end of the first endurance test on the Model 1D-1700 12-cyl engine. Tap water was used as a coolant. Upon inspection, 10 of the 12 liners were pitted. The general location of the attack was as shown in Fig. 4. The attack was 90 deg from the crankshaft centerline and was more pronounced on the thrust side although it did exist on the antithrust side on a few liners. There was an indication of a pear-shaped pattern, but most of the actual pits were just below to about 1 in. below the support shoulder and ranged in size from a pin point to about the size of a dime.

The appearance of the pitting was as shown in Fig. 5. The area attacked was sharply defined with abrupt boundaries between affected and unaffected metal and had a honey-combed appearance. There were no corrosion products, and the pits had rounded bottoms.

Early Experiments

The plating was changed from nickel to nickel cadmium, and a phenolic varnish was applied to the finished liner. The purpose of this change was to determine if galvanic effects were the prominent factor. Another endurance test was conducted, and those liners were also pitted.

Fig. 1—Parts and assembly of cylinder

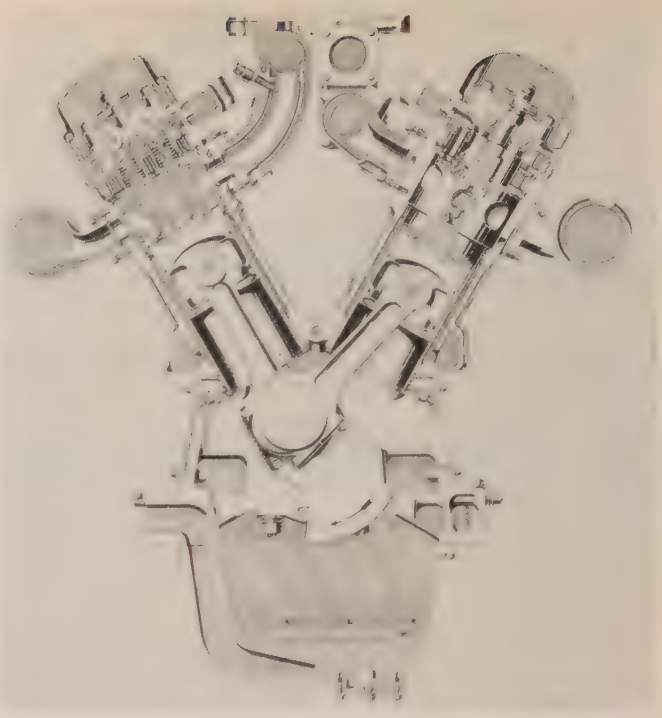


Fig. 2—Position of liner in engine

A nickel-chrome plate was tried next with chrome thickness varying from 0.001 to 0.005 in. The 0.003-in. nickel-, 0.0005-in. chrome-plated liners ran for a considerably longer time before pitting appeared.

The next experiments consisted of thicker liners and also thicker chrome plate. The liner wall thickness was increased from 0.156 to 0.187 in. This liner became known as the "thick wall" design and the original production as the "thin wall" design. The chrome plating was increased from 0.005 to 0.010 in. Some of these liners were operated for over 900 hr without any signs of etching or pitting; however, additional tests on other engines were not so successful.

Since most of the pits were occurring in the welded area, it was thought that a brazed construction might have higher resistance to attack.

Further tests included brazed constructions, tapered wall surfaces giving additional thickness in the critical location, and liners nitrided on both the inside diameter and outside diameter. None of these eliminated the trouble.

Soluble oil was tried as an inhibitor, and tests indicated some reduction of attack as compared with tap water.

Although the use of chrome-plated, thick wall liners and soluble oil prolonged the life of the cylinder liners considerably, it was obvious that the trouble was not completely eliminated.

[1] SAE Transactions, Vol. 62, 1954, pp. 363–375: "New Packard Light-Weight Diesel Engine," by M. Ware, R. E. Taylor, and J. Witzky.

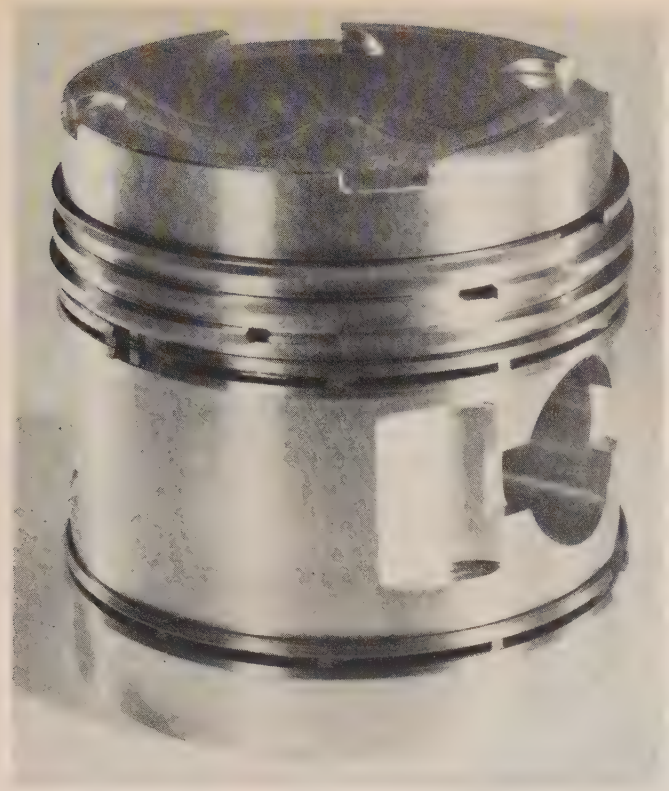


Fig. 3—Forged AMS 4142 heat-treated aluminum piston used in engine

Fig. 4—General location of pitting on cylinder liner

Review of two technical papers [2,3] indicated that the possibility of cavitation attack should be explored. A conference was arranged with F. L. LaQue of the International Nickel Co., Inc. and representatives of the Bureau of Ships to discuss the problem. It was concluded that the plated material used should be tested in a cavitation test machine, and that other inhibitors such as sodium chromate or sodium nitrate-sodium nitrite should be investigated.

Investigation of Plating Materials

Arrangements were made with W. J. Rheingans of Allis-Chalmers Mfg. Co. for testing the plated material specimens. Eight specimens of specially machined AMS 6470 material were prepared. These samples were tested in a vibratory type magnetostriction cavitation test machine.[2] The test was conducted at a frequency of 6500 cps through a travel of 0.0034 in. with the samples submerged ⅛ in. below the surface of the water which was 2½ in. deep. The temperature was maintained at 76 F ± 1 F. Air pressure was 760 mm of Hg. Each sample was weighed before and after each 30-min test. The original weight of each sample was approximately 13 g. A comparison of the attack on conventional and self-regulating

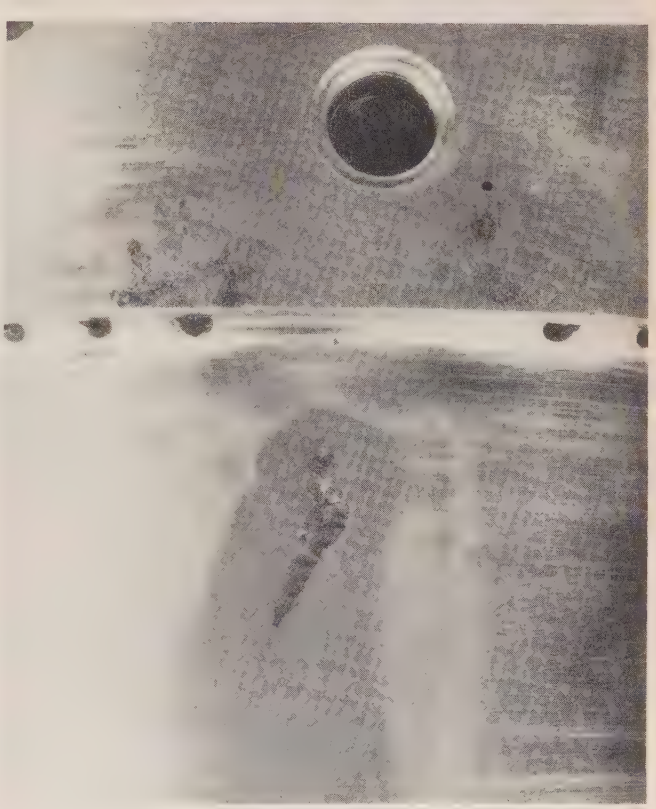


Fig. 5—Appearance of pitting

2 "Accelerated Cavitation Research," by W. J. Rheingans. Presented before ASME, Nov. 29, 1949.

3 *Corrosion*, Vol. 6, July, 1950, pp. 209–215: "Water-Side Deterioration of Diesel-Engine Cylinder Liners," by F. N. Speller and F. L. LaQue.

chrome is shown in Fig. 6. A photograph of a Fax film replica is shown in Fig. 7. A summary of the cavitation test on plated materials is shown in Table 1. The attack obtained on the test buttons was found to be similar to that experienced on the cylinder liners.

The weight loss of all chrome-plated samples was considered low. The chrome-plated materials were better than any previously tested material except rolled stellite which had a weight loss of only 0.6 mg after a similar 120-min test. The only explanation for the variation between samples one and two is a possible difference in surface finish. A smooth surface increases resistance to attack. The weight gain experienced with sample three was due to water absorption. Visual examination revealed that there was no attack during the first 30 min; so the sample apparently gained 0.3 mg of water which would make the actual 120-min weight loss 1.4 instead of 1.1 mg as shown in Table 1. The only explanation for the higher resistance of the self-regulating chrome plate as compared to conventional chrome plate is the grain size. Self-regulated chrome has approximately 3000 grains per sq in. as compared to 720 grains per sq in. for conventional chrome. The hardness of both ranges between 1000 and 1200 Vickers.

Inhibitor Test

Alcoa recommended against the use of sodium chromate in an aluminum-iron system based on past experience with aluminum cylinder heads and cast-iron block. An inhibitor consisting of 28.35-g sodium nitrite, 13.1-g sodium nitrate, and 18.9-g sodium dihydrogen phosphate was tested in a Model 1D-1700 engine. There appeared to be some improvement, but

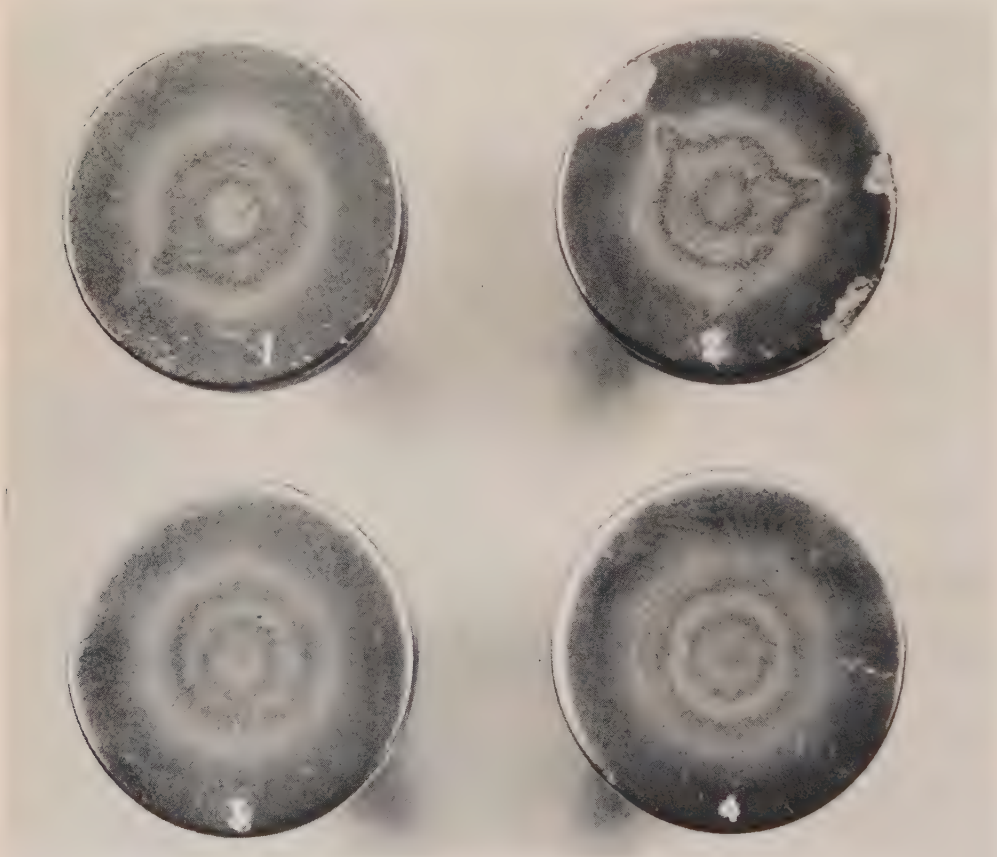


Fig. 6—Comparison of attack on conventional and self-regulating chrome-cavitation test buttons

Chromium Plated Cavitation Test Buttons Tested at Allis-Chalmers

#1 and 2 Conventional Chromium Plating Solution
#3 and 4 Self Regulating Chromium Plating Solution

Sample No.	Plating Thickness Ni	Plating Thickness Cr	Cr Plate Vickers Hardness	Loss of Weight in Milligrams after 2 hr. test.
#1	.0053"	.011"	1020–1160	3.5
#2	.0044"	.010"	1139–1200	6.6
#3	.0075"	.010"	1106–1223	1.1
#4	.0029"	.0105"	1096–1210	2.0

Fig. 7—Fax film replica of No. 2 cavitation button after 120-min test run

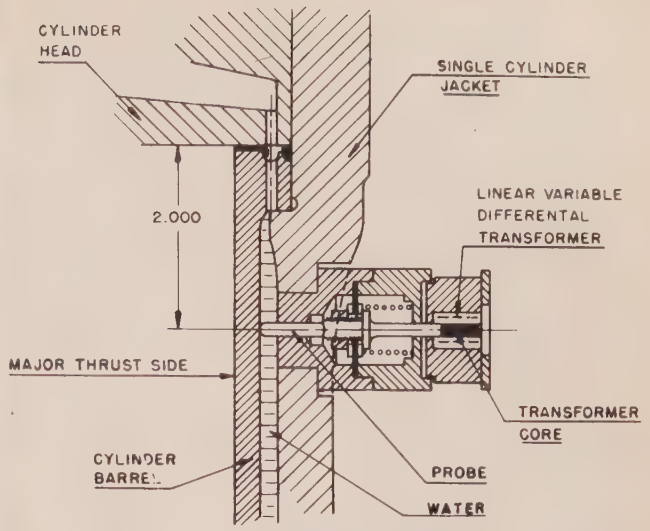


Fig. 8—Mounting of linear variable differential transformer to measure movement of cylinder liner

3 of the 12 liners were slightly pitted indicating that this was not a cure.

Investigation of Cylinder-Wall Vibration

It became clear that the problem could not be completely solved by the use of plating and inhibitors. Cavitation tests indicated that the materials being used were among the best. The next step was to find out more about the cause of cavitation. A test was set up to measure cylinder-liner vibration at the point of attack. The apparatus used in this test consisted of:

1. Single-cylinder test engine.
2. Cylinder-liner wall vibration recording equipment.

   a. Jackson audio frequency oscillator Model 652.
   b. Dumont oscillograph Model 304H.
   c. Linear variable differential transformer.
   d. Polaroid Land Camera.

A linear variable differential transformer was calibrated and mounted on a thrust-side wall of the single-cyl water jacket with a spring-loaded probe in contact with the cylinder liner as shown in Fig. 8. A schematic diagram of the electrical connections used is shown in Fig. 9.

The resultant piston side thrust was calculated for a complete cycle. These forces are shown in Fig. 10. This gave a clue as to what part of the cycle would produce liner vibration.

Liner vibration amplitude was measured on the following liners:

Table 1—Summary of Cavitation Test on Plating Materials

Sample Number	Plate: Flame-Plated Tungsten Carbide	Plate: Nickel	Plate: Hard Chrome	Plate: Self-Regulating Chrome	Weight Loss, mg: 30 min	60 min	90 min	120 min
1		0.0045	0.010		0	0.1	0.8	3.5
2		0.0045	0.0095		0.7	1.3	2.6	6.6
3		0.008		0.010	0.3 Gain	0.2 Gain	0	1.1
4		0.003		0.0095	0.04	0.09	1.5	2.0
5	0.0073				11.4	24.3	39.8	57.5
6	0.0067				10.1	20.2	31.2	44.1
7	0.0083		0.010		0	0.9	2.8	5.2
8	0.0067		0.010		0.7	1.7	3.8	6.3

1. Original short taper (0.156 in.) wall with 5.562 radius starting 0.830 in. below shoulder.
2. Thick cylinder (0.187 in.) wall.
3. Thin cylinder (0.156 in.) wall.
4. Heavy tapered (0.233 to 0.286) wall.
5. Two-piece heavy (0.250) wall cylinder employing:

   a. Metal "O" ring head seal.
   b. Embossed metallic head seal.
   c. Rolled type head seal.

6. A thick (0.187 in.) wall cylinder wrapped with stellite wire in the area of attack.

A number of pistons were tested with each of the above cylinders. These included:

1. Standard production pistons of different weights.
2. Standard production pistons except ⅛ in. offset piston-pin hole.
3. Standard production pistons with one wear band on the top land.
4. Standard production pistons with two wear bands on the top land.
5. Graphited piston with 20% less cold skirt clearance.
6. Piston with 20% less cold skirt clearance, Key-

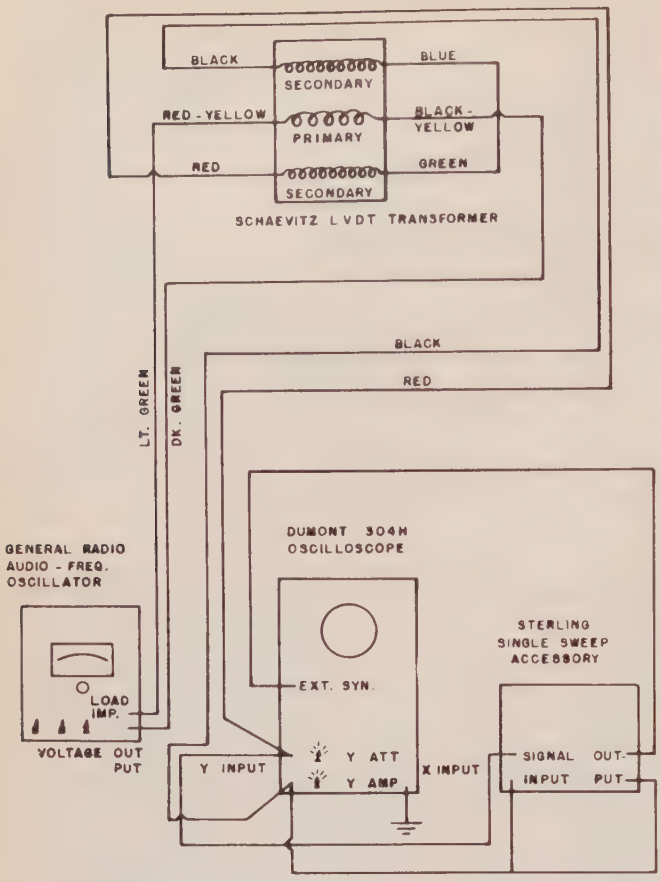


Fig. 9—Schematic diagram of electrical connections of linear variable differential transformer for cylinder vibration test setup

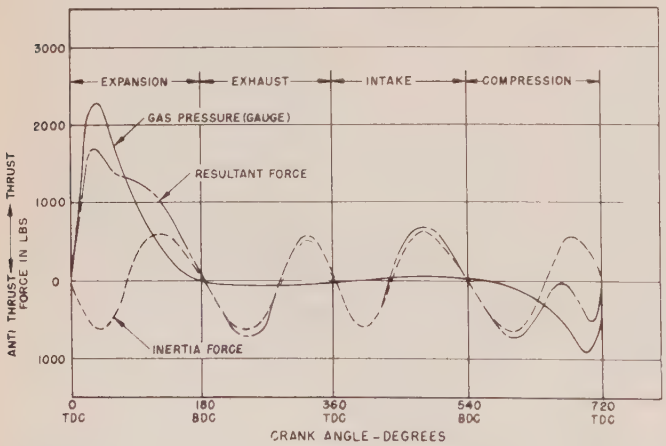


Fig. 10—Piston side-thrust forces at rated bhp and 2000 rpm

stone top compression ring, and two wear bands on the top land.

These data were recorded for several engine loads and speeds and with different water pressures and cylinder-retention loads. Fig. 11 shows the amplitude measurements for a 0.156-in. wall cylinder with a standard production piston for water pressures of 9 and 30 psi. Fig. 12 shows the amplitude for a 2-piece

Single Cylinder Engine
Liner Wall Vibration

Cyl. No. Thin Wall	–	344
RPM	–	2000
BHP	–	42.7
LVDT No.	–	100
Calibration	–	.3"/.001"
Oscillator Frequency	–	20,000 cps
Oscillograph	–	304-H
Piston	–	Std. "D" WT
Amplitude (RT to LT)	–	
Fire	–	.00434"
Exh.	–	.00633"
Int.	–	.00316"
No "O" ring in groove in head.		
Std. Water Pressure	–	9 PSI

3-1-54

Single Cylinder Engine
Liner Wall Vibration

Cyl. No.	–	344
RPM	–	2000
BHP	–	42.7
LVDT No.	–	100
Calibration	–	.3"/.001"
Oscillator Frequency	–	20,000 cps
Oscillograph	–	304-H
Piston	–	Std. "D" WT
Amplitude (RT to LT)	–	
Fire	–	.00417"
Exh.	–	.00433"
Int.	–	.00233"
No "O" ring in groove in head.		
High Water Pressure	–	30 PSI

Fig. 11—Amplitude measurement for 0.156-in. wall cylinder with standard production piston for water pressures of 9 and 30 psi

heavy wall cylinder with a piston having 20% less skirt clearance. These are typical of the data obtained. These data were then plotted as shown in Fig. 13 for further analysis.

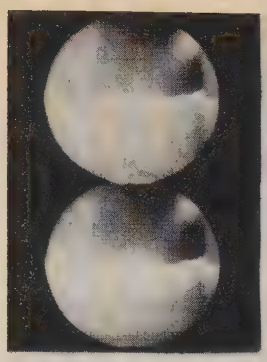
The following conclusions were reached:

1. Liner wall vibrations were found to vary from slightly over 0.006 in. for a thin wall liner with a production piston to less than 0.001 in. for a 2-piece heavy wall cylinder with a piston having 20% less clearance. A comparison of wall vibration for a thin wall liner with production piston, heavy tapered wall liner with production piston, and 2-piece heavy wall liner with 20% less clearance piston is shown in Fig. 14.
2. An excellent relationship existed between the calculated piston side thrust and the measured amplitude as shown in Fig. 15.
3. The offset piston pin did not represent a significant improvement.

Observation of Cavitation Bubbles

The range of cavitation bubbles was investigated next to determine how it varied with different amplitudes of vibration. The following equipment was used:

1. Single-cylinder test engine.
2. Strobatac.
3. Strobolux.



Single Cylinder Engine
Liner Wall Vibration

Cyl. No. (2PC "O" Ring)	–	XA84351-7
RPM	–	1400
BHP	–	11.8
LVDT No.	–	100
Calibration	–	.30/.001"
Oscillator Frequency	–	20,000 cps
Oscillograph	–	304-H
Piston (2 Scuff Bds., 20% less skirtclear)		XA85255
Amplitude (Rt. to Lt.)	–	
Top 9 PSI Water Press. (Fire	–	.001"
Top 9 PSI Water Press. (Exh.	–	.00073"
Top 9 PSI Water Press. (Int.	–	.0004"
Bottom 15 PSI (Fire	–	.001"
Bottom 15 PSI (Exh.	–	.0007"
Bottom 15 PSI (Int.	–	.00046"

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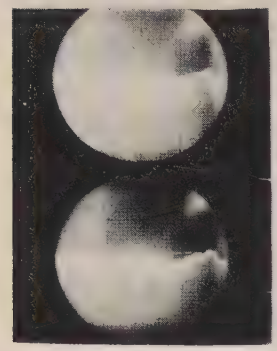

Single Cylinder Engine
Liner Wall Vibration

Cyl. No. (2 Pc. "O" Ring)	–	XA84351-7
RPM	–	1800
BHP	–	28.65
LVDT No.	–	100
Calibration	–	.30/.001"
Oscillator Frequency	–	20,000 cps
Oscillograph	–	304-H
Piston (2 Scuff Bds., 20% less skirt clear)		XA85255
Amplitude (Rt. to Lt.)	–	
Top 9 PSI (Fire	–	.00107"
Top 9 PSI (Exh.	–	.0006"
Top 9 PSI (Int.	–	.00043"
Bottom 15 PSI (Fire	–	.001"
Bottom 15 PSI (Exh.	–	.00053"
Bottom 15 PSI (Int.	–	.0004"

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Fig. 12—Amplitude for 2-piece heavy wall cylinder with piston having 20% less skirt clearance

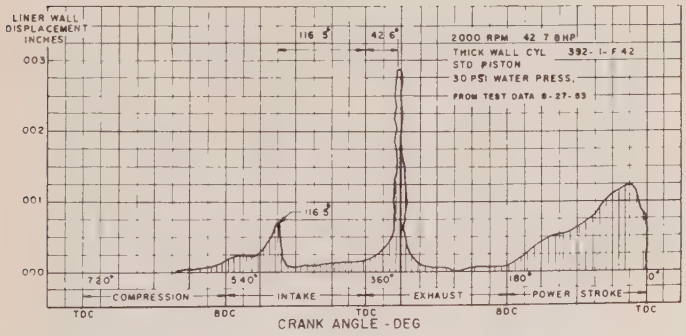

Fig. 13—Cylinder-liner wall displacement

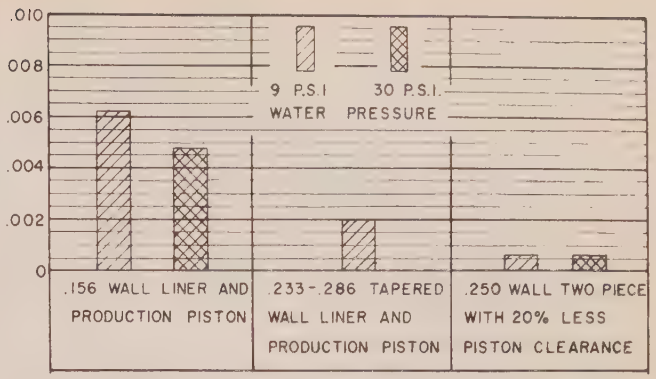

Fig. 14—Cylinder-liner wall vibration at 2000-rpm rated load

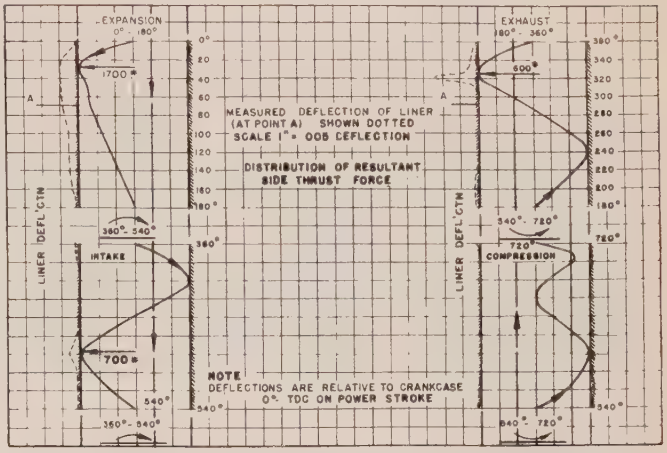

Fig. 15—Standard liner deflection for single cylinder at 2000 rpm and 42.7 hp

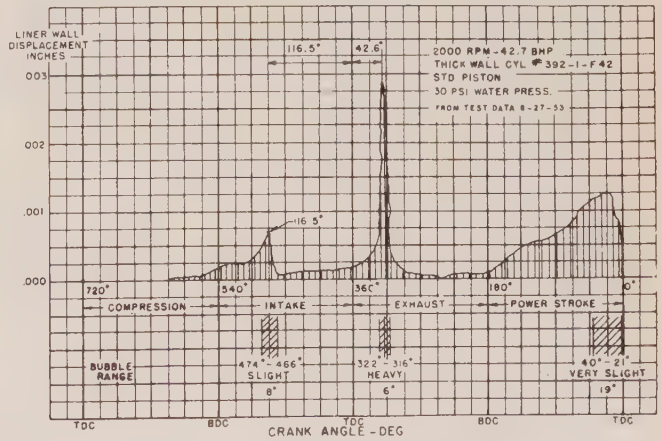

Fig. 16—Composite diagram showing cylinder-liner wall displacement and cavitation bubble range

4. Brush contactor triggering mechanism.
5. Cathometer telescope.

The differential transformer was replaced by a 1-in. plexiglas window. The flywheel was drilled and tapped for 20 contactors, and the brush had a range of 20 deg so the entire cycle could be investigated. The contactors were used to trigger a Strobatac which in turn triggered a Strobolux. By the use of this lighting system and a telescope the exact bubble range could be determined for any part of the cycle for any piston-liner combination. The brush was adjusted until the bubbles appeared and moved slowly until they disappeared. The range, as well as the intensity of the bubbles, was recorded. Bubbles were observed at three points in the cycle, and there was excellent correlation with the amplitude measurements as shown in Fig. 16.

Endurance Test of Piston with 20% Less Skirt Clearance

Single-cyl tests had indicated such a reduction in liner vibration amplitude when a 2-piece heavy wall

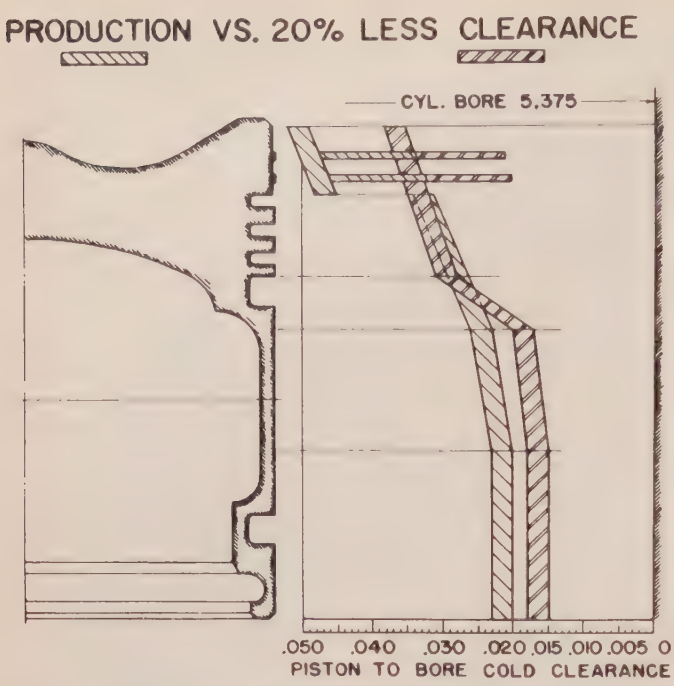

Fig. 17—Comparison of skirt clearance for standard pistons and pistons with 20% less skirt clearance

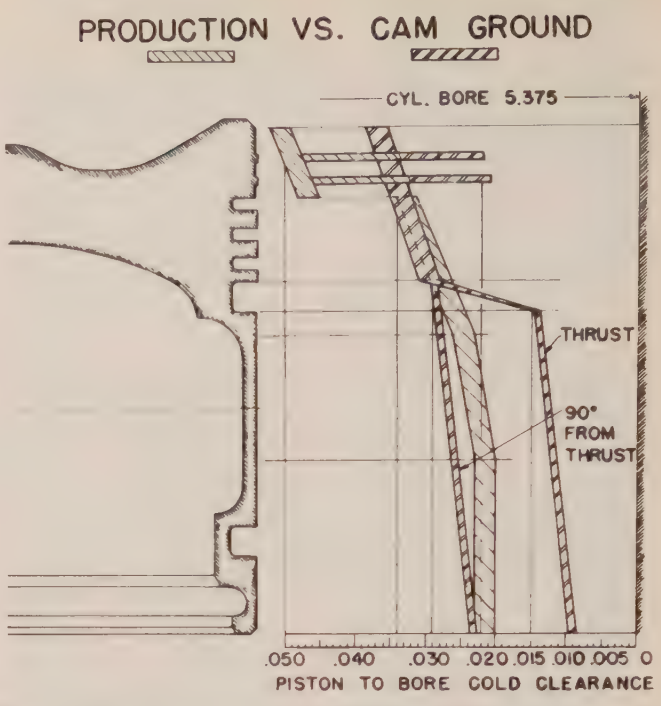

Fig. 18—Cold clearance of production piston versus cam-ground piston

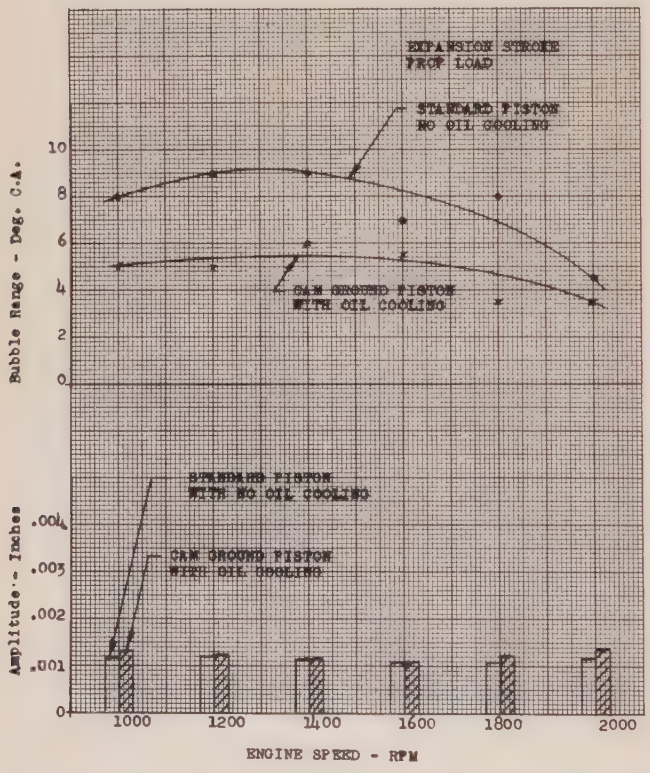

Fig. 19—Cavitation corrosion bubble range and cylinder-liner vibration for the expansion stroke based on propeller load curve for two pistons

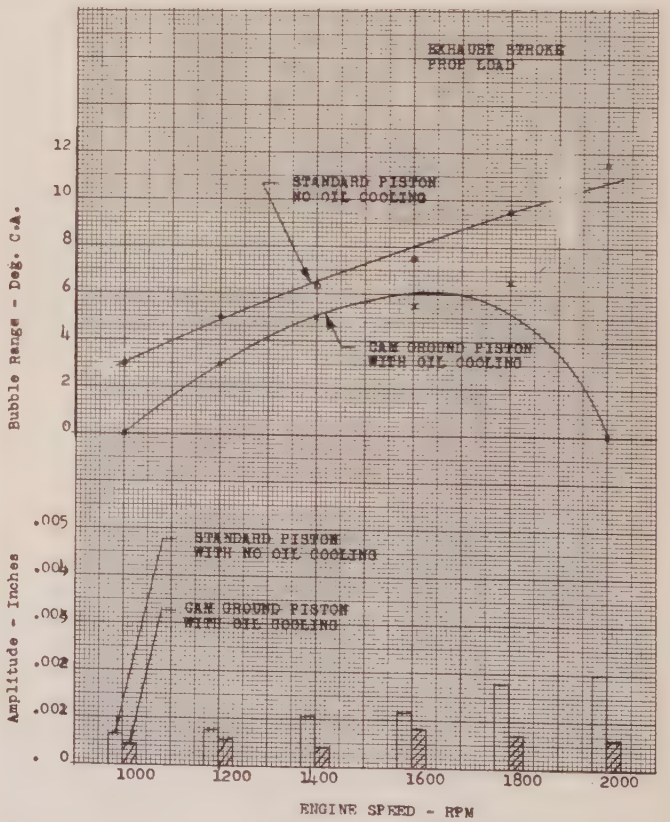

Fig. 20—Cavitation corrosion bubble range and cylinder-liner vibration for exhaust stroke based on propeller load curve for two pistons

liner and a piston with 20% less skirt clearance were used that an endurance test was desirable. A Model 1D-2270 16-cyl engine was equipped with sixteen 2-piece heavy wall nickel-chrome-plated cylinder liners. Eight standard pistons were installed in the even-numbered positions, and eight pistons with 20% less skirt clearance were installed in the odd-numbered positions. The difference in skirt diametrical clearance for the two pistons in shown in Fig. 17.

The engine was tested with 1¼ % by volume soluble oil inhibited coolant.

Inspection revealed that all liners with the standard production pistons were pitted or etched, but none of those with the 20% less skirt clearance pistons were etched or pitted; however, three of the eight lower clearance pistons were scuffed.

This test indicated that good control of piston clearance was the most effective method of eliminating cylinder-liner cavitation pitting.

Cam-Ground Piston Development

The production pistons were forged 4142 heat-treated aluminum. Other materials which would have a lower temperature coefficient of expansion, thereby reducing the clearance control problem, were considered, but it was decided to keep the same material and to develop a cam-ground piston which provides the

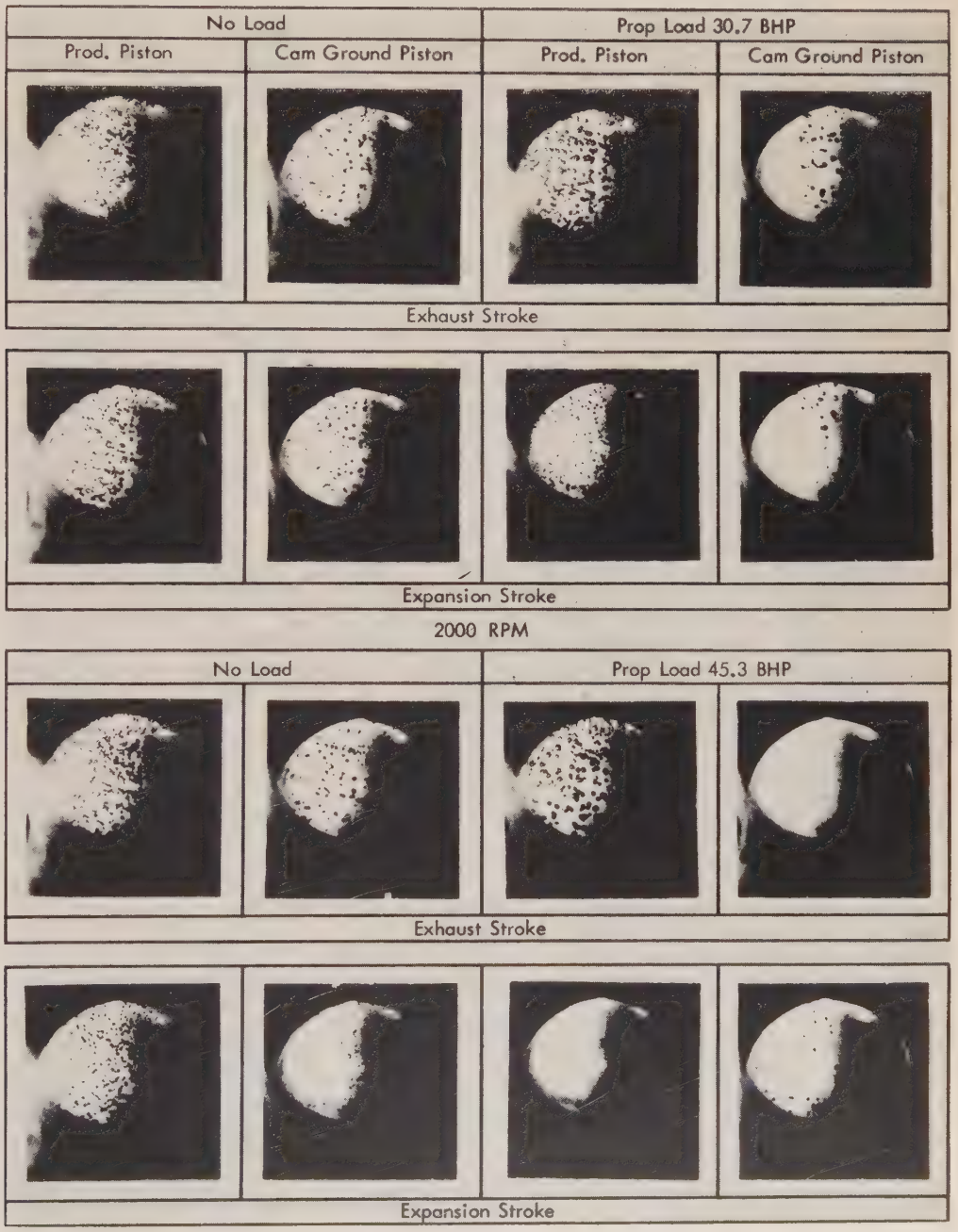

Fig. 21—Cylinder-wall cavitation bubbles for no load and propeller load at 1800 and 2000 rpm for both cam-ground and production pistons on expansion and exhaust stroke

reduced clearance in the plane of the piston side thrust. Oil cooling was also added for better clearance control, and the manufacturing tolerances were reduced to improve control. Fig. 18 shows the cold clearance of the production piston versus the cam-ground piston. Liner vibration and bubble range tests were made with the cam-ground piston and production piston operating in the 2-piece heavy wall nickel-chrome-plated cylinder liner. Fig. 19 shows a comparison of vibration amplitude and bubble range for the expansion stroke based on propeller load curve for the two pistons, and Fig. 20 shows the same comparison for the exhaust stroke.

Photographs were taken through the plexiglas window with a Speed Graphic Camera. The same lighting arrangement as previously used for visual observation was used except the Strobolux was fired only once. The photographs showing the bubbles for no load and propeller load at 1800 and 2000 rpm for both the cam-ground and production pistons on the expansion and exhaust strokes are shown in Fig. 21.

High-Speed Movies

High-speed movies of the cavitation bubbles were taken through the plexiglas window of the single-cyl test engine. A 2-piece (0.250 in.) wall cylinder liner and a production piston were used. The film speed varied due to inertia, but the 60-cps marker was used so the actual film speed for any portion of the cycle could be determined. The final speed was approximately 3150 frames per sec which means slightly less than 4 crank angle deg per frame with the engine operating at 2000 rpm.

Two 50-ft rolls of film were taken. One roll was taken at 1200 rpm, 25% load, and 9-psi water pressure. The second roll was taken at 2000 rpm, full load, and 23-psi water pressure. Cavitation bubbles existed on the expansion, exhaust, and intake strokes as previously noted. The maximum number of consecutive frames showing bubbles varied from one to three. This indicated that the bubble duration was between 0.001 and 0.003 sec. As in the still photographs, the exhaust stroke showed the heavier formation. Generally, there were two frames with heavy bubble formation followed by one blank frame and then one frame with a very, very light formation indicating very high damping.

Recent Endurance Tests

Three multicylinder endurance tests have been conducted on the Model 1D-1700 12-cyl engine during the past year. Some cam-ground pistons and 2-piece heavy wall nickel-chrome-plated cylinder liners have been used in each of these tests, and none has ever pitted. One of the tests was conducted with a split cooling system. A 40% Prestone solution was used in one bank and 1% chromate solution, consisting of 0.25% dichromate and 0.75% sodium chromate, was used in the other. Each bank had two cam-ground pistons and four standard pistons. All positions were equipped with 2-piece heavy wall nickel-chrome-plated cylinder liners. All four of the liners with standard pistons in the Prestone-inhibited bank were pitted, whereas only one of the liners with standard pistons was pitted in the chromate-inhibited bank, and that was a very small pit; however, the chromate inhibitor does not appear to be compatible with the aluminum even when a low pH (8.0 to 8.5) is maintained and cannot be recommended for use with aluminum in the presence of steel.

Conclusions

1. Control of piston clearance is the most effective method of reducing pitting on diesel-engine cylinder liners.
2. Nickel-chrome plate has very good resistance to cavitation pitting attack.
3. High-frequency vibration amplitudes in excess of 3 mils are extremely difficult to protect.
4. Amplitude of vibration can be reduced by increasing cylinder-wall thickness.
5. Inhibitors are beneficial.

Acknowledgments

The Bureau of Ships and the USN Engineering Experiment Station have provided helpful suggestions throughout the program.

Appreciation is also expressed to F. L. LaQue and other members of the International Nickel Co., Inc., who provided advice on cavitation and general corrosion phenomena.

DISCUSSION

Needs Further Examination of Cavitation Causes and Controls

—J. I. Bregman and R. G. Bielenberg
National Aluminate Corp.

WHILE the results given by Margulis et al. are very interesting, there exists considerable doubt that the magnetostriction apparatus used in this work can be considered to give accurate duplication of diesel-engine cylinder-liner cavitation. One example of the flaws inherent in this type of testing is the short 2-hr test period. Since this paper deals with the effect of water treatments on cavitation, the short time interval could not be expected to show the effect of general corrosion control on cavitation. The correlation between general corrosion and cavitation has not been established, but if there is any correlation, the short test period would not permit its evaluation.

Potassium chromate is apparently used as the standard chro-

mate type of inhibitor in the tests described in this paper. In actual diesel operation, the chromate-type inhibitors used are in almost all cases pH-buffered chromates in the sodium state. As earlier recommendations by diesel-locomotive builders have indicated that a minimum pH of 8.5 is desirable for control of cavitation, it would be of interest to know the pH of test solutions used and the effect of pH on cavitation in this apparatus.

The statement is made that chromatic treatments are "of proved potency in subduing cavitation damage" while boron-nitrate inhibitors are "ineffective in preventing cavitation." Actually, in the field it has been found that protection against cavitation with either chromate or boron-nitrate treatments varies appreciably. Also, we have found many cases where in a given engine with the same water, the same water treatment (either chromate or boron-nitrate), the same operating conditions, and presumably the same materials of construction, one or more liners would be badly cavitated while other liners in the same engine would be in excellent condition. This would seem to underline the facts that the problem is essentially mechanical rather than chemical in nature and that the ultimate solution lies in the design rather than in the water-treatment field. As a further example of this, in a type of engine in which cavitation is commonly experienced on liners and jackets, when the separate liner and jacket is replaced by an integrally cast liner and jacket, no cavitation has been detected.

Many railroads are unable to use chromates for reason of hazards to the employees. Soluble oils have not been found acceptable in most cases because of attack on hose connections and seals. For these reasons more complete testing of possible alternatives would have been desirable. For example, conclusions are drawn on the behavior of a boron-nitrate treatment of 5000 ppm only (Fig. 7). It should also be pointed out that this dosage is lower than that which is recommended by inhibitor manufacturers.

A point of considerable interest is the fact that the weight loss for chromates appears to rise rapidly with temperatures above 170 F at 20 psi while at atmospheric pressure the weight loss for chromates decreases markedly at temperatures over 170 F. This possibly should be explored in more detail since there is an increasing interest in higher cooling system temperatures such as in vapor-phase cooling.

Properties of Material, Local Pressure Affect Formation of Cavitation Bubbles

—R. S. Carleton

Bureau of Ships, Department of the Navy

DURING the past several months the Bureau of Ships has become increasingly aware of the effect of damage due to cavitation as a maintenance problem in some internal-combustion engines. For this reason, I would like to discuss a few factors contributing to this type of damage.

According to Phillip Eisenberg in the David Taylor Model Basin Reports 712 of July, 1950, and 842 of June, 1953, on the mechanism and prevention of cavitation, cavitation will usually begin when local pressure reaches the vapor pressure corresponding to the temperature of the liquids. In the same report he further states that cavitation may begin above or below the vapor pressure depending on the condition of the liquid (air content and air or vapor nuclei which may exist in a stable condition or which may be stabilized on solid particles). According to the above-mentioned reports and reports of other investigations, nuclei in the form of free gas, vapor bubbles, or bubbles trapped on solid particles seem to be necessary for the formation of cavitation bubbles. Local pressure may be affected by several factors. The pressure may decrease in areas of increased velocity due to restrictions or turbulence. We may also have a pressure drop due to withdrawal of the cylinder liner as a result of liner vibration. Mr. Eisenberg in his report 712 quotes from an investigation by J. Ackeret. For a receding piston in a tightly fitted cylinder the pressure is $p = P - \rho \times \frac{\partial u}{\partial t}$ I believe that this is analogous to the rapid withdrawal of the cylinder liner during high-frequency vibrations. The reference pressure P would be the system pressure minus any pressure drop due to increased velocity or turbulence. This equation shows that the pressure drop p is proportional to the product of the acceleration and the amplitude of vibration of the liner. It is pointed out that damage increases with increased amplitude and frequency of liner vibration. However, I think that more stress should be put on decreasing cavitation by equalizing the water flow throughout the sleeve area and thus eliminating any low-pressure areas due to increased velocity. Mr. Eisenberg also points out that deep crevices with sharply pointed bottom contours are positions where vapor phase may easily occur. These type shapes which are commonly found in cylinder-liner lower sleeve areas should be avoided if possible.

As pointed out, hardness is perhaps the major property affecting cavitation erosion resistance of a material, excluding materials such as neoprene where resilience seems to be the major factor. However, other investigators have found that cavitation erosion resistance of a material can be affected by other properties of the material. J. M. Mousson discussed the importance of the work-hardening properties of the materials, and W. Mantel and J. M. Mousson in separate works found that grain size can affect erosion resistance. Other investigators have found some correlation between tensile strength of a material or inclusions found in a material and erosion resistance.

Speller and LaQue [a] say that damage associated with cavitation probably is a combination of mechanical and electrochemical effects with either mechanical or corrosion effects predominating, depending on the severity of the cavitation effects, as reflected in the time in which the action occurs, the nature of the material, and the corrosivity of the environment. They further state that the effectiveness of chromate inhibitors suggests that corrosion frequently, if not always, predominates over mechanical forces in the cavitation erosion of diesel cylinders.

Solution to Problem Requires Attention to Engine Construction

—F. L. LaQue

International Nickel Co., Inc.

MY principal reaction is one of satisfaction that the problem of deterioration of the water side of cylinder liners has been recognized as one that requires attention from many angles and is not simply one of doing something about corrosion in conventional ways.

When Dr. Speller and I looked into this matter about 10 years ago the problem had been laid in the laps of chemists who were told to do something to make the cooling water less corrosive. We realized that much more attention would be required as indicated by the following quotation from the paper [a] we presented before the National Association of Corrosion Engineers following our studies: "The ultimate solution to this peculiar corrosion problem will require the cooperation of the mechanical engineers with attention to mechanical features of the design and construction of engines."

In this connection, it is significant to take special note of conclusion No. 1 of the paper by Mr. Joyner: "Control of piston clearance is the most effective method of reducing pitting on diesel-engine cylinder liners." This supports our original suggestion that high-frequency vibrations were at the root of the difficulty.

Depending on the severity of the cavitation erosion forces to be overcome, other steps may be more or less effective. These steps may include one or more of such actions as water treatment, inhibitors, protective coatings, and the use of materials having superior resistance to cavitation effects. But I repeat my conviction that the design engineers have the greatest opportunity to eliminate the trouble by reduction of destructive vibrations of the cylinder walls and jackets.

While the high-frequency vibratory cavitation erosion testing

[a] See footnote 3 of Joyner paper.

apparatus used or referred to by several investigators produces effects of the same kind as those responsible for the damage on diesel-engine cylinder liners, the intensity of the mechanical forces is so magnified in the laboratory apparatus that it is almost impossible to study properly the corrosion aspects of the cavitation erosion with which we are concerned. A test which is completed in an hour or two does not provide enough time for corrosion to have much influence on the results. Such a test cannot make a proper distinction among materials having important differences in corrosion resistance with about the same mechanical properties, nor will it enable the effects of inhibitors or the relative merits of different inhibitors to be observed. We are currently engaged in the development of a more rational cavitation erosion test which will run for a longer time and achieve a better balance between the mechanical and corrosive forces in cavitation erosion.

Alkaline Chromates Serve Best as Cooling System Corrosion Inhibitors

—R. W. Seniff

Baltimore and Ohio Railroad Co.

IT is known that there are several designs of diesel engines in use on railroads which are subject to severe cavitation of cylinder liners on the water side. These engines are of the type having a high relative velocity between coolant and liner. An example is a 12-cyl engine having a 280-gal capacity cooling system with one water pump of 650-gpm capacity with 30-psi discharge pressure.

However, work in connection with alkaline chromate inhibitors has clearly demonstrated the cavitation can be eliminated by the maintenance of a 2000–3000-ppm chromate concentration at a 8.5–9.5 pH value. Certain liners known to perforate in a few months unless the cooling water is adequately inhibited are now nine years old and show no signs of cavitation.

Therefore, in so far as engines in extensive use on railroads are concerned, undue emphasis seems to have been placed on the mechanical factors, and the need for corrosion inhibitors has been overlooked even in the absence of cavitation.

Localized corrosion such as pitting or grooving is caused by a combination of localized conditions. Changing one factor of the combination generally eliminates the corrosion. Which factor is changed depends on economics; in some cases it is cheaper to change mechanical design, in others to treat the water.

Engine designers, in some instances, have made some bad choices of metal combinations and designs in engine water systems. Cooling water in contact with stress members, welds, or aluminum of certain compositions creates obstinate corrosion problems. However, alkaline chromate water treatment maintained at adequate concentrations will prevent harmful corrosion within the cooling water system. Aluminum alloys are available which perform successfully in cooling systems where alkaline chromates are used.

We have tried some 60 different chemicals and combinations of chemicals as cooling system corrosion inhibitors without finding anything equal to alkaline chromates for the purpose.

ORAL DISCUSSION

Reported by A. B. Neild, Jr.

USN Engineering Experiment Station

H. Chessin, Van der Horst Corp.: There are apparently two schools of thought on the cause of coolant jacket deterioration in diesel engines. There is overwhelming evidence that this deterioration is a combination of electrochemical corrosion and cavitation, but the primary cause is cavitation.

It is basically a design problem, but either an inhibitor or a protective coating may improve the situation. By stopping either the electrochemical corrosion or the cavitation attack, the problem will be solved.

J. C. Miller, Cummins Engine Co.: Deterioration is a mechanical process whereby the protective coating is removed and the surface left free for electrochemical action. The problem cannot be solved by closer piston clearances due to scuffing under certain conditions of customer operation.

Conditions for minimum corrosion are:

1. Higher coolant temperature.
2. Control of cylinder-liner wear.
3. Control of coolant pressure.
4. Control of inhibitor to minimize electrochemical action. We have correlated this in the field by use of high-coolant pH, chromate inhibitor, and a coolant filter.

There is not much hope for a cure through design in lightweight engines. A protective coating is effective to a point, but it cannot be applied on the full surface of the block. However, the method of control is at hand.

V. Vanderbilt, Perfect Circle Corp.: Has any test work been done to eliminate electrochemical effects by the use of oil?

Mr. Joyner: In our light-weight engines (under 5 lb per hp), piston clearance had a great effect. Originally the cylinder liners in the engines had a vibration amplitude exceeding 6 mils. Water treatment is not effective for above 1-mil amplitude.

Thickness of the coolant passage has an effect, but the detrimental effects may not be due to velocity.

A. K. Blackwood, International Harvester Co.: I agree with Mr. Miller that an increase in water temperature would decrease cavitation. There would also be an increase in liner deflection as the load increased to full load.

A decrease in piston clearance to decrease piston slap is limited by operating conditions which may not allow close tolerances. Some conditions of operation with close piston clearances may cause scuffing.

A. R. Schrader, USN Engineering Experiment Station: Most cases of severe coolant side deterioration are in high-speed lightweight engines where not much design information is available. Cavitation bubbles form and collapse so rapidly on the surface of a magnetostriction test button that it would take an abnormally high coolant velocity to have any effect. I believe that the damage is concentrated at the center of the cavitation test button by edge effects. Perhaps the surface of the button itself may vibrate.

Even if an oil coolant is used to remove electrochemical effects, there would be insufficient heat transfer in most engines in which cavitation erosion is experienced.

Winterization Of Construction Equipment

Report of CIMTC Subcommittee XV—Winterization

This symposium was presented at SAE National Tractor Meeting, Milwaukee, Sept. 11, 1956.

Introduction to Winterization Program

by

M. G. Mardoian
International Harvester Co.

and

F. M. Baumgardner
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TODAY, we of the CIMTC Subcommittee XV present what we believe to be the best current engineering thinking on the techniques of winterizing construction and industrial machinery for operation at extremely low temperatures. In connection with winterization studies we have been asked, what is the purpose of the Winterization Program, and why do we need special engineering on construction and industrial machinery for low-temperature operation. The answers to these queries are that there is a growing military and civilian need for equipment which can operate effectively in the extremely frigid environmental conditions of Alaska, Canada, Greenland, and similar areas. In addition, present equipment with limited winterization for low-temperature operation is inadequate for temperature extremes as low as —65 F. This inadequacy is manifest in the endurance and functioning of the equipment and in the operator's comfort and safety.

Now let us take a look at the military requirements. In Fig. 1, we see the locale of some of the extreme low temperatures recorded on earth. The lowest unofficial temperature recorded on earth was —108 F in Siberia. The lowest temperature recorded in North America was —82 F in the Yukon; the lowest temperature recorded in Alaska was —76 F; and in

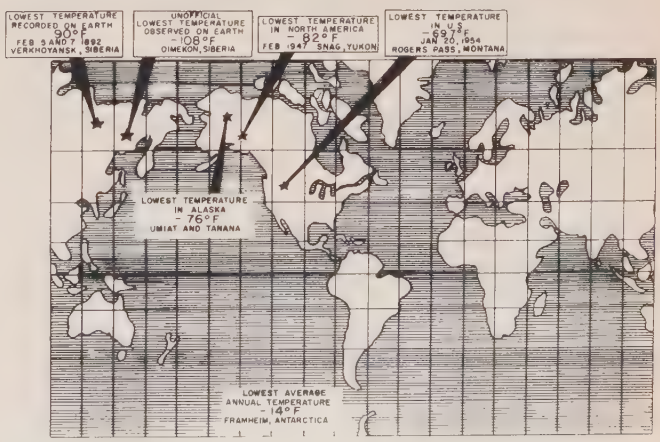

Fig. 1—Location of low-temperature regions of world

SINCE 1954 the CIMTC Subcommittee has been engaged in a program to meet military requirements through industry's production of construction equipment which can give satisfactory cold weather performance down to temperatures of —65 F. Individual contracts for three crawler tractors and one motor grader were negotiated by ERDL for these projects, and their performance is discussed. Industry participation was subsequently expanded to include engineering tests in the cold weather conditions of the Mesabi Iron Range.

This joint report of the Winterization Subcommittee of the CIMTC and ERDL Winterization Section consists of separate papers by various members and consultants of this Subcommittee and ERDL personnel.

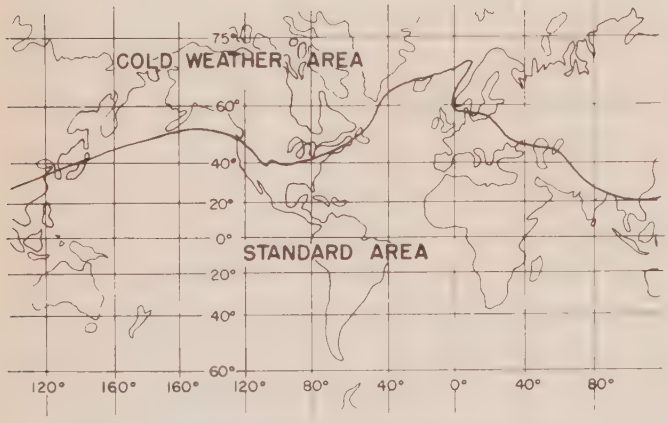

Fig. 2—Southern boundary of cold weather areas

Rodgers Pass, Mont., it was —69.7 F.

In Fig. 2, we see the southern boundary of the cold weather areas of the earth. These are the areas where equipment winterization assistance is required. If we were to superimpose the southern boundary of the cold weather area on Fig. 3, we would see the great land mass of North America and Greenland contained therein. These land masses are critical aerial defense approaches to our country and are currently being guarded by the Continental Air Defense Command (CONARD). Our radar defense screen begins with the Dew Radar Defense running through the Aleutians to Point Barrow, then following just south and parallel of the seventieth parallel to the north of Iceland. To the south following the sixtieth parallel and traversing the land mass of Canada we have the Mid-Canada Network and still further south following the fiftieth parallel we have the Pinetree Network. This vast $3,500,000,000 a year project will require extensive ground support and ground-support vehicles. Fig. 4 illustrates a tractor train supply operation in Greenland. Supply trains such as these as well as various items of construction and electrical power equipment are essential to this endeavor.

Military tactical operations depend upon the engineers carrying out their normal combat, combat support, and technical support missions. To fulfill these requirements, present plans call for modifications to existing equipment. Moreover, specialists are required with troop units to cope with the problems incident to arctic operations. In general, cross-country movement throughout the Arctic is more difficult during the late summer and early fall than in the winter, spring, and early summer. This is explained by the fact that vehicle movement becomes increasingly difficult due to the fact that the ground thaws to its maximum depth just prior to the first freeze. The dense forests of the sub-Arctic put a heavy demand on military construction in the clearing of roads and airfields.

The large number of arctic lakes, streams, swamps, and the muskeg make necessary considerable military bridging and the building of corduroy-type roads. At present, the primary winter mission of engineer troops changes from facilitating movement to providing shelter for personnel, maintenance, and administration activities. In semipermanent bases, considerable engineering effort is directed at clearing snow, operating electric generator sets, water pumps and fire-fighting equipment. Water supply, which is relatively simple in the summer, becomes more complicated in the extreme winter. Needless to say, equipment requires more maintenance as the temperature falls.

The policy of the Department of the Army is to attain the maximum capability of conducting military operations in any area of the world. Limits for satisfactory operation were established as —65 to +125 F. The Engineer Research & Development Laboratories implemented the policy established for the Corps of Engineers, which requires that all equipment must start at temperatures down to —25 F without external aids other than those provided as standard attachments, such as electrical glow plugs and diethyl-ether priming devices. For temperatures ranging from —25 to —65 F, provision must be made to preheat the engine to enable it to start at —65 F within a 1-hr time limit. In addition, provision must be made for special design, special lubrication, and special protection for equipment and operator personnel.

The winterization program at ERDL was directed toward obtaining fundamental information on the characteristics of equipment and material at lower temperatures and on the study of systems and techniques which will permit successful operation at these extremes.

We do not have to go far to visualize a civilian counterpart for equipment winterization require-

ments. In a constant search to augment dwindling mineral, petroleum, and timber reserves, industry has been pushing into the North Country. This push was accelerated during and subsequent to World War II. Interest in the Antarctic has also been increased by Admiral Richard E. Byrd's recent expedition, "Operation Deepfreeze," which began preparations for our country's participation in the International Geophysical Year 1957–1958. Airstrips and other facilities will be built to accommodate and supply scientists who will gather from many nations to conduct coordinated observations in meteorology, geomagnetism, glaciology, and cosmic rays. New frontiers are again being explored, and obviously winterized equipment is needed. To survive economically and to compete on the world market, the industries mentioned above may be required to operate large quantities of construction and industrial machinery all year long.

Alaska's second largest industry is minerals. Over $1,000,000,000 have been extracted, and many sources remain to be discovered. The United States Bureau of Mines and the United States Geological Survey are continuing explorations to uncover new deposits. Table 1 gives some of the developed and undeveloped Alaskan minerals.

In the Northwest Territories of Canada, intense geological exploration has resulted in the discovery of rich mineral deposits over a vast area. An immense deposit of very rich radium-bearing pitchblende and silver is being worked at Eldorado at the east end of the Great Bear Lake, close to the Arctic Circle. This is the most important development in mineral industries in the territories. From here, rich radium concentrates are shipped 4000 miles to the refinery at Port Hope, Ontario. This is only one of the major industries rising in the cold regions of the Canadian North.

More recently, we have heard of the far-visioned plan of Admiral Richard E. Byrd for the Antarctic. Here is envisioned a vast deep freeze for indefinite storage of surplus foods and materials. It has been established that within the Antarctic lies a great wealth of minerals. All of this, and more, may be available to industry and the world if we can build the machinery to develop it safely and economically.

For the most part, it appears that the problems for military and civilian operation in the Arctic are basically the same. The main differences may be found in the standby and emergency character of military equipment, which is usually geared to the maximum performance that industry can reasonably produce. We do feel that the problem of winterization is a mutual problem of the military and civilian user and will increase significantly in the future.

The corps of Engineers has always been associated closely with industry because of its missions. Consequently, it uses civilian products with slight modifications. It is dependent upon the functioning of these products and upon the ability of the manufacturers to anticipate its needs. Our country is dependent upon its industry for its defense. Industry, knowing the needs, can quickly supply the equipment. In this field of winterization as in other fields, the Corps of Engineers wishes to continue these close relations with industry. Recognizing this, the Corps of Engineers requested that the SAE CIMTC Steering Committee set up a Subcommittee to study the winterization of construction equipment and the possibilities of standardizing the winterization equipment used on construction machinery.

In January, 1952, Subcommittee XV was organized under the Chairmanship of J. E. Jass, Cater-

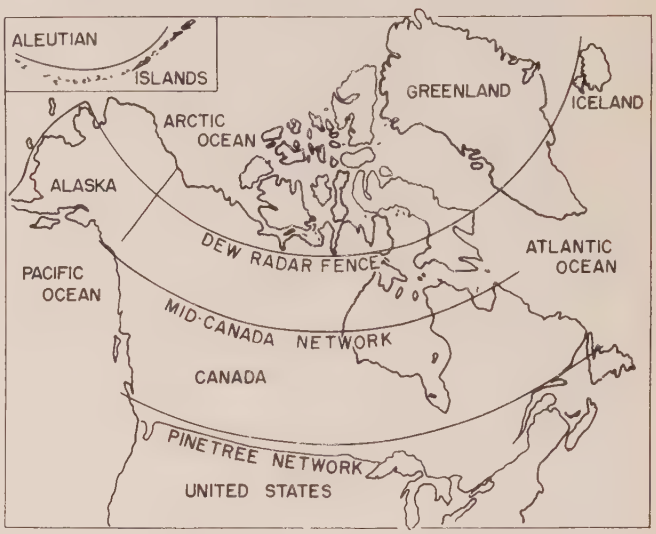

Fig. 3—Radar defense screen set up by CONARD for North America and Greenland

Table 1—Alaska's Mineral Deposits

Antimony	Iridium	Platinum
Arsenic[a]	Lead	Palladium
Asbestos	Magnesite[a]	Pumice
Barite[a]	Mercury	Silver
Bismuth[a]	Nickel[a]	Tin
Bituminous Coal	Osmium	Tungsten
Copper	Petroleum Reserves (USN)	Uranium
Gold		Zinc

[a] Undeveloped

Fig. 4—Tractor train operation in Greenland

Table 2—Manufacturers Providing Equipment

Austin-Western Co.	99L Motor Grader
Caterpillar Tractor Co.	D7 Tractor
International Harvester Co.	TD-18 Tractor
Allis-Chalmers Mfg. Co.	HD-16A Tractor

pillar Tractor Co., and P. W. Espenschade as liaison representative for the Corps of Engineers. In July, 1954, T. A. Haller, (formerly of Allis-Chalmers Mfg. Co), succeeded Mr. Jass until October, 1955, at which time M. G. Mardoian, International Harvester Co., became Chairman of the Subcommittee and is serving in that capacity at the present time.

The committee has been a very active group, with meetings held on the average of once a month. It is our firm belief that only through cooperative efforts of industry and the military will a satisfactory solution to the winterization of construction equipment be found.

It was recommended in the Subcommittee's Progress Report No. II that much of the committee's work could be evaluated best if it were possible to build a number of prototype construction units incorporating the thinking of the committee on many phases of its work, and that perhaps the U. S. Army Corps of Engineers could initiate a program to have several items of construction equipment built and tested under both cold-room and field conditions. This recommendation was accepted by the Corps of Engineers. Table 2 gives manufacturers providing the winterized crawler tractors and motor grader.

Each manufacturer was to design a different approach to winterization. This approach was to be coordinated closely with the Winterization Subcommittee in order that mutual benefit and a minimum of duplication could be derived from each development. In order to permit a maximum freedom of design, only certain basic requirements and military standards were required of the suppliers. These requirements were as follows:

1. Each of the three tractors was to be equipped with SAE CIMTC Subcommittee I (Standardization of Drawbars and Tractor Equipment Mountings) current standards. These consisted of SAE Standard Rear-Power Take-Off and Mounting Face and SAE Standard Bulldozer Trunnion Mountings, plus one pair of additional holes forward on these latter mountings to allow the use of present standard dozers. Tractors were supplied with hydraulic dozers having the standard moldboard bolt-hole spacing, and two drum rear-mounted cable power control units. The grader was equipped with the standard Austin-Western Co. commercial snowplow, scarifier, and grader blade attachments.

2. All items were to be equipped with an insulated engine enclosure designed to prevent weather elements, particularly blowing snow, from affecting the operation of mechanical controls, linkages, and accessories sensitive to moisture or to clogging with ice or snow.

3. A heating system was provided with suitable controls to provide proper temperature of the engine assembly and storage batteries prior to starting. (Heating of the battery electrolyte is essential for efficient battery charging and discharging at low temperatures.)

4. A heated cab was furnished for personnel heating to +35 F with ambient temperatures of −65 F. In addition, adequate window defrosting was required.

5. A suitable means of escape through a top escape hatch was provided. This provision was made for instant exit should the vehicle fall through river ice or into an unexposed crevasse.

6. Accessibility to controls, adjustable devices, and valves together with simplification of maintenance for operators handicapped by bulky clothing were to be given full consideration.

7. "Slave" electrical receptacles conforming to MIL-C-13993C were provided. Where applicable, cables were of adequate capacity for external engine cranking as an emergency feature should self-contained batteries become ineffective.

8. Wherever possible, all hose and fabrics were certified for operation between −65 and +125 F and for prolonged storage at −80 F.

Table 3—Stock Military Fuels, Lubricants, Hydraulic Fluids, and Antifreeze for Winterized Construction Equipment

Product	Stock No. and Container	Federal Item Identification No.	Kinematic Viscosity, cs		Applicable Temperature Range			
			Maximum	Minimum	+32 to −25 F Specification	Symbol	−25 to −65 F Specification	Symbol
Fuel Oil, Diesel, Arctic Grade		286-5284		1.4 to 4.0 at 100 F	VV-F-800	DF-A	VV-F-800	DF-A
Gasoline, Auto, Combat (91 octane)	07-O-00146-050 (55 gal)	221-0685			MIL-G-3056A, Type II		MIL-G-3056A, Type II	
Lubricating Oil, Internal Combustion Engines, Subzero	14-O-02375-075 (5 gal)	242-7603	8500 at −40 F	5.75 at 210 F	MIL-O-10295, Amd 1	OES	MIL-O-10295, Amd 1	OES
Lubricating Oil, Gear, Subzero	14-L-00184-935 (15 gal)	257-5441	30,000 at −65 F	3 at 210 F	MIL-L-10324(ORD)	GOS	MIL-L-10324(ORD)	GOS
Greases:								
Aircraft and Instrument (for Sealed Bearings)	14-G-00611-010 (1 lb)	261-8298			MIL-G-3278	GL	MIL-G-3278	GL
Automotive and Artillery (for Chassis Elements)	14-G-00623-055 (25 lb)	190-0906			MIL-G-10924, Amd 3	GAA	MIL-G-10924, Amd 3	GAA
Automotive (Water Pump)	14-G-01384-001 (1 lb)	235-5503			VV-G-632, Type A, Grade 4	WP	VV-G-632, Type A, Grade 4	WP
Hydraulic Fluid, Non-Petroleum, Arctic Type (Brake System)	51-F-361-10 (1 gal)	252-6375	500 at −65 F	1.0 at 210 F			MIL-H-13910	
Hydraulic Fluid, Petroleum Base	14-O-00882-315 (1 gal)	223-4134	500 at −40 F	10.0 at 130 F	MIL-O-5606	OHA	MIL-O-5606	OHA
Lubricating Oil, Instrument	14-O-01342-000 (1 qt)	223-4129	2000 at −40 F	8.0 at 130 F	MIL-L-6085	OAI	MIL-L-6085	OAI
Antifreeze, Arctic-Type	51-C-1553-755 (55 gal)	243-1991	1400 at −65 F		MIL-C-11755		MIL-C-11755	

9. All drive belts were furnished to conform to MIL-B-11040, Type I or II, Class A.

10. Equipment lighting was to be 24-v design and arranged and installed to conform to the latest military specifications.

In addition to these requirements, each item was to be serviced during manufacturing assembly with arctic military fuels, lubricants, hydraulic fluids, and antifreeze in accordance with the recommendations tabulated in Table 3. This feature precluded the costly and time-consuming practice of disassembly, cleaning, and relubricating with special arctic lubricants.

It may be important to recall that under extreme low temperatures, conventional diesel fuels, lubricants, and hydraulic fluids become frozen solid, rendering all engine and vehicle movement impossible. When such items are furnished under the properly indicated specification, the fluidity and lubricity become adequate for effective and safe operation. Specifications for arctic gasoline provide sufficient volatility to assure adequate vaporization for engine starting and efficient operation.

The arctic-type antifreeze is premixed and does not require or permit the addition of water. This precaution assures that an optimum eutectic mixture is retained. It further eliminates one more precaution for inexperienced operators operating under adverse conditions. A typical analysis of antifreeze provided under this specification might contain 45% ethylene glycol, 31% water, 22% dimethoxethenol, and 2% borax inhibitor.

Subsequent to assembly and servicing, each item was tested in the cold room at −25 and −65 F to determine if the following requirements could be met:

1. Satisfactory cold starting at −25 F without application of engine heater.

2. Starting at −65 F within not more than a 1-hr time limit during which time the engine heaters were employed to preheat the engine.

3. Maintenance of a minimum cab air temperature of +35 F and adequate defrosting of all windows with an outside ambient temperature of −65 F.

For logistical reasons, it was decided to conduct engineering field tests in the United States during the winter of 1955–1956. Subsequently, troop service tests were to be conducted in Alaska during the winter of 1956–1957. The engineering tests could thus serve as a "shake-down" test so that various deficiencies could be corrected with a minimum amount of lost time. Thus, in the following year, the winter tests in Alaska would be with tractors more nearly assured of meeting requirements.

Out of the five different test sites considered in the United States, the location in the vicinity of Aurora, Minn. at the Erie Mining Co. site, was selected because it had the coldest temperatures and because the Corps of Engineers successfully conducted tests last winter at this location on a winterized 20-ton truck-mounted crane.

During the months of January and February, 1956, and under the control of the Erie Mining Co., the motor grader was used to surface truck haul roads, and the tractors operating as dozers were used for road building and clean-up work in the taconite pits. Without regard to ambient temperatures, the engine heaters and personnel heaters were cycled each morning. These operations served as limited durability tests and as general suitability tests for winter operations.

A meeting of the Winterization Subcommittee was arranged at the test site to give the group an opportunity to observe operation of the units. Inspection teams were formed during this meeting in which personnel present were assigned tasks to observe and report their findings on the various features of the winterized machines. Although the lowest recorded temperature encountered during the Minnesota field tests was only −25 F, it was believed that, coupled with cold-room tests at −25 and −65 F, adequate information would be available to prepare this joint paper so that others could benefit from our experiences.

We have found that the winterization field offers many challenging problems. A few of these are as follows:

1. Low-temperature engine starting.
2. Maintenance of engine operating temperatures.
3. Human engineering.
4. Low-temperature polymers.
5. Low-temperature metallurgy.
6. Lubrication.
7. Fuels.
8. Vehicle traction and flotation.

In the following papers you will see how we have approached some of these problems. These winterized test machines are equipped with special winterization components for evaluation purposes and are not necessarily production units. The papers that follow will describe the four items of construction equipment which were integrally winterized, cold-room tested, and field tested.

Description of Winterization System on Austin-Western Model 99L Motor Grader

by

John A. Klisch

Engineer Research & Development Laboratories

General Description

One of the winterized items procured under this program was an Austin Western Model 99L Motor Grader manufactured by the Austin-Western Works, Division of the Baldwin-Lima-Hamilton Corp. of Aurora, Ill., as shown in Fig. 5. In addition to the winterization, the grader was furnished with standard 4-wheel drive, 4-wheel steering, hydraulic con-

Fig. 5—Winterized Austin-Western Model 99L Motor Grader

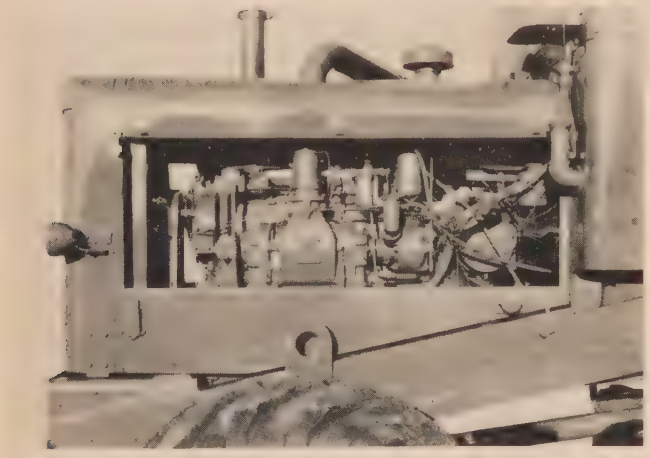

Fig. 6—Right-side view of engine compartment illustrating application of ether priming

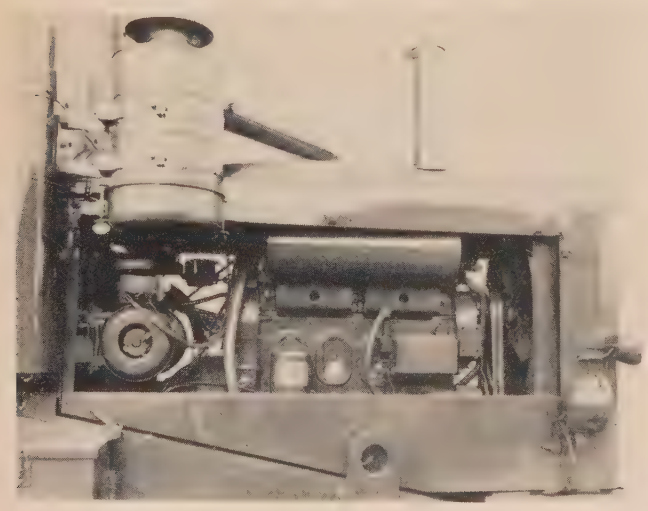

Fig. 7—Left-side view of engine compartment illustrating engine coolant heater

trols, a 115-hp Detroit diesel, No. 4-71, 2-cycle engine, a front scarifier, and V-type snowplow.

Priming Aids

To aid in the starting of the engine at low temperatures, a GMC bulk-type ether-priming system was used as shown in Fig. 6. This consists of a 1-pt container and a hand-operated primer that injects 2½ cc

Fig. 8—View of cab showing thermal and acoustical insulation

of ether per stroke into the air intake chamber. With this device the time and rate of ether injection is controlled by the operator.

Powerplant Heating

To preheat the engine block and crankcase oil when starting at low temperatures a Perfection E-511, (24-v) coolant heater as illustrated in Fig. 7 is used. This heater has an output rating of 60,000 Btu to the coolant and 30,000 Btu to the contaminated air. The heated coolant is pumped through the engine block and then through a heat exchanger that is installed in the oil pan.

A mastic insulation, known as Permalastic, having a "K" value of 0.70 was applied to the engine housing.

Cab Design and Heating System

The cab shown in Fig. 8 was insulated with the thermal and acoustical-type insulation. This is a sandwich-type insulation consisting of an adhesive, fiber glass, and a perforated sheet-metal surface. The function of this insulation is to reduce both the heat loss and sound level of the cab, the idea being to provide a better atmosphere for the operator. This in turn reduces operator fatigue and increases the efficiency of both operator and machine.

The floor was covered with a sponge-type material, Ensolite, which has a "K" factor value of 0.25. An escape hatch is built into the top of the cab to permit escape of the operator in event of an accident. Win-

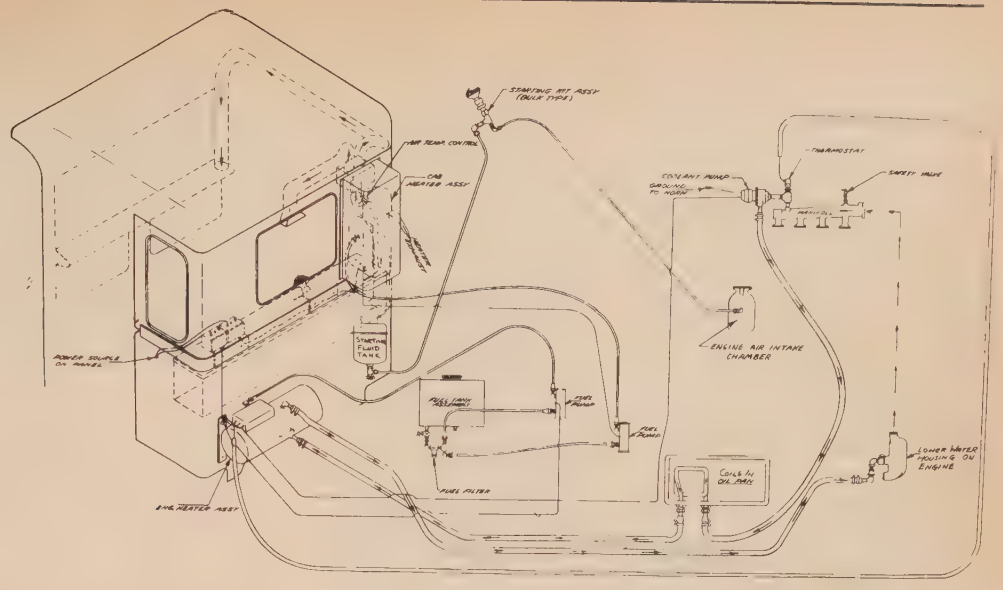

Fig. 9—Winterization scheme

dows are tinted safety glass. Cab heating was accomplished by using a Perfection E-500, 24 v, gasoline-burning fresh air heater having an output of 30,000 Btu to the fresh air. All heater controls were mounted in the cab, and heaters were protected by overheating devices.

Basic Cranking System

A 24-v, conventional, heavy-duty electrical starting motor is used for engine cranking.

Electrical System

Four, 6-v 4H, 150 amp-hr batteries, conforming to military requirements, were connected in series to provide 24 v. Hot fresh air from the personnel heater was used to heat the batteries.

1. The coolant heater is located in the engine compartment. The coolant flow, as can be seen by the direction of the arrows, is from the heater to the engine block, then through the heat exchanger in the oil pan and back to the heater.
2. The personnel heater is mounted in a housing on the rear of the cab. Hot air is ducted to the front and rear windows. This method is intended for both personnel heating and window defrosting.
3. A separate hot air duct is provided to furnish hot air for battery heating. The battery box is located on the inside-rear section of the cab.
4. The ether priming pump is located on the back seat; the bulk ether container is mounted in the engine compartment.

Modifications

Subsequent to the Mesabi Iron Range tests, the winterization system on this grader was modified at ERDL, Fort Belvoir, Va. (See Fig. 9.) Some of the minor modifications consisted of the installation of coolant line shut-off valves, coolant drain valves, hinged cover for the personnel heater, and dry-type engine air cleaner for low-temperature operation.

Fig. 10—Quick opening and locking escape hatch mechanism in locked position

The major modifications consist of the installation of a sturdier winterfront and a new escape hatch mechanism. The escape hatch mechanism was developed at Fort Belvoir and is being patented. The

Fig. 11—Quick opening and locking escape hatch mechanism in opened position

Fig. 12—Operator escaping through hatch

Table 4—Specifications of TD Series 182 Crawler Tractor (74-in. Gage with 20-in. Track Shoes)

Horsepower, Corrected Drawbar	103
Belt	117
Drawbar Pull, lb	
	At Rated Governed Engine Speed / At Engine Speed of Maximum Torque
First Gear	24,300 / 27,450
Sixth Gear	6100 / 6900
Speeds Computed at Rated Governed Engine rpm	mph / fpm
First Gear	1.6 / 141
Sixth Gear	5.5 / 486
Engine Clutch-Single Plate, Over Center, with Automatic Clutch Brake.	
Transmission-Two Range Selective Sliding Gear Type.	
Steering Clutches-Hydraulic Power Booster Controlled (One for Each Track)	
Engine 4-Cycle Diesel, Precombustion-Chamber Design Valve-In-Head Type with Built-In Gasoline Conversion All-Weather Starting.	
Net Horsepower at Engine Flywheel at Sea Level	124
Number of Cylinders	6
Bore and Stroke, in.	4¾ x 6½
Piston Displacement, cu in.	691
Rated Governed Speed, rpm	1450
Maximum Torque, lb-ft	507
Lubrication-Crankshaft, Connecting Rods, Camshaft, Piston Pins, and Valve Rocker Levers	Full Pressure
Crankshaft, Number of Mainbearings	7
Electric Starting and Lighting, v	24
Operating Weight (approximate lb) Bare Tractor	29,050

primary purpose of this mechanism is to enable the operator to open the escape hatch in event the hatch cover is frozen or stuck tight.

Fig. 10 shows the mechanism in the closed position. It is so designed that by applying a 100-lb force to the pulldown bar, a 2500-lb force is exerted upon the hatch cover thus forcing it outward. This 2500-lb force alone is sufficient to break loose the hatch cover even when it is covered with approximately 1 in. of ice. However, to assist in ejecting the hatch cover, the cover is loaded with four coil springs. Thus, when the operator trips the opening mechanism, the hatch cover is immediately forced outward permitting the operator to escape. Fig. 11 shows the mechanism in the open position.

The operator climbs through the hatch as shown in Fig. 12. The hatch cover is entirely free from the cab. This allows the operator to proceed either to the front, back, or sides in the event of an accident such as upsetting, falling into a ravine, or when breaking through ice or snow on a trail.

Fig. 13 is a view of the grader undergoing cold-chamber tests at ERDL, Fort Belvoir, Va. The escape hatch is covered with approximately 1 in. of ice.

The operator has opened the hatch, breaking the built-up layer of ice and is proceeding to climb through the hatch opening as shown in Fig. 14.

The escape hatch and its operating mechanism are strictly safety features which are designed not only to permit the operator to escape but also to provide him with a better sense of security.

Throughout this entire program, you will notice that we are chiefly concerned with proper engine operation and with providing a better environment for the operator. This of course pays dividends, since in return, we receive a greater overall operating efficiency.

Description of Winterized TD-18 Tractor

by

M. G. Mardoian

International Harvester Co.

A TD-18 series 182 diesel crawler tractor with the specifications as shown in Table 4, was winterized for the United States Corps of Engineers to operate at ambient air temperatures as low as —65 F.

The tractor was also built to accommodate SAE interchangeable tractor attachments, and to incorporate the following features:

1. Front radiator support group with sheaves and cable cover tube with flange for connecting to either front or rear power control unit, including lower

sheave block and necessary cable for connection to rear cable control unit.

2. SAE rear attachment mounting and drive facilities.

3. SAE track frame mountings modified by adding two extra holes to accommodate standardized dozer mountings.

4. Front-mounted hydraulic power control unit including hydraulic rams, hoses, and additional equipment to operate interchangeable dozers.

5. Angle dozer with SAE trunnion mountings and mold board bolt-hole spacing, cable, and hydraulic control mounting brackets suitable for use with the standardized tractor with cable or hydraulic power control unit. The minimum blade lift of the dozer, when mounted on the tractors was not less than 36 in.

6. SAE standard mountings for rear-mounted single-drum winch.

7. A 2-drum rear-mounted cable control unit with dozer cable for use on the TD-18, as well as on the standardized tractor, with latest SAE rear mountings and drive facilities.

This tractor is illustrated in Fig. 15.

Priming Aids

No priming fluids such as diethyl ether are required to enable starting of the TD-18 diesel engine, because it employs a gasoline starting system which is integral with the engine as shown in Fig. 16.

The design incorporates an auxiliary combustion chamber above each cylinder which is opened to or sealed from the cylinder by a valve. These starting valves are actuated by an operator-controlled linkage. The compression ratio of the diesel engine (15.04/1) is therefore much lower (6.5/1) when the starting valves are open and the auxiliary chambers are connected to the main combustion chambers. When starting the diesel engine the operator first pulls back the cycle changeover lever (1) to the gasoline starting position which does four things:

1. It actuates the starting valve linkage (2), which opens the starting valves (3).

2. It energizes an electrical ignition system which is isolated during diesel operation. The spark plugs (4) of this ignition system are located in the auxiliary combustion chambers (5).

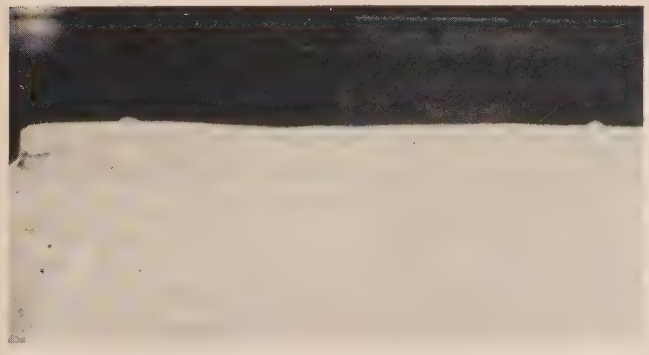

Fig. 13—Hatch covered with approximately 1 in. of ice

3. It opens the gasoline shut-off valve located in the carburetor (6).

4. It closes the butterfly valves (7) located in the intake manifold (8), bypassing the air through the carburetor and into the gasoline cycle air passage (9), past the regular intake valves, and into the combustion chambers, as indicated by the arrows in Fig. 16.

The engine is then started in the conventional manner by the electric cranking motor.

After a brief period of operation on the gasoline

Fig. 14—Hatch opened with operator proceeding to escape

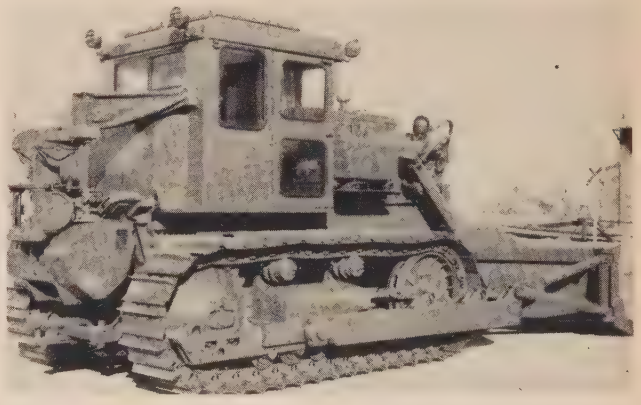

Fig. 15—International Harvester TD-18 series 182 winterized tractor

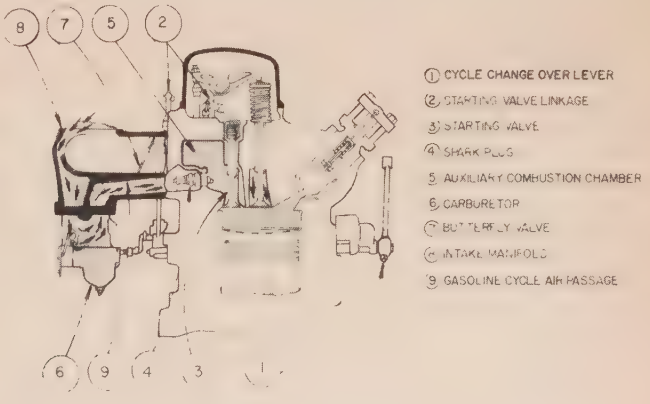

Fig. 16—Gasoline cycle

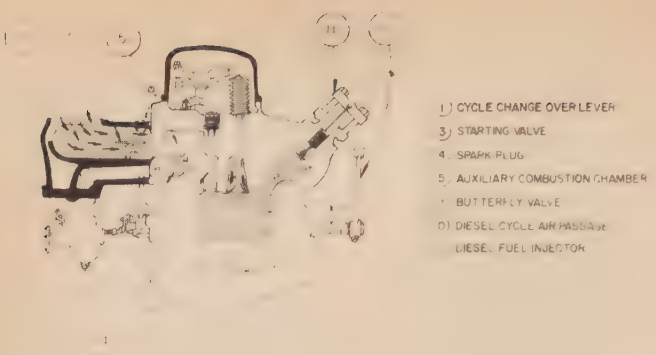

Fig. 17—Diesel cycle

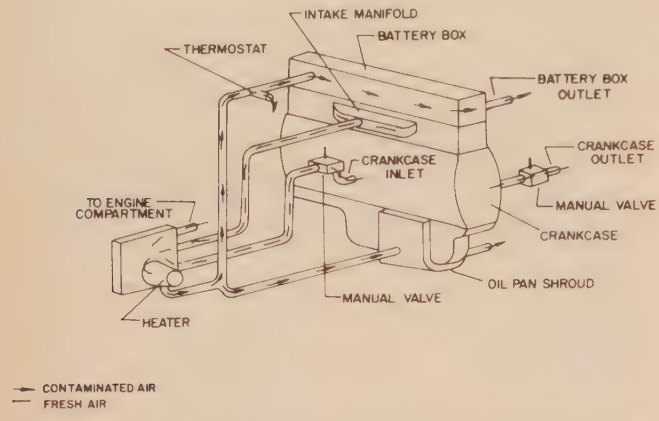

Fig. 18—Engine and battery-box heating system used on initial winterized tractor

Fig. 19—Cab design of winterized tractor

cycle (1 to 4 min, depending on the ambient air temperature), the operator returns the cycle change-over lever (1) forward to the diesel position and advances the speed-control lever to the operating speed desired. The engine then runs on diesel fuel with the ignition circuit grounded (Fig. 17).

When the lever (1) is moved to the diesel position, the starting valves (3) close and isolate the auxiliary combustion chambers (5) from the cylinders, thus increasing the compression ratio. The electrical starting system (4) is switched out of the circuit by an ignition cut-out switch in the manifold and the gasoline shut-off valve is moved to the closed position. The butterfly valves (7) in the intake manifold diesel-cycle air passage (10) are opened, passing air directly from the air cleaner into the engine cylinders as indicated by the arrows in Fig. 17.

This winterized unit was provided with a gasoline hand-primer attachment which directs gasoline through nozzles into the intake air passage towards the intake valves. Its use prior to cranking, provides a definite aid in reducing cranking time before a "first fire" is obtained, thus reducing the battery current drain.

Two Chevron pressure primers were installed for test purposes on the diesel cycle and were not intended as a starting aid on the gasoline cycle. It was thought that possibly their use might be required at −65 F to obtain improved firing characteristics when the engine was first changed over from the gasoline to the diesel cycle; however, tests revealed that if the engine were operated on the gasoline cycle for the specified time, the Chevron pressure primers were not required.

Powerplant Heating System

A Perfection E-510 series fresh air heater utilizing gasoline fuel was employed for heating. This heater is designed to provide, on "high" position, a fresh air output of 60,000 Btu per hr and a contaminated gas output of 30,000 Btu per hr. It was located on the right side of the tractor just ahead of the cab and was provided with a sliding cover over its air inlet to prevent entry of snow when the unit is parked. Fig. 18 illustrates the powerplant heating system used on the initially designed winterized tractor.

The heater fresh air output was ducted to a plenum chamber and from there divided equally into three branches and directed to the following:

1. Engine compartment to warm up the air surrounding the engine.

2. Intake manifold shroud, after which it was exhausted to the engine compartment. Heating the intake manifold helped to vaporize the gasoline during the start attempt.

3. Into the engine crankcase permitting the engine to warm up internally, directly heating the engine oil, cylinder walls, crankshaft main bearings and journals, bottom of pistons, and the like, instead of attempting to pass through the barrier of the crankcase when applying heat externally. Greater use of the available heat can be made when the engine is heated internally and results in higher cranking speeds. A valve located at the crankcase hot air entrance prevented oil splash from entering the heat duct when the engine was operated. Another valve was located at the oil filler cap to permit venting of the heated fresh air entering the engine crankcase on the opposite side. This larger vent was provided in addition to the regular engine breather pipe for increased ventilation. Both valves, operator-controlled, were

opened during engine preheating and closed after the engine was started.

Half of the heater contaminated gases were employed to heat the batteries, and the remainder was directed to an enclosed shroud over a portion of the oil pan, after which it was exhausted to the atmosphere.

One double-walled battery box accommodated two 12-v batteries. The outer shell was of hot rolled steel, while the inner shell was of stainless steel for corrosion resistance against battery acid. The contaminated gases were passed through the double walls of the box to heat the batteries. The box projected part way up out of the engine hood sheet, and this exposed area was insulated with a fused spun glass material.

To permit transfer of heat from the engine compartment to battery case during tractor operation, that portion of the box projecting under the hood sheet was not insulated. A thermostatic control unit regulated heat flow through the battery box to avoid battery overheating. Two drain holes were provided in the box to facilitate cleaning.

Cab Design

A cab design and crew compartment heating system must also be considered for a winterized tractor (See Fig. 19.)

All of the cab safety glass windows were tinted to reduce sun glare. The rear windows were mounted on channels to permit them to slide open for ventilation if desired.

The cab was mounted on strip rubber placed all around the base for shock absorption. From past experience with insulating materials, the cab interior was insulated with a mastic insulation, "Permalastic," mixed in the ratio of two parts by volume of the mastic to one part of 10–20 mesh granulated cork.

An escape hatch covering most of the cab roof area was made to accommodate two men. A "panic bar," located near the front ceiling of the cab unlocked the hatch. It employed a high-mechanical-advantage cam arrangement to break any ice formations.

A second Perfection E-510 fresh air heater located on the left side just ahead of the cab provided heat for personnel and defrosting of the cab windows. The contaminated gases from this heater were not utilized; however, provision was made to duct this heat to the battery box to supplement that provided by the engine heater. The door window defroster nozzles were designed to mate with the heat duct when the doors are closed as indicated in Fig. 20.

The cab heating and window defrosting system is shown in Fig. 21, and was designed to defrost all of the cab windows.

Distribution of the heated air to individual defroster nozzles could be controlled by manually operated damper valves. The cab heater fresh air output was directed to a main plenum chamber and from there distributed to the various defroster nozzles; however, manually operated adjustable louvers located in the plenum chamber were provided to permit heat directly into the cab.

Basic Cranking and Electrical Systems

The basic engine cranking system consisted of a 24-v electric cranking motor incorporating a manually engaged overrunning clutch drive.

The tractor employed a 24-v electrical system con-

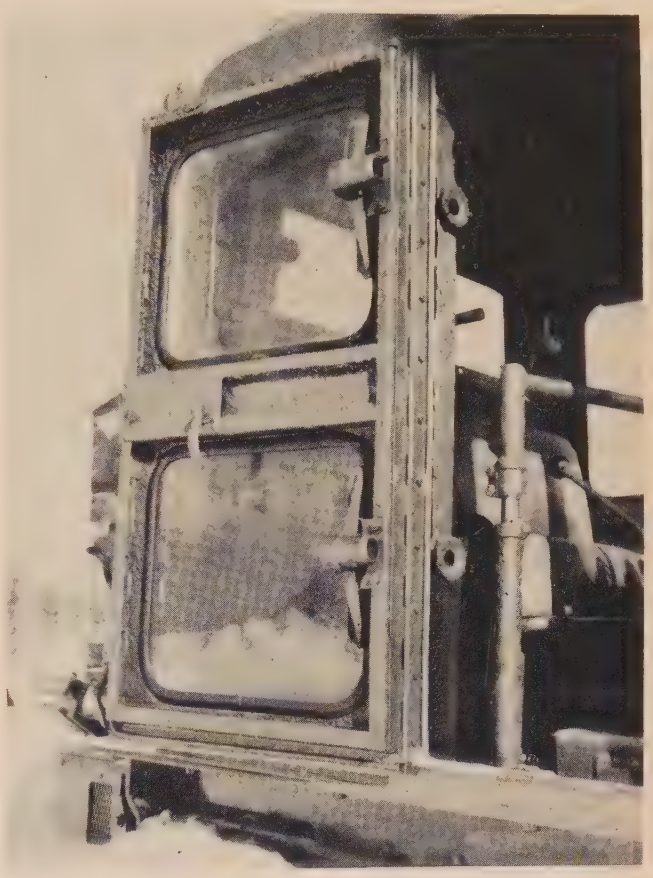

Fig. 20—View showing defrosting ducts on cab doors

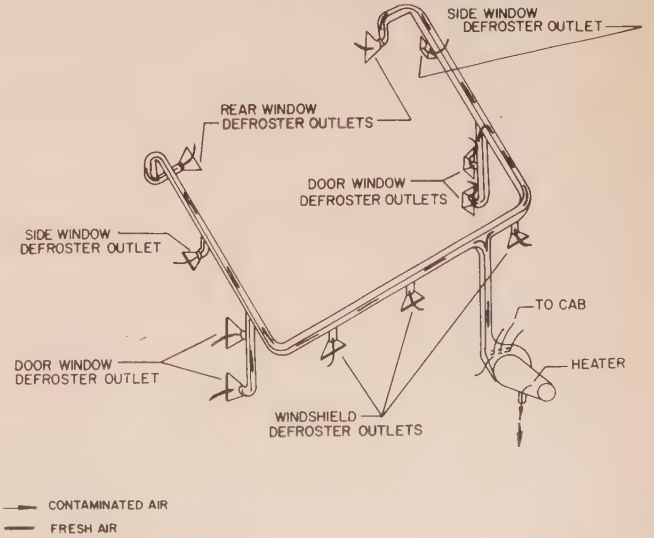

Fig. 21—Cab heating and window defrosting system

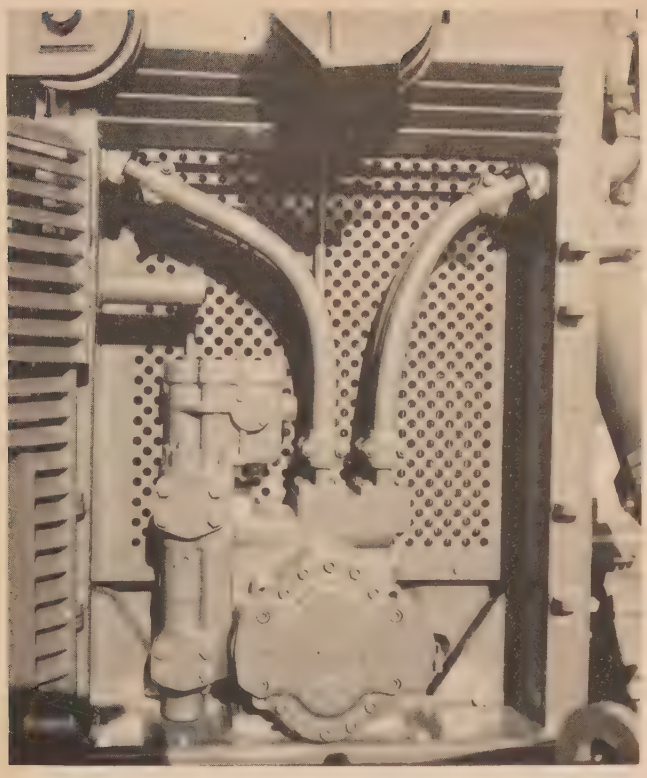

Fig. 22—Manually operated sliding plate winterfront

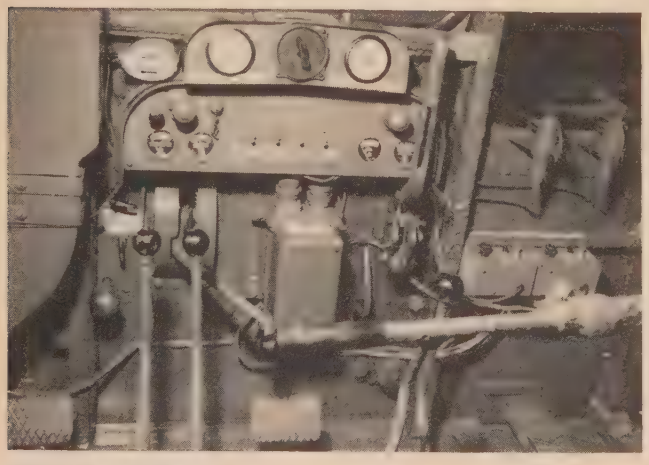

Fig. 23—Thermocouple selector switch and gages for regulating battery-box temperature

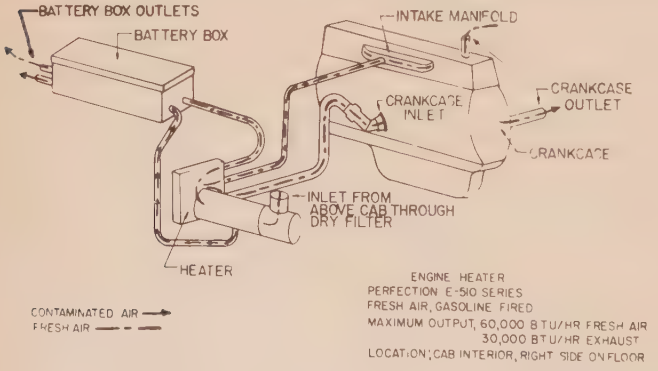

Fig. 24—Redesigned engine and battery-box heating system

forming to military specifications and was powered by two 12-v, 8T arctic-type batteries mounted on strip rubber for shock absorption.

Special Materials

A manually operated sliding plate winterfront, as illustrated in Fig. 22 was employed for engine coolant temperature control. A bell-crank linkage operated from the cab permitted high mechanical advantage to shear off any ice or snow formations.

The air-cleaner inlet pipe was cut off so that air was drawn from the warmed engine enclosure, thereby preventing the entrance of snow into the air cleaner. The hood sheet had a removable cover so that an air pipe could be installed later for temperate zone operation. A manually operated vent on the hood side cover was provided to insure adequate intake air to the engine after it was started.

Instrumentation was provided to determine temperature measurements of the engine coolant, oil, battery-box air temperature, and the like during the field tests. Fig. 23 shows the themocouple selector switch and two gages provided to indicate battery-box air temperatures.

The cab doors and battery-box sealing members were of silicone rubber strips riveted in place. The sealing members between the cab and diesel fuel tank were also of silicone rubber strips. A synthetic, cellular-type rubber was used for the floor mat.

Based on the above initial design winterized TD-18 tractor, the following significant deficiencies were noted:

1. The heaters, box-encased, were mounted one on each side of the engine just ahead of the cab on heavy brackets which also formed the lower part of the engine enclosure. This arrangement was not conducive to engine accessibility for servicing and maintenance.

2. The vertical duct which was part of the cab window defrosting system was located at the juncture of the left and center windshield panes. The battery box protruded partially above the hood line. Each of these factors, to some degree, resulted in impairment of the operator's visibility.

Realizing that the tractor would be further tested in Alaska during the winter of 1956–1957, our company decided to correct these major deficiencies and to make further improvements based on the experience gained in the Minnesota field tests and cold-room tests.

The following describes the tractor winterization system redesign:

1. The original cab was replaced with a wider one which provided maximum interior space and operator visibility.

2. The front windows had a reverse tilt.

3. Visibility through the windshield was not obstructed by defroster ducts.

4. An underhood engine muffler was installed to improve visibility further.

The engine and personnel heaters which were previously mounted respectively on the right and left side of the engine compartment were removed and relocated. This cleared the area along both sides of the engine permitting increased accessibility into engine compartment and improved visibility of blade and tracks. Fresh air intake to the engine heater was taken from above the cab and filtered to insure a clean air supply when the engine was internally heated.

Although satisfactory starting was achieved at —65 F with the initially designed heating system, cold-room tests showed that the available heater output could be utilized to greater efficiency by its redistribution to the various components. (See Figure 24.)

The engine heater fresh air output was ducted to a plenum chamber, and from there divided in various proportions by selecting proper duct sizes and then directed to the following:

1. Internally into the crankcase.
2. Intake manifold.
3. Battery box.

All of the engine heater contaminated gases were ducted to the insulated battery box. In addition to the contaminated gases, which were passed through the hollow walls of the battery box, a portion of the heater fresh air, as mentioned above, was directed to the space over and between the batteries and the battery-box inner walls.

The battery box was moved from the engine hood sheet location to the cab to augment visibility. This location also protected the batteries from cold air and helped to keep them warm so they could accept a charge during tractor operation.

A cover sealed the batteries from the crew compartment. All heat, both contaminated gases and fresh air supplied to the battery box were vented to the outside of the cab.

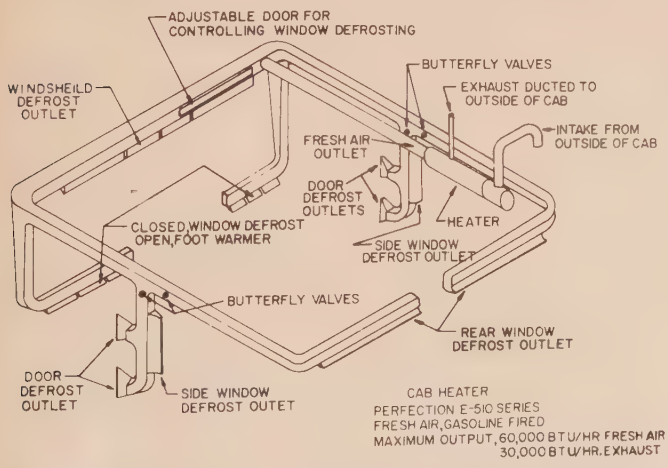

Fig. 25—Redesigned cab heating and window defrosting system

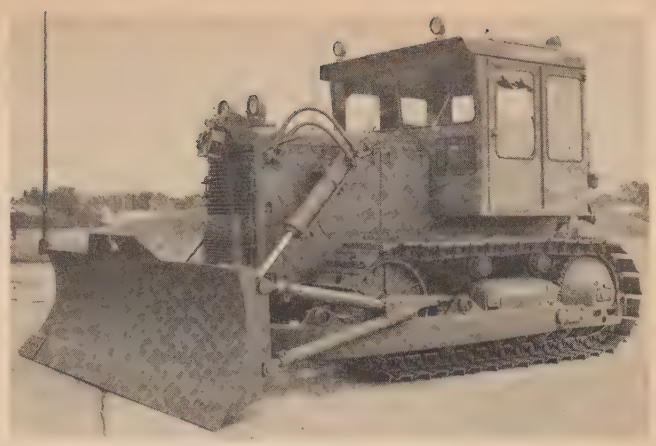

Fig. 26—Redesigned winterized International TD-18 crawler tractor

During tractor operation, the engine heater is not operated; however, the personnel heater is utilized for operator comfort. Contaminated gases from the personnel heater were not utilized and were ducted to the outside of the cab. Fig. 25 illustrates the redesigned cab heating and window defrosting system.

A cab door seal material, which permitted greater flexibility at the lower temperatures was provided.

After the various improved winterization design changes were made on the TD-18 tractor, we believe that it will meet arctic operation requirements.

Fig. 26 illustrates the redesigned winterized International' TD-18 crawler tractor which the United States Army Corps of Engineers will test in Alaska during the winter of 1956–1957..

Description of Winterization System on Caterpillar D7 Tractor

by

T. H. Fones

Caterpillar Tractor Co.

General Description

A standard D7 tractor was winterized for the Corps of Engineers to meet prescribed requirements for starting and operating at ambient temperatures as low as —65 F. This tractor is rated at 102 dbhp and is powered by an 831-cu in. displacement, naturally aspirated engine. The standard tractor basic weight is 26,930 lb; the fully winterized tractor with hydraulic dozer and rear-mounted cable unit weighed approximately 40,000 lb.

The winterization features visible in Figs. 27, 28, and 29 are as follows:

1. The enclosed engine compartment to minimize the effect of weather elements, particularly blowing snow on the operation of mechanical linkages, controls, and accessories which might be sensitive to moisture or clogging from snow.
2. The Perfection E-500 fresh air heater which

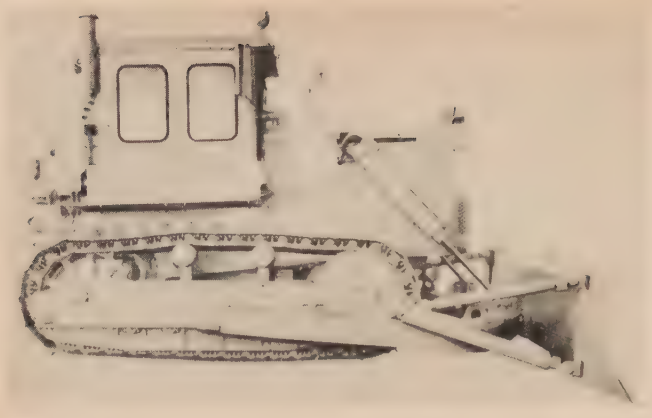

Fig. 27—Engine enclosure, cab, battery box, and perfection heater on winterized D7 tractor

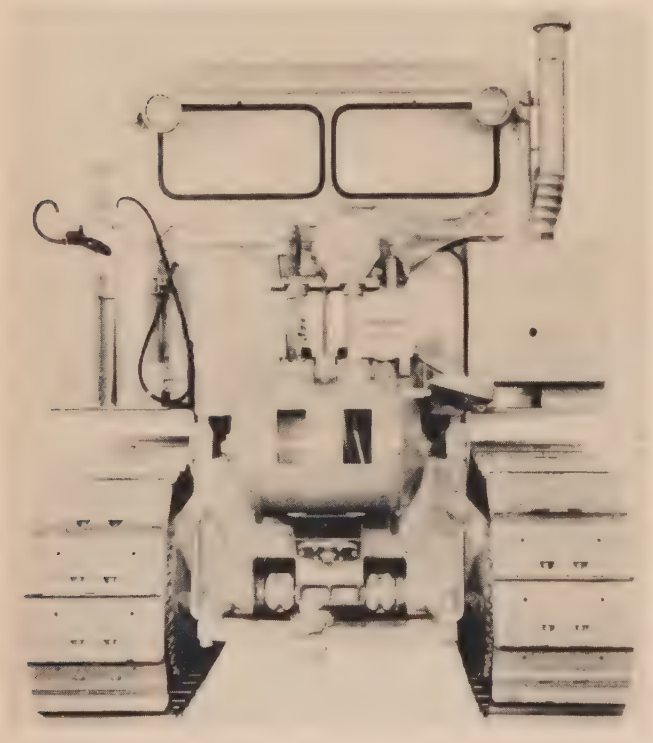

Fig. 28—Cab, Perfection heater, auxiliary Hauck burner, and supply tanks, and boot-covered cable controls on winterized D7 tractor

provides heat for the insulated battery box and for the cab.

3. The battery box.
4. The insulated Crenlo cab.
5. The gasoline supply tanks for the engine compartment Hauck 251 heater and the auxiliary portable Hauck heater.
6. The winterized fabric boot covering the cable controls.

Other features not in evidence in these figures include:

1. A slave receptacle to permit the use of an external electrical power supply.
2. The readily accessible controls and valves for ease of handling by operators clothed in bulky equipment.
3. The aluminum sliding plate winterfront controllable from the cab seat.
4. The engine compartment Hauck heater.
5. The ether priming system.

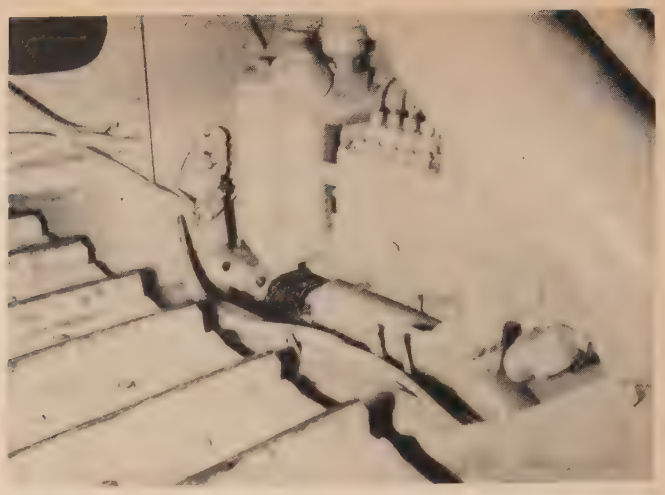

Fig. 29—Hauck engine compartment heater in winterized D7 tractor

Basic Cranking and Priming System

The conventional D7 2-cyl vertical starting engine was equipped with a 24-v electric starting system. The electrical system energy is supplied by four series-connected 4H batteries. The diesel is cranked through the standard 2-speed transmission and starter pinion. With this arrangement, starts to −25 F can be made without diesel or starting engine priming aids. However, the starting engine was equipped with one and the diesel engine with three ether applicators mounted to permit the rapid successive firing of Chevron pressurized capsules for starts at temperatures as low as −65 F or for quicker starts at −25 F or above.

Powerplant Heating System

A gravity-furnace-type engine compartment heating arrangement using a Hauck 251, 265,000-Btu per hr gasoline burner provided heat for the engine enclosure. (See Figs. 29 and 30.) Past experience has proved these heaters to be reliable. This burner was mounted immediately in front of the cab and beneath the oil filters on the right side of the engine compartment. The fuel for the burner comes from a hand pump pressurized gasoline storage tank of approximately 4-gal capacity. This provides sufficient fuel for several starts at −65 F.

An auxiliary Hauck 251 burner and storage tank mounted on the left rear fender provided a portable heating source. It was assumed that this extra heat source would be useful in removing heavy ice and packed snow from vital tractor components.

Operation of the burner for the engine compartment is begun by preheating vaporizing coils with an open pan of gasoline provided with the burner. A

deflector plate pivots in front of the fuel nozzle to fill the preheat pan. A minimum of 2-min preheating was required to generate a blue flame. This hot flame then discharges into a stainless steel plenum chamber mounted beneath the diesel engine. A gap between the end of the burner and the entrance to the plenum allows heat to warm the oil filters and the oil filter base. Heat is discharged from the plenum through one large opening to the external surface of the diesel-engine crankcase and through two smaller flexible ducts to the starting engine oil pan and cylinder head. A third small duct carries heat to the front-mounted hydraulic power unit although the need for heat at this point is questionable.

The diesel and starting engine breathe from the heated enclosure surrounding the engine. A simple changeover piping arrangement allows the diesel engine to draw cooler ambient air in through the air cleaner during more temperate climatic operations.

Cab Design and Crew Compartments Heating System

A standard-production Crenlo cab as shown in Fig. 31 with modifications to meet the requirements for winterization was used on the D7. The cab was mounted on ½ in. thick steel fenders for stability. The attention always given to such things as visibility, headroom, and rigidity on standard cabs became an inherent part of the winterized cab. The winterization modifications included the insulating of the side and back panels, doors, and air cleaner with ¼ in. thick Ensolite plus a floor matting of harder Ensolite ½ in. thick. The mats were not cemented to the floor plates, thereby permitting easy access to adjustment points beneath the plates. The possibility of these loose mats shifting was considered, and shifting can be prevented should the test program prove it to be necessary.

The cab roof was acoustically and thermally insulated with loose fiber glass held in place by a thin perforated steel plate.

The rubber used in all hoses, heat ducts, and the mountings for the tinted safety glass windows met the requirements for −65 F performance as outlined in the ASTM D-736-54T specification.

A quick-exit escape hatch in the cab roof was held in position by over center spring-screen-door-type clips. A cam action release mechanism controlled by an escape bar provided the mechanical advantage needed to break accumulations of ice and snow loose from the hatch.

Cab heating, defrosting, and battery heating was accomplished through use of an E-500 Perfection fresh air heater. This heater has a rated heat output on high fire of 30,000 Btu per hr and an output on low fire of 15,000 Btu per hr. The heater air intake was covered to minimize the problem of snow infiltra- and was extended above the cab to assure a clean air supply. One duct from the heater supplies heat to the cab while four smaller ducts channel heated air to the various windows for defrosting. Heat flow through through the defrosting ducts was controlled by manually operated butterfly valves.

A butterfly valve in the heat duct to the insulated battery box, controlled through a capillary tube and

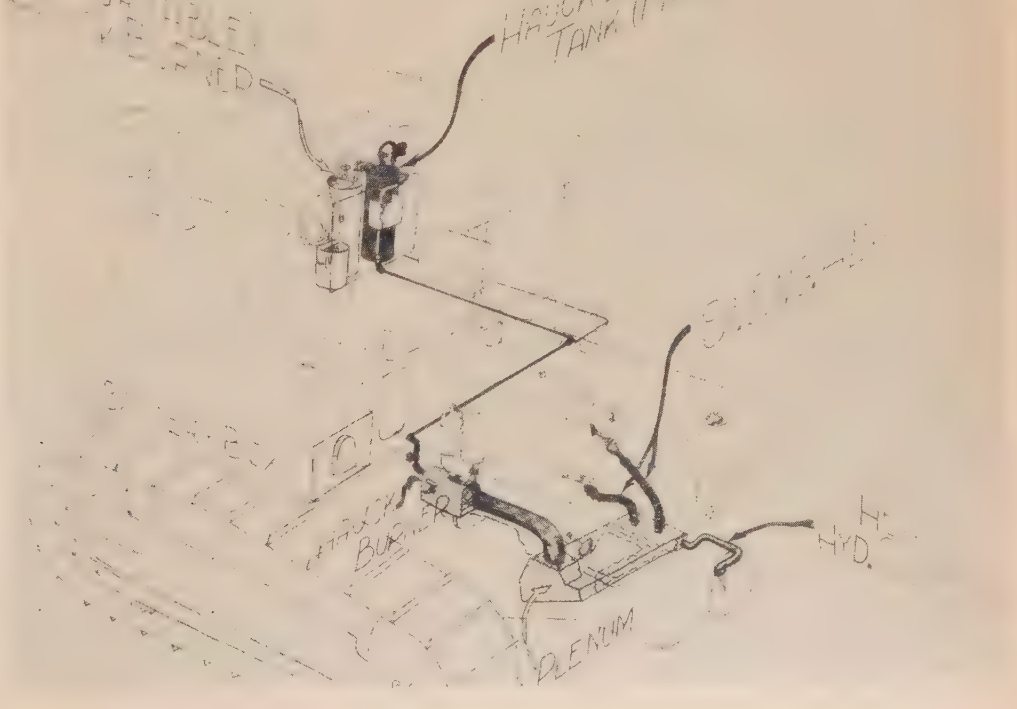

Fig. 30—Schematic drawing of engine compartment heating system and location of auxiliary portable Hauck burner

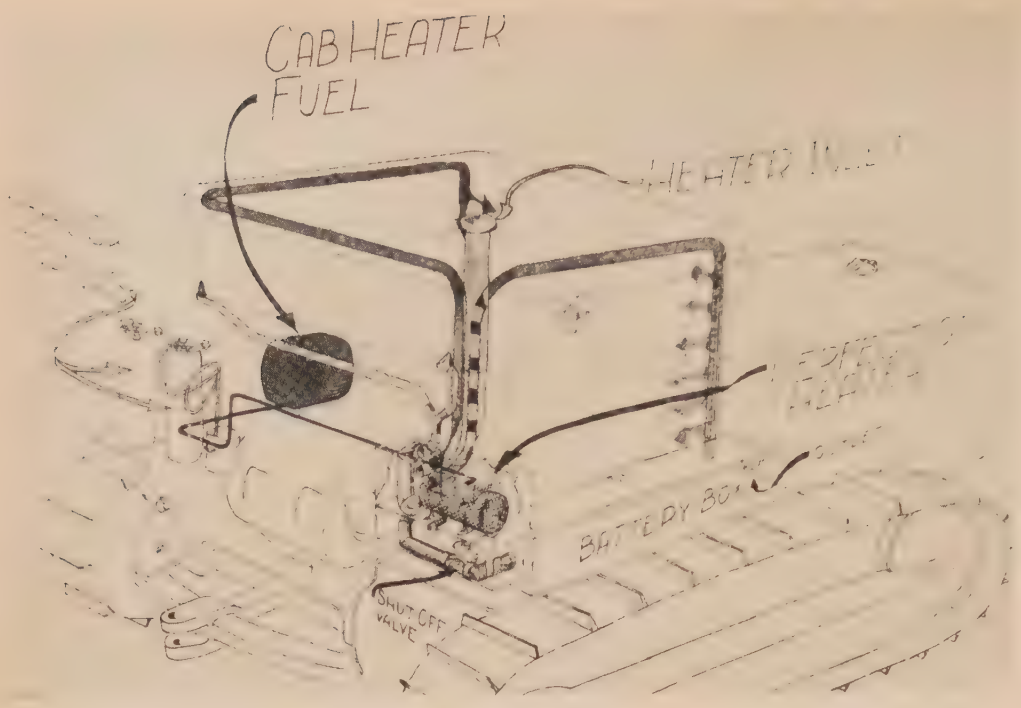

Fig. 31—Crew compartment and battery-box heating system

sensing element placed near the center of the battery box compartment, maintains temperatures in the battery box between the approximate limits of +35 and +100 F. Heat enters at the rear of the box and is discharged to the atmosphere at the front. The four 4H batteries lay end to end within the box along the tractor fender. A quick-release latch mechanism permits rapid battery servicing.

Electrical System

The gasoline-starting engine was equipped with a 24-v cranking motor which provided unaided starting at —25 F. The lights, windshield wiper, horn, Perfection heater, and other components were connected to the battery and returned through a common ground. A shielded 24-v–40-amp charging generator and regulator as well as shielded starting engine magneto, spark plugs, and cables were used in accordance with military specifications.

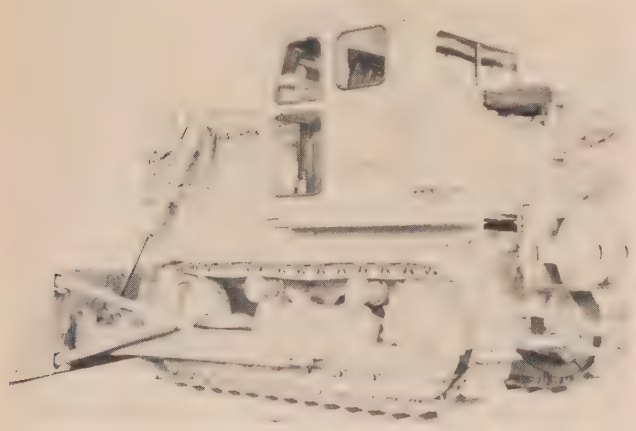

Fig. 32—General description of Allis-Chalmers HD-16A winterized crawler tractor.
Basic Tractor Weight: 31,500 lb
Gross Weight (Tractor-Equipment-Winterization): 41,500 lb
Drawbar Horsepower: 125
Engine: 844-cu in. Displacement
Naturally Aspirated, 4-Cycle, Diesel

Description of Winterization System on Allis-Chalmers HD-16A Crawler Tractor

by

W. W. Cornman

Allis-Chalmers Mfg. Co.

General Description

The Allis-Chalmers HD-16A crawler tractor, having specifications as illustrated in Fig. 32, was winterized for the Corps of Engineers to operate in the Arctic at ambient air temperatures down to —65 F.

Priming Aids

Preliminary tests indicated that approximately 70-cc ether would be required to insure reliable engine starts at —65 F ambient temperatures.

Cold-room tests also proved that it was almost impossible for the operator to reload an ether applicator during the starting and warmup period. Bulky arctic clothing and rapid engine deceleration handicap the operator to the point where the engine will usually die before the applicator can be reloaded.

On the basis of the above preliminary tests an Ampco-Sinclair bulk ether applicator as shown in Fig. 33 was selected for experimental use on this vehicle.

This primer provides 100-cc pressurized ether at the time of starting. There is also an additional 190 cc of ether available, if required, after a start is made. No reloading is necessary during the starting or warmup period. The operator can control the flow of ether to the engine by merely opening or closing a toggle valve.

Sinclair Military Starting Fluid was recommended for use in this applicator. This fluid consists of an ether base, with certain additives present which reduce engine detonation if there is an excessive use of fluid.

Cab Design

The cab used on this vehicle was designed and built by the Allen Industrial Products Co. of Battle Creek Mich. (See Fig. 34.)

A standard tractor cab design was modified incorporating a 2-man escape hatch in the roof, permalastic insulation throughout, and a defroster system capable of defrosting all windows (front, side, and rear). Simple, manually operated flap valves were provided in all ducts so the defrosting system could be balanced to insure good visability and operator comfort.

Powerplant Heater

A Perfection E-510 fresh air heater (Fig. 35) was installed on this vehicle. This heater has a fresh air output containing 60,000 Btu maximum. There is also an additional 30,000 Btu in the heater exhaust. This heater was designed into the vehicle so that it would serve both as a powerplant heater and a crew compartment heating system.

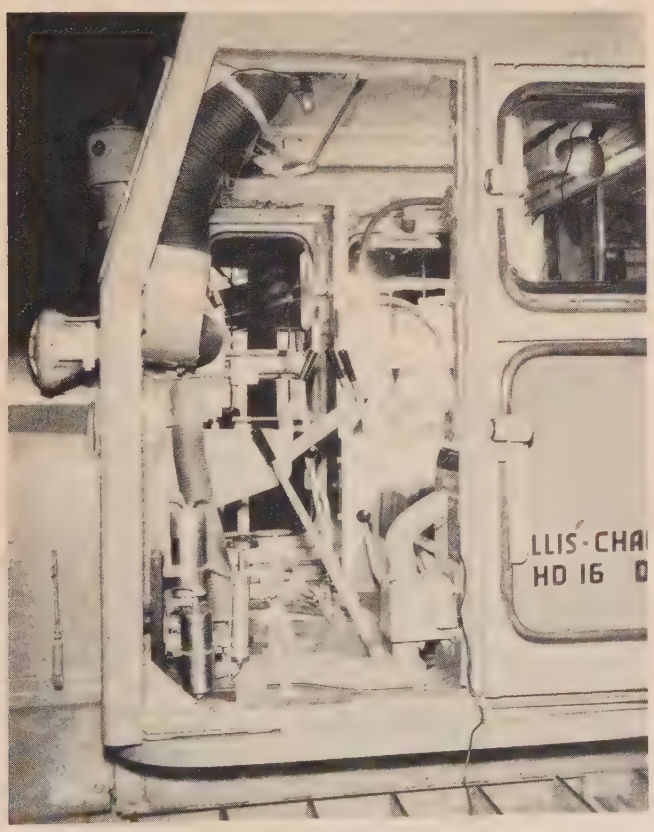

Fig. 33—Ampco-Sinclair bulk ether applicator

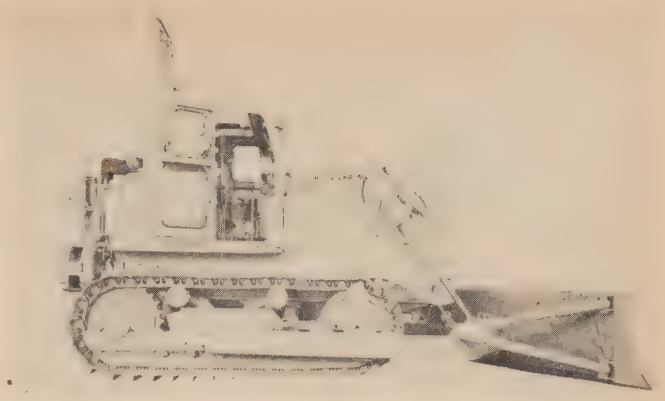

Fig. 34—View of winterized cab

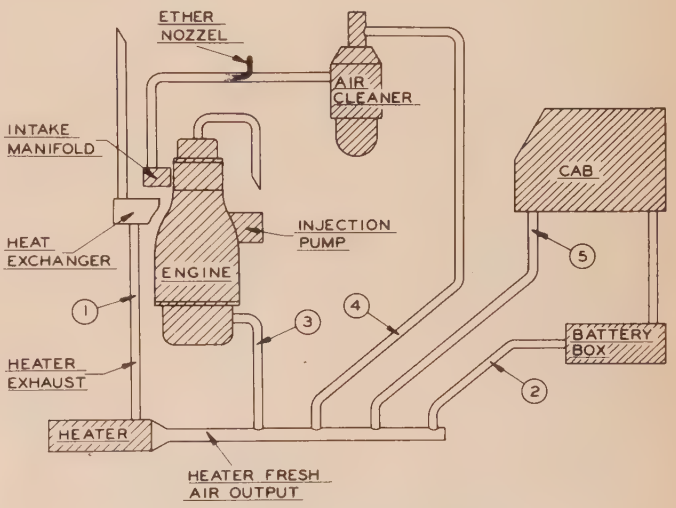

Fig. 35—Powerplant heating circuit

The operation of this heating system shown in Fig. 35 is as follows:

Duct (1) carries contaminated heater exhaust to a heat exchanger mounted under the engine intake manifold. This improves engine efficiency and assists ether atomization.

Duct (2) carries 50% of the heater fresh air output to the insulated battery box. This improves battery efficiency. A thermostatically operated flap valve was installed in this line to avoid battery overheating.

The operator can direct the remaining 50% of the heater fresh air output into ducts (3), (4), or (5) at his option, by manipulating flap valves located in the cab.

Duct (3) internally heats the engine during the

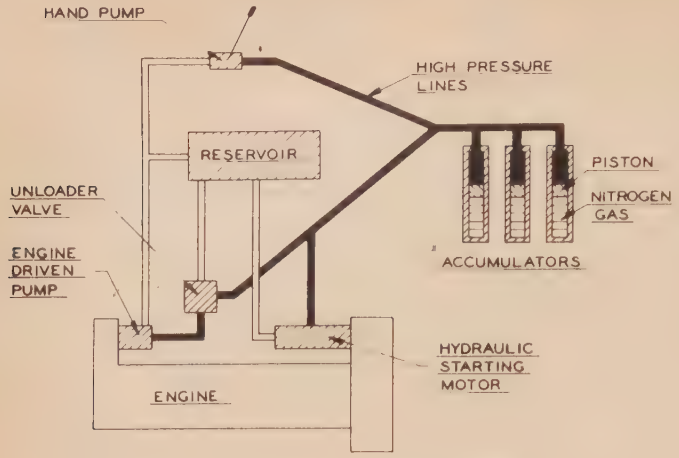

Fig. 36—Hydraulic cranking system

Table 5—Insulating Materials

Material	Location
Permalastic products PL-113 spray viscosity mastic type	Cab walls Engine compartment walls
Silicon rubber strip minimum temperature range—65 to +125 F approximately 1/16 thick	Cracks around cab, doors, escape hatch, fuel tank, and cowl
Felt—SAE F-13 1/4 thick MIL-M-2312 type III rubber coated—1/32 to 1/16 thick-MIL-R-3065—class SC-512 ABFF	Cracks around engine compartment Cab floor mat
Aeromotive Specialities Co.	
Johns-Manville Marinite—36 board type	Battery box top and sides 1/2 thick bottom 1 1/2 thick

warmup period prior to starting.

Duct (4) heats the engine air intake system during starting.

Duct (5) heats the cab and defrosts the cab windows after the engine is running.

Basic Cranking System

As an experimental part of the winterization program a hydraulic cranking system was installed on this tractor. This unit was supplied by Hydramotive Inc. of Cleveland, Ohio.

This installation is outlined in Fig. 36. It should be noted that an engine-driven pump was installed to recharge the accumulators automatically. The hand recharging pump was intended only for emergencies.

This system has the following advantages:

1. If the operator exhausts the cranking system he can manually recharge it with the hand pump.
2. This unit provides a high initial torque for turning the tractor engine over.
3. This unit provides a high initial cranking speed. This is a big assist in cold weather starting.

Electrical System

The electrical system on this machine was a conventional tractor military installation except for the batteries. Sintered-plate, nickel-cadmium batteries, (Part No. 10–60M–220), supplied by the Sonotone Corp. of Elmsford, N.Y., were used on this unit to operate the engine heater and lights.

It was felt that an evaluation of these batteries under cold-room and field testing would be of interest.

An electric starting motor was retained on this tractor in addition to the hydraulic starter. This was installed to provide a use for the electrical slave kit receptacle which is a standard item on every military vehicle. This arrangement provides an alternate electric starting system based upon any available outside source of electrical power, such as a generator set.

Special Materials (Insulation, Winterfronts, and Rubber Parts)

Table 5 gives a list of insulating materials, used on this tractor.

The winterfront used on this tractor was originally designed by the Corps of Engineers at their Research & Development Laboratories, Fort Belvoir, Va. It consists of one fixed and one sliding plate with matching holes. The plates are heavy gage steel or aluminum and will shear ¼ in. of ice without distortion or

Fig. 37—View of Austin-Western 99L at test site in Minnesota

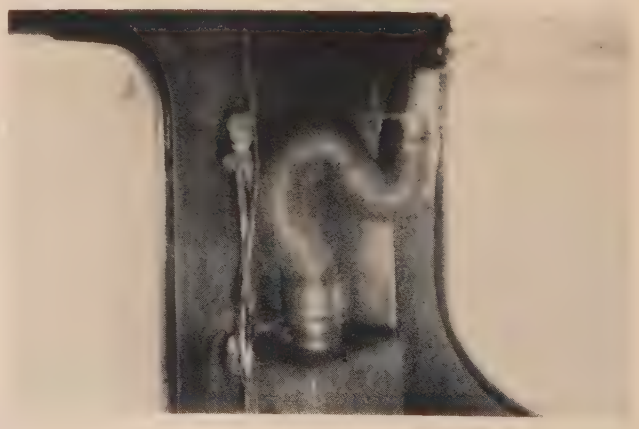

Fig. 38—View of Austin-Western 99L cab interior. Note cemented sheet insulation, thermostatic control for battery-box heat, and flexible tubing used for hot air duct

binding. A manual control for this shutter was provided for remote control in the operator's compartment.

The provision of suitable rubber parts for this tractor proved to be a problem. Rubber companies today do not normally have −65 F rubber parts in stock. They can be procured, but must be specially ordered. It is exceedingly important that the proper rubber be used. An "O" ring suitable for −30 F can become as hard as glass at −65 F and can deadline the complete vehicle.

Test Results on Winterized Construction Equipment

by

Alan Q. Spitler
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Results of Cold Chamber Tests

Test Results—Austin Western 99L Motor Grader (See Figs. 37 and 38.)

1. *Starting Test at −25 F*—This test was conducted at Cleveland, Ohio, in the cold chamber of Perfection Industries, Inc. The engine was primed with ether and after 10 sec of cranking started but did not continue to run. Approximately 1–2 sec of cranking were required to restart it. Each time the engine started to "die," it was primed with a stroke on the ether primer. It was necessary to continue priming for approximately 4 min before the engine would run on diesel fuel alone.

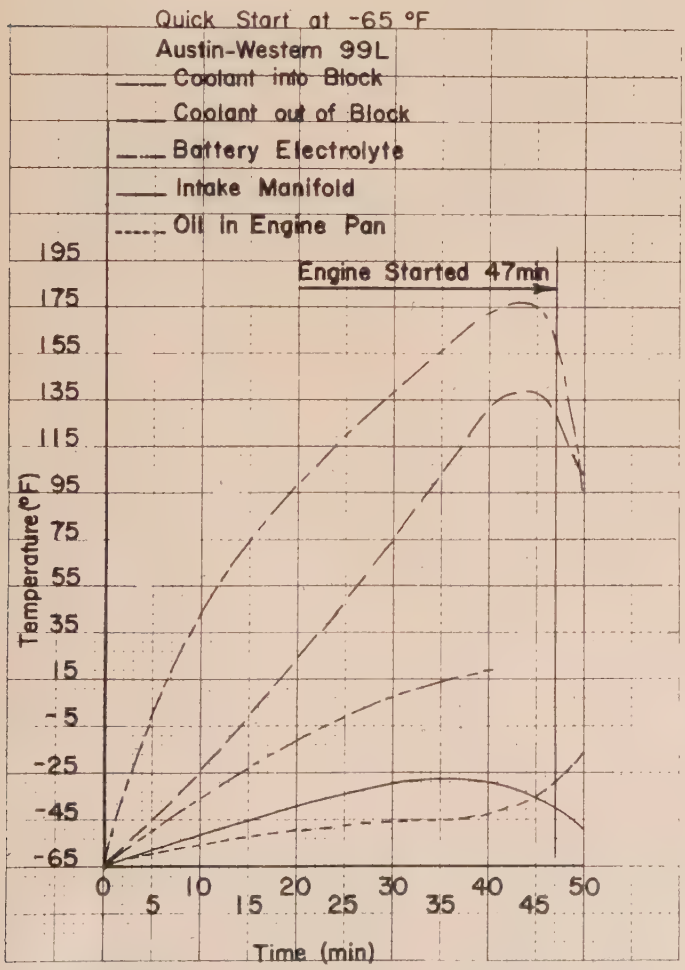

Fig. 39—Engine warmup curves of Austin-Western 99L

Fig. 40—View of Caterpillar D7 at test site in Minnesota

2. *Starting Test at −65 F*—The same test facility, as above, was used for this test. The doors on the cab were left standing open during the pull-down from −25 to −65 F to allow the batteries (located inside the cab) to "soak." At −65 F the doors could not be closed because the weather stripping was frozen solid. The doors were closed partially by thawing the weather stripping with an acetylene torch. With the doors standing open approximately 2 in. the air temperature in the cab was raised to +40 F in 30 min.

An attempt was made to start after 45 min of heating. During this attempt the engine started and continued to run after 47 min of preheat. Only 3 sec of cranking were required for this start. The ether primer was pumped three times (approximately 7½ cc of ether). Fig. 39 shows engine warmup curves drawn from data recorded during this test.

Test Results—Caterpillar D7 Tractor (See Figs. 40 and 41.)

1. *Starting Test at −25 F*—This test was conducted in the cold chamber at Caterpillar Tractor Co. The gasoline starting engine started after 30 sec of cranking with the electric starting motor. The gasoline engine was allowed to warm for approximately 2 min after which the clutch was engaged and the diesel engine cranked. The diesel engine started after approximately 5 min of cranking, making a total of 7-min elapsed time.

2. *Starting Test at −65 F*—The same test facility as above was used for this test.

The torch heater was started and allowed to heat

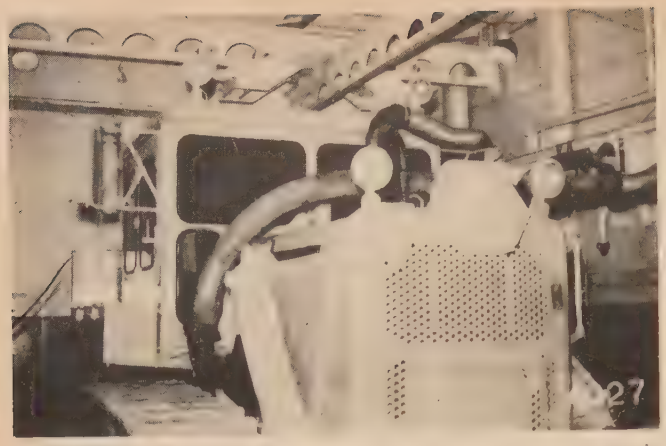

Fig. 41—Caterpillar D7 in cold chamber at Peoria, Ill.

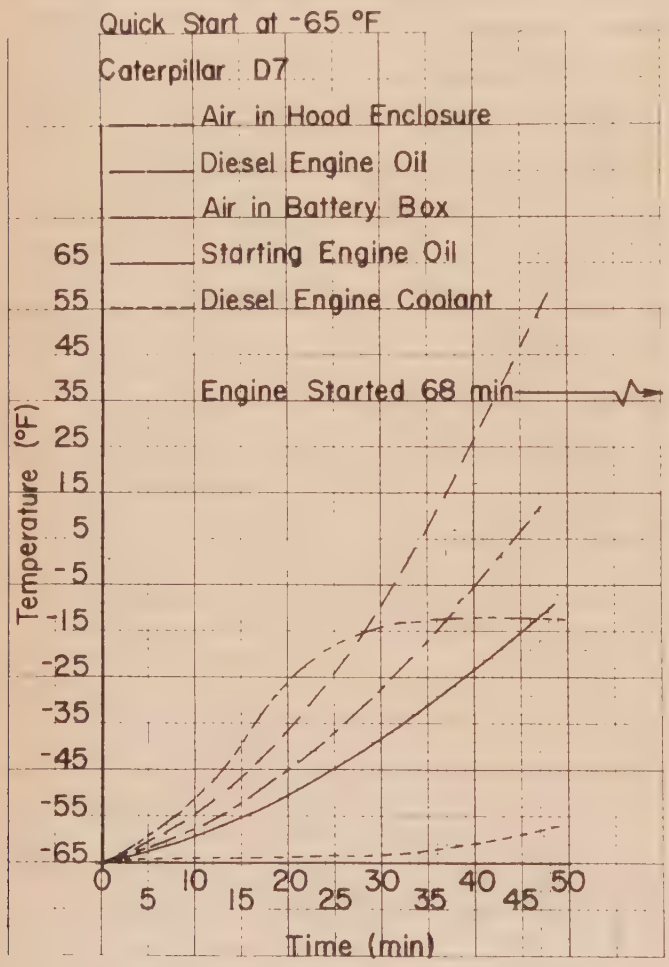

Fig. 42—Temperature curves for Caterpillar D7

the cranking engine and diesel engine. Heating took place for 47 min after which several attempts were made to start the cranking engine. Starting difficulty was determined to be due to a faulty ignition wire. The insulation was cracked, and the wire was grounded on the engine block. In addition, noxious gases collected in the operator's compartment when the torch heater was in operation.

With the cranking engine started, the first attempt to crank the diesel engine stalled the cranking engine.

Fig. 43—View of International-Harvester TD-18 at test site in Minnesota

Fig. 44—View of International-Harvester TD-18 cab interior. Note spray-type mastic insulation and sheet metal defroster ducts

The cranking engine was restarted and the diesel engine started after approximately 7 min of cranking and after firing three ether priming capsules. The diesel engine was running smoothly only 40 sec after starting. Total elapsed time for this start was approximately 1 hr. The temperature inside the cab was raised from −65 to +20 F in about 20-min heating time. Cab temperatures were not recorded after 20 min. Fig. 42 illustrates temperature versus time curves drawn from data recorded during this test. Fig. 41 shows the test setup in the cold chamber.

Test Results—International-Harvester TD-18A Tractor (See Figs. 43 and 44.)

1. *Starting Test at −25 F*—This test was conducted in the cold chamber at International-Harvester Co. No difficulties were encountered, and the engine was started on the gasoline cycle after 15 sec of cranking. No heat was applied prior to starting. The engine was operated on the gasoline cycle for 4 min before switching to diesel. The Chevron pressurized

ether primers were not used. The actual temperature during this test was −30 F.

2. *Starting Test at −65 F*—The engine heater was operated for 53 min prior to starting the engine. The engine was started on the gasoline cycle after cranking for 25 sec and operated for 3 min on the gasoline cycle. It was then switched to diesel operation 57 min after the test commenced. The Chevron pressurized ether primers were not used. Fig. 45 illustrates temperature rise of the engine components during the period of engine preheat.

During the −65 F test, difficulty was encountered in closing the cab doors, as the silicone rubber door seals would not compress at this temperature. The temperature within the cab was raised from −65 to +60 F in 40 min of heating. During the first 15 min the cab doors stood open approximately 2 in. Apparently the door seals softened during this period because after 15 min the doors could be closed.

Test Results—Allis-Chalmers HD-16A Tractor (See Figs. 46 and 47.)

1. *Starting Test at −25 F*—This test was conducted at Cleveland, Ohio, in the cold chamber of Perfection Industries, Inc. No difficulties were experienced, and the engine started in 7 sec of cranking with the hydraulic starter. Ether was used to start and keep the engine running until warm enough to run on diesel fuel alone. Ether was injected in small amounts each time the engine started to "die."

2. *Starting Test at −65 F*—The same test facility, as above, was used for this test. The engine heater was started and allowed to heat the engine for 50 min. The engine was primed with ether priming fluid and the hydraulic starter engaged. The engine started with only 2–3 sec of cranking. Total elapsed time for this start was 53 min. Approximately 45 cc of ether were used on this start. The cab temperature was raised from −65 to +27 F in about 25 min of cab heating. The cab temperature was raised to +43 F in 72 min of cab heating. Heat was not supplied to the cab until after 50 min of heating the engine. Fig. 48 illustrates warmup curves drawn from data recorded during this test. Fig. 44 shows the test setup in the cold chamber. This tractor was equipped with a separate electrical cranking system intended for use with slave power. To satisfy our own curiosity we attempted a start at −65 F. The nickel-cadmium batteries furnished to supply current for heaters and

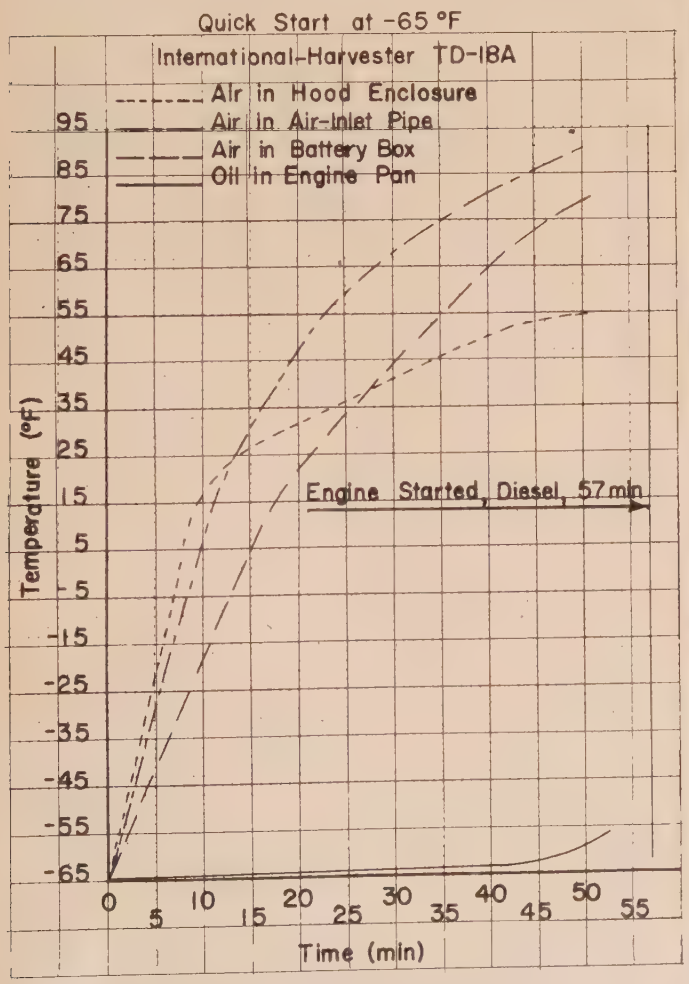

Fig. 45—Tempearture rise of engine components of International Harvester TD-18A

Fig. 46—View of Allis-Chalmers HD-16A at test site in Minnesota

Fig. 47—View of Allis-Chalmers HD-16A in cold chamber at Cleveland, Ohio

Table 6—Average Temperature Rise, F (10-min Heating)

	AW-99L	AC-HD-16A	Cat-D7	IHC-TD-18
Head, Coolant			0	12
Block, Coolant		0		15
Oil Pan	29	0	5	12
Air, Oil Pan		4		
Intake Air	20	30		74
Cab Heater Outlet	142		144	200
Battery, Air		86		
Coolant Out, Block	20			
Coolant In, Block	100			
Ambient	−12	−12	−12	−12

This table is for one week operation during which ambient temperature averaged −12 F. The various points at which thermocouple readings were taken do not apply to each machine; therefore, some of the data cannot be filled in.

Table 7—Engine Oil Consumption for Arctic Engine Oil (MIL-O-10295)

Week (Average Temperature)	AW-99L	AC HD-16	Cat D7	IHC TD-18A
Jan 16-20 (+2 F)	5 qt used		0 qt used	4 qt used
Hours of Operation	36 approximately	0	23	36 approximately
Jan 23-27 (+17 F)	2 qt used	0 qt used	0 qt used	4 qt used
Hours of Operation	74 approximately	38	63	74 approximately
Jan 30 Feb 3 (−12 F)	0 qt used	0 qt used	2 qt used	5 qt used
Hours of Operation	110 approximately	68	92	110 approximately
Feb 6-10 (+17 F)	4 qt used	8 qt used	4 qt used	4 qt used
Hours of Operation	148 approximately	105	130	148 approximately
Total Quarts Used	11	8	6	17
Total Hours	148 approximately	105	130	148 approximately
Average Quarts per Hour	0.0744	0.0762	0.0462	0.115

Table 8—Mechanical Deficiencies of Austin-Western 99L Grader

Deficiency	Probable Cause	Corrective Action
Heater igniter failed	Fluctuating voltage[a]	New igniter installed
Heater igniter failed	Fuel vapor lock	New igniter installed
Heater fuel tank did not have a breather hole	Unknown	Hole drilled in fuel tank cap
Lid on escape hatch fell off and was bent	Faulty latches	Lid was straightened and replaced
Heater igniter failed	Fluctuating voltage	New igniter installed
Heater failed to light	Fuel wick was charred and burned through probably due to insufficient fuel	Wick was twisted around until an unburned section was opposite the igniter
Heater igniter failed	Fluctuating voltage	New igniter installed
Broken tilt adjustment link	Unknown	Tilt adjustment link was welded

[a] Igniters could not withstand normal voltage fluctuation, characteristic of standard tractor electrical components. Moldboard cutting edge was worn down and was turned over at approximately 200-hr operation.

lights have insufficient voltage capacity for cranking at −65 F. The effective voltage drop-off was very severe. Starting with the batteries was not successful, but slave power (24 v) started the engine with 5 sec of cranking. However, it was observed that capacity of the batteries was satisfactory for their intended operation of lights and heaters.

The "O" ring seals delivered with the hydraulic accumulators were not the correct seals for use at −65 F and had to be replaced. Purchasers of hydraulic accumulators must make certain that the "O" rings are good for use at −65 F.

Field Test Results

Results of tests last winter at the Erie Mining Co. on the Mesabi Iron Range were very gratifying. The lowest area temperatures encountered were −20 to −25 F, and the average low daily temperature was approximately 0 F. No major difficulties were encountered in starting any of the equipment even after cold soaking for as long as 48 hr. Minor difficulties, which prevented starting or increased starting time were not attributed to failure of the winterization to meet minimum requirements. Figs. 37, 40, 43, and 46 show the machines at the test site in Minnesota.

Table 9—Mechanical Deficiencies for International-Harvester TD-18A Tractor

Deficiency	Probable Cause	Corrective Action
Battery cable connector broken	Vibration, no slack in cable	New connector soldered on and cable rerouted
Frozen fuel line	Water in diesel fuel	Line was disconnected and thawed out with a blowtorch
Engine failed to start on gasoline	Distributor rotor was loose	Loose screw was replaced and rotor tightened
Heater igniter failed	Fluctuating voltage	New igniter installed

Bulldozer cutting edge turned over and end bits replaced after approximately 150-hr operation.

With few exceptions, the winterized units started with approximately 3-sec cranking time after a 10- to 15-min heating period. The engines were started on many occasions without any heating because our military specification precludes the use of heat at temperatures above −25 F. The heaters were started and run a short time each morning to log a maximum number of starting cycles.

On several occasions this equipment was used to tow unwinterized units at the Iron Range which were not capable of self-starting. Starting unwinterized units required considerable time at the temperature extremes encountered at the Iron Range.

In general, the completely enclosed and insulated cabs were too warm at the temperatures encountered. Operators wore heavy clothing, such as wool shirts and jackets, although the normal clothing was not nearly so bulky as that required for an American soldier in the Arctic. Therefore, cab heaters were not used in the tractors except on the very coldest days, and heat radiated from the engine and transmission was normally sufficient. The cab heater in the grader was used nearly every day because no appreciable amount of heat was received from the engine and transmission. Defrosting was considered adequate on all of these machines.

Temperature recordings were made each morning for several weeks during warmup. Thermocouples were placed at various strategic points in the engines, and readings were taken with a potentiometer.

The heaters were started each morning for 20 working days and run for about 10–15 min to warm the engines and operator's cabs before starting the engines. Thermocouple readings were taken, but the short warmup time did not show an appreciable temperature rise except in the installation using a coolant heater. The fresh air heaters raised the engine intake air temperature, which was a definite aid to starting. This feature aided in vaporizing the priming fluid and in heating the engine interior. Recording of temperatures, except ambient temperature, was suspended

after the first 20 working days. Subsequently, the machines were worked an additional 20 days, 24-hr a day, five days a week. Equipment was shut down on Saturday and Sunday. Table 6 indicates the average temperature rise of various components.

The most frequent trouble with the heaters was igniter failure, which was attributed to several causes which have to be remedied as follows:

1. Positioning of heater control panel should be in front of the operator so he can readily observe the warning light. This light will go off if for any reason the heater stops, but if the trouble is not corrected the heater may cycle on and off causing possible igniter failures.

2. Condensate in heater fuel lines forms ice to block the fuel supply to the heater. This would cause repeated heater starting cycles and prolonged ignition to aggravate the igniter problem.

3. Operators fail to keep heater fuel supply tanks full. An aid in this direction would be to install visible fuel gages in the cab. When the heater is allowed to run out of fuel the on-off cycle will start and eventually lead to trouble.

Excessive window breakage in all cabs was noted. This was attributed to improper design mounting and location of defroster ducts. Investigations of ERDL have shown that this type of window breakage usually follows when hot defrosting air is concentrated narrowly at one side of the window. It was also noted that all of the cabs on test had defrosting of this type. Rubber seal material must be flexible at low temperatures to allow for expansion and contraction.

Military-issue arctic lubricants and greases recommended by the SAE CIMTC Subcommittee XV were used exclusively in all of the machines. Arctic diesel fuel was used for several weeks until the supply was exhausted, after which Erie Mining Co. provided No. 1 diesel fuel (Pour point −40 F). Oil consumption was not excessive. Table 7 gives engine oil consumption.

Mechanical deficiencies were not serious; however, any repairs which must be made in the field were aggravated by unpleasant weather. It was extremely difficult to make repairs at subzero temperatures without shelter. Tables 8, 9, 10, and 11 contain information recorded in the field on mechanical deficiencies, probable causes, and comments for each of the winterized items.

Rough operating conditions were the general rule on the Mesabi Iron Range. Table 12 gives the climatic conditions. The temperature was consistently low, the ground remained frozen, and the overburden was very hard. Considerable wear on tracks, blades, and tires was noted on all machines. Operators were not selected for proficiency; therefore, some were good, and some were poor. The operators were not closely supervised because it was desired to operate the equipment under normal conditions.

Table 10—Mechanical Deficiencies of Caterpillar D7 Tractor

Deficiency	Probable Cause	Corrective Action
Heater thermal relay failed	Unknown	New thermal relay installed
Bracket for spare torch heater was broken	Vibration	Bracket was welded
Left door latch inoperative	Broken spring	New spring installed
Heater igniter failed	Fluctuating voltage	New igniter installed
Heater fuel supply line broken	Vibration	New tubing installed
Heater igniter failed	Fluctuating voltage	New igniter installed
Escape hatch lid fell off and was bent	Faulty latches	Lid was straightened and replaced
Heater igniter failed	Fluctuating voltage	New igniter installed
Torch heater fuel tank will not hold pressure	Faulty cap	Cap from spare tank was exchanged

Bulldozer cutting edge turned over and end bits replaced after approximately 150-hr operation.

General Observations

1. Operators were pleased with these units and in general enjoy working a unit when they were kept warm and protected from the elements.

2. Most of the maintenance difficulties can be solved by relocating components of the winterization system.

3. The removable side panels of certain units

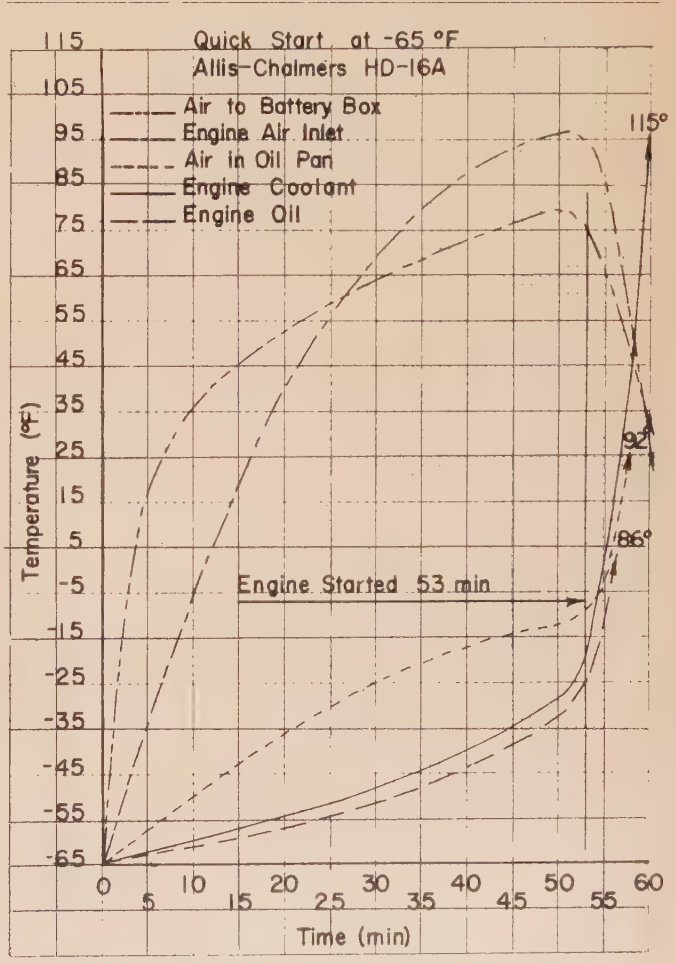

Fig. 48—Warmup curves of Allis-Chalmers HD-16A

should be larger for easier engine access and made to fit better.

4. In some installations winterization components obstructed vision seriously.

5. The slave receptacle, for utilization of outside cranking power, on one unit could not be used because it was so located that removal of the cap was impossible.

6. In every installation protective covers built over the heaters were difficult to remove for maintenance.

7. Latches and handles in most installations were too small for easy operation with mittened hands.

8. Covers over heater inlet air openings, to prevent ingress of snow, were not satisfactory on most of the units.

9. Grease fittings on some units were not easily accessible.

10. The escape hatches were inclined to come loose during operations on two of the units.

11. Engine enclosures on some of the units were not tight enough to keep out snow or keep in heat.

12. The radiator shutter on one of the units was not considered satisfactory because it was too difficult to operate.

13. The arctic gasoline was prone to vapor lock on warm days.

Table 11—Mechanical Deficiencies for Allis-Chalmers HD-16A Tractor

Deficiency	Probable Cause	Corrective Action
Broken fuel line	Vibration	New tubing installed
Fuel gage inoperative	Broken fuel gage line	Line was repaired
Broken fuel line	Vibration	Flexible tubing installed
Flexible pipe from heater to manifold was broken	Vibration	Pipe was welded
Fuel oil return line was broken	Vibration	New tubing installed
Heater fuel supply tank cracked and started leaking	Vibration	Crack was welded
Main electrical cable to heater burned off	Vibration which rubbed through the insulation	Electrical cable was repaired and taped
Heater igniter failed	Unknown	New igniter installed
Heater fuel supply tank cracked and started leaking	Vibration	Tank was removed and all joints were rewelded
Hydraulic starter inoperative	Pinion shaft twisted, thrust bearing cracked	New hydraulic motor installed
Motor driven hydraulic pump leaking	Misalignment caused bearings and front seal to wear	New pump installed
Heater igniter failed	Unknown	New igniter installed
Heater flame switch failed	Unknown	New flame switch installed

Bulldozer cutting edge turned over and end bits replaced after approximately 150-hr operation.

Conclusions and Recommendations from Description and Tests of Winterized Construction Equipment

by

P. W. Espenschade

Engineer Research & Development Laboratories

Program Concepts (Industry Liaison)

THE story about this winterized equipment for the Army Corps of Engineers really dramatizes the importance of coordinating promotional (publicity and planning) as well as technical (design and production) considerations. The detailed record of accomplishments also confirms what can be done with literally a chilling prospect, that is, assuring for subzero operations a variety of diesel-powered items of commercial origin, when responsible suppliers are honestly read-in on the tasks and patiently encouraged to meet the challenge head on. In the final analysis, events reflect gratifying progress in the persistent effort by all concerned to avoid the dogmatic approach often attributed to dealings between the military and industry. In this respect the SAE CIMTC Subcommittee XV has been playing a major role and setting a high standard for "missionary" work.

The present state of affairs did not materialize without a struggle against a prevalent, although understandable, inertia and ignorance factor. The situation appeared particularly dark when the topic was originally introduced to the 1950 SAE National Tractor Meeting [1] under the shadow of the developing Korean conflict. This briefing outlined the fundamental problems to overcome and forecast general areas of attack for purposes of the Corps of Engineers. This was subsequently augmented by a more comprehensive analysis of the complex arctic operations directly affecting materiel and personnel.[2] The basis for subsequent winterization provisions in engineer end-item specifications was analysed at that time in line with the official Army policies concurrently established [3] and doctrine outlined for ERDL.[4]

In the meantime, significant actions were effected in related matters. First, a Mesabi Iron Range reconnaissance by engineer and industry personnel was completed in June, 1951. This constituted an evaluation of severe cold weather operations (for this country) of heavy automotive mine equipment in conjunction with a consideration of ways and means to conduct useful shake-down tests of comparable military versions of such items under similar rugged conditions. Next, official approval was given in September, 1951, to organization of the aforementioned SAE Subcommittee to study winterization. Finally, a combined United States-Canadian Engineer User Trial Exercise (Eager Beaver I) was successfully launched in the Yukon during the winter of 1951–1952, to practice on modified stock-issue items (for an engineer construction battalion) the requirements for design integration and original fabrication under

[1] "Winterization of Construction Equipment," by R. W. Beal. Presented at SAE National Tractor Meeting, Milwaukee, Sept. 12, 1950.

[2] SAE *Quarterly* Transactions, Vol. 6, October, 1952, pp. 689–709: "Subzero Winterization of Diesel-Engine Power Equipment," by P. W. Espenschade, R. C. Navarin, and W. W. Van Ness.

[3] SR 705–70–5, Department of Army, December, 1950, "Research and Development of Materiel: Operation and Protection of Materiel under Adverse Conditions of Temperature."

[4] ERDL SOP 700–3, November, 1952, "Winterization of Corps of Engineers Troop Equipment."

anticipated procurement specifications. The influence of these results has been duly publicized.[5]

Implementation in new or original construction equipment from industry was delayed however as procurement action focused primarily on winterized generator sets involving industrial (mostly gasoline) engines. Stationary powerplant winterization techniques and production experiences therefore developed ahead of the associated requirements and know-how for vehicular cab fabrication and crew comfort, and large diesel-powered items generally. The SAE Subcommittee's two progress reports[6] eventually noted and weighed these circumstances in offering the final recommendations which led to the current program. Considerable freedom was granted the respective end-item contractors accordingly in negotiating for an overall winterization system performance objective at this time rather than arbitrary requirements component-wise. Accordingly, the systems and hardware employed indicate different approaches and degrees of emphasis by various manufacturers. The combinations justify subsequent evaluation in proper prospective considering:

1. The state of the art at this time.
2. The devices comprising these systems are not intended necessarily as the final answer.
3. The end-item manufacturers are not committed for their commercial lines by the various options selected for purposes of the military contracts pertaining.

Overall Performance Factors

Generally, no inherent mechanical difficulty was encountered in starting any of the winterized machines during the reported field tests to date, confirming preliminary trial runs in the cold room. With ambient temperatures down to —25 F, engine starts were consistently made after 3–7 sec of cranking. Engine heaters, when employed, were operated 10–15 min prior to starting to obtain a routine performance record of individual characteristics (ignition and control) rather than optimum preheating throughout the systems. Under the circumstances, ease of relatively cold starting may be attributed mostly to the arctic fuels and lubricants employed, plus priming to some degree. No single contributor, however, stands out from the data at hand.

Generally, there was too much cab heat for operator comfort under the relatively favorable conditions prevailing, and specific interference with operator visibility by such things as the associated winterization heating system ducts was noted. Subsequent utilization of escape hatches for ventilation led to malfunctions and disclosures of mechanical deficiencies in those devices. Excessive window breakage also developed from the application of too much localized heat, combined possibly with vibration and shock not readily ascertained.

Table 12—Climatic Conditions (Erie Commercial Taconite Plant)

Date		Air Temperature, F (8:00 a.m.)	Wind Velocity, mph	Temperatures, F Maximum	Temperatures, F Minimum
January	9	17	3	20	15
	10	17	15	26	12
	11	4	5	16	4
	12	10	6	23	6
	13	11	5	27	11
	16	8	14	32	8
	17	8	8	9	6
	18	18	5	24	8
	19	12	2	24	12
	20	−16	8	15	−20
	23	−18	4	16	−18
	24	9	8	16	5
	25	18	7	23	13
	26	16	5	24	14
	27	20	10	22	16
	30	3	9	11	3
	31	−4	8	10	−4
February	1	−4	0	10	−6
	2	−6	6	10	−6
	3	−2	18	10	−6
	6	−24	5	5	−24
	7	15	6	30	15
	8	14	15	25	14
	9	20	17	25	14
	10	17	12	30	16
	13	9	20	27	9
	14	−8	10	30	−8
	15	−9	8	11	−14
	16	−14	13	22	−14
	17	−20	5	21	−20
	20	−2	6	15	−13
	21	−6	10	18	−10
	22	−5	12	15	−14
	23	−8	10	33	−8
	24	20	8	25	17
	27	−7	5	9	−12
	28	−6	0	12	−7
	29	1	6	16	−6
March	1	18	5	30	2
	2	14	10	36	13

Visual fuel gages were lacking, and heater control box pilot lights in some cases were poorly positioned, thus limiting an operator's knowledge of his situation and contributing to subsequent heater difficulties during long periods of running. Instrumentation with thermocouples to provide key temperature checks proved unreliable, and the anticipated record of events was not forthcoming.

Ease of maintenance and accessibility problems, especially with the heaters, some batteries, and various enclosures, left something to be desired. (See Appendix I.)

Concept implementation reflected occasional inexperience in the choice of some hardware, its arrangement, and quality control, rather than basic design conflict with the military philosophy, materials, components, and procedures now described for the purpose in the proposed Winterization Guidebook sponsored by ERDL and coordinated with the aforementioned SAE Subcommittee. (See Appendix II.)

Significant Features

Field results with priming aids were generally satisfactory but not conclusive because effectiveness of priming was usually influenced by preheating, directly or indirectly, of the diesel-engine block or intake manifold concerned according to the nature of the starting procedures actually employed. Thus, the increased intake air temperature favored the vaporization (and transportation) of any ether-type fluid drawn

[5] *Product Engineering*, Vol. 24, June and July, 1953: "Design Features in New Products" and "Technical News," by P. W. Espenschade.

[6] SAE CIMTC Subcommittee XV Winterization Progress Report No. 1, January, 1953, and Report No. 2, April, 1954.

into the respective systems. These circumstances, therefore, varied the conditions contributing to the priming problems previously reviewed for lower temperatures by the Pennsylvania State University.[7] From the operator's standpoint, location and control of some discharger arrangements challenged the ingenuity of an encumbered individual to execute the entire starting procedure by himself.

Theoretical possibilities of different powerplant heating systems were reflected in the laboratory by acceptable engine starting after 1 hr of preheating at −65 F. Priming and/or cranking devices employed assisted in the final results to some degree. Hazard to personnel (during manual ignition and from fumes) and material (due to flame propagation and soot) of the exposed torch heater was a penalty of that approach which counters its potency. Operation of other units was relatively safe and satisfactory, excluding cycling difficulties and igniter failures associated with other factors.

Size, arrangement, heat control, and individual mechanical features of the cabs came in for specific criticism. Ducting provided good hot air heat distribution generally. Location of ducting in some instances created visual obstructions. Complex systems of dampers and levers should be simplified on production vehicles. The prevalence of window crackage, escape hatch breakage, Ensolite floor mat and thermal insulation deterioration, and heater igniter failures indicate areas for improvement. Heater igniter failures were mostly experienced on personnel heaters operated when the end item was operated, that is, with engine running and relatively high voltage on the electrical circuit from the charging generator; or cycling developed from lack of fuel, shut down by such things as overheat or flame switch controls to aggravate the igniter situation through extra long periods of energizing.

The hydraulic cranking system indicated desirable possibilities in all operations as far as basic hydraulic motor and hydraulic accumulators were concerned. Difficulty developed through malfunctioning of the cranking motor engaging device which in turn was the fault of the linkage employed. Other cranking systems at prevailing temperatures operated in a conventional manner. The set of nickel-cadmium batteries recommended by the manufacturer as comparable to the lead-acid type for heavy-duty automotive use, did not measure up to expectations at subzero temperatures.

The sliding plate winterfronts on all but one unit and mastic thermal insulation were mechanically satisfactory and durable. Sheet Ensolite type floor mat was not. Glass-wool sheet insulation in battery boxes also showed signs of disintegration and separation. Slave receptacles on two items discouraged use for the purpose intended through inaccessibility. Flap valves or shutters in heater air intake lines or blowers intended for snow protection were not positively secured in the open position to prevent accidental closure during operation. Arctic engine oil (MIL-O-10295) was satisfactorily employed across the board and for the duration of these tests.

Proposals

It is generally recommended that:

1. The initial success of these winterized items not mislead all concerned into a false sense of complacency. (Practical accomplishments should be created from the exacting lessons brought home periodically from the Arctic.)
2. End-item manufacturers take advantage of these experimental units to adopt appropriate variations and effect recommended improvements for their commercial equipment applications (as forerunners of military versions) in Canada and northern areas of the United States.
3. More persistent and comprehensive low-temperature check tests be conducted, beginning with engineering materials themselves, to secure detailed design knowledge and performance log data for the record.
4. Instrumentation for engineering test measurements in the field be more carefully applied and checked out on the end items prior to delivery, to establish constant and corrective factors before use.
5. Manufacturers define, explain, and prominently identify special instructions or controls for unusual procedures, to facilitate operator understanding and familiarity under conditions of obscurity (such as darkness) and awkwardness (such as arctic clothing complications).

It is specifically recommended for accessories and components that:

1. Ease of maintenance and accessibility considerations for encumbered users be reflected all the way from individual component design to final end-item fabrication and assembly. The immediate relief should be to minimize use of tools and waste motions by (1) furnishing more quick disconnect fittings for plumbing and electrical circuits, (2) making all servicing points directly accessible, and (3) substituting hinged connections and a few large snap or pull-down latches or hasps for numerous bolts and screws in the frequently handled covers and panels.
2. More research be pursued to resolve the questions of how, when, and where to exploit best in a simple and economical manner the possibilities of priming as a starting aid for conventional engines at subzero temperatures.
3. The high-mortality heater parts (that is, igniters) be improved for mechanical and electrical durability and be designed for more easy replacement or adjustment. Torch-type heaters be improved to facilitate ignition and control, and to

[7] Final Report, Contract No. DA-44-009 ENG-1041, Pennsylvania State University for ERDL, September, 1955, "Priming Fluids for Cold-Starting Diesel Engines," by W. E. Meyer and J. J. DeCarolis.

minimize inherent hazards. Heater and/or ventilator air intake and exhaust arrangements be designed to prevent the ram effect of snow and wind during operation and storage.

4. The coordinated design of completely winterized personnel enclosure as an assembly be emphasized, comprising (1) rugged integrated structure of frame, doors, windows, and hatches, (2) optimum insulation, (3) regulated heating and ventilation, (4) maximum ease and comfort of an encumbered operator for the duty to be performed, and (5) in-cab starting controls.

5. Major recommendations of the Winterization Subcommittee, as announced in its second progress report, still stand as the immediate goal of further investigations in these matters under topics (1) batteries and electrical (or cranking) systems, (2) fuels and lubricants, (3) personnel and engine heaters, (4) cab design and defrosting, (5) insulation materials, and (6) battery boxes.

APPENDIX I

Ease-of-Maintenance Accessibility Check-off Chart [8]

	Tractors		
	X	Y	Z
1. Drive belts and adjustable pulleys	No	No	No
2. Spark plugs or injectors	No	Fair	Fair
3. Magneto adjustments	No		Fair
4. Governor adjustments		#	
5. Choke throttle and primer controls	Yes	Yes	Yes
6. Safety controls and shutdown devices	Yes	Yes	Yes
7. Coolant drains and thermostats		#	
8. Heater drain, burner, and controls	Fair	Fair	Fair
9. Heater wick, fuel regulator, and igniter	Fair	Fair	Fair
10. Heater disconnects and shut-off valves		#	
11. Heater wiring disconnects and switches	Fair	Fair	Fair
12. Heater accessory pumps		#	
13. Air filter and carburetor	No	Yes	Yes
14. Oil drain	Yes	Yes	Yes
15. Oil filter drain	Fair	Yes	Yes
16. Oil filter cap	Yes	Yes	Yes
17. Oil dip stick	No	Yes	Yes
18. Oil filler cap	No	Yes	Yes
19. Cranking motor brushes and flange mounting		#	
20. Fuel line connections	No	Fair	Fair
21. Fuel pumps, filter, and sediment bowls	No	Fair	Fair
22. Fuel tank filler cap and means to fill from dispenser	Fair	Fair	Fair
23. Engine timing marks (flywheel)	No	No	No
24. Engine connecting bolts to driven load		#	
25. Voltage regulators, battery-charging generators, and brushes		#	
26. Battery-charging receptacle	No	Yes	Yes
27. Mounting bolts and footings		#	
28. Storage batteries, cell caps, and terminals	Fair	Yes	Yes
29. Lubrication fittings	No	Fair	Fair
30. Air tank drains		#	
31. Hydraulic system drains		#	

No—Not accessible
Yes—Satisfactory
#—Not reported

[8] "Design and Ease of Maintenance," by R. C. Navarin and K. H. Usow. Presented at SAE Golden Anniversary Tractor Meeting, Milwaukee, Sept. 13, 1955.

APPENDIX II

Winterization Guidebook Contents

Sections	Coverage	Paragraphs
I	Introduction and Environmental Considerations	1– 3
II	Effect of Extreme Temperatures on Basic Materials	4–12
III	Low-Temperature Aspects of Fuels, Lubricants, and Related Fluids	13–20
IV	Critical Influence of Environmental Factors on Man and Materiel	21–36
V	Cabs for Vehicles and Personnel Heating	37–42
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VII	Internal-Combustion Engine Design Factors	49–52
VIII	Design Data Based on R&D Investigations Below 0 F	53–58
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DISCUSSION

Discusses Construction Equipment Parts Developments and Performance

—S. H. Knight
Northern Pacific Railway Co.

SUBCOMMITTEE XV should be complimented on the thorough manner in which they have investigated the subject of winterization of construction equipment for operation and use in temperatures between −25 and −65 F.

While the subject was investigated from a military standpoint,

it would appear that much information of value has been obtained that can be used by industry in northern United States latitudes as well as in Canada and Alaska.

In so far as we are concerned, there is no question but that during the past 10 to 15 years our use of certain types of construction equipment, and particularly crawler tractor bulldozers, in subzero weather has increased and has progressed. Where 15 years ago we used to tie up such equipment at the end of the normal season, today it is used during winter for snow and ice removal and other work. With a toothed scarifier, earth-moving operations on a year-around basis are continued throughout the winter even in Minnesota. There has been little or no recognition in the past by tractor manufacturers that unusual problems exist as a result.

From an industrial standpoint we are not interested in some of the features considered necessary by the military for arctic operations in temperatures to −65 F. However, we in this country and in Canada and Alaska find −25 F and lower not uncommon in which equipment stands unprotected for long intervals. So we do need some relief, bearing in mind that we must get along with ordinary commercial grades of fuels and lubricants for use in machines that are not new and not in top operating condition and where average maintenance only can be hoped for.

Batteries, when employed for starting purposes, have been our greatest problem. Those batteries that have been available to us in the past, and which so far as I know to the contrary still are, simply do not have what it takes to overcome the starting friction of a cold engine. Very frankly, the attitude of the battery industry has seemed to be "take it or leave it—we are not interested in your problems." Better batteries are needed, if that is possible. More battery is needed, and heat to the battery box and crankcase should be provided.

We have had some success with ether starting aids. Heaters, both gasoline and propane, experimented with in the past have not been successful principally because of insufficient heat, inadequate heat transfer, or both. There have been operating troubles, and mortality from broken fuel lines, brackets, and other attachments, together with ignition troubles, has been high. The Perfection E-510 heater referred to in the report sounds interesting.

We would not wish for an airtight cab enclosure, direct cab heat, or tinted glass, and it is doubtful if we would buy them or that they could be built at a price that we would be willing to pay. Our principal concern is over the long delays that are commonly experienced in attempting to start a cold engine that has stood and soaked in a below-zero temperature from 16 to 64 hr or longer, and what can be done to get it going. Let me emphasize here that engine starts are our prime difficulty.

Starting delays are not immaterial items of expense to contend with in the winter use of construction equipment. A question not yet answered is, can the winterized equipment be built and priced to sell at a figure which industry will be willing to pay and which will show a return on the investment?

Discusses Subzero Testing of Engines

—H. D. Young
Sinclair Research Laboratories, Inc.

THE development program carried out by SAE CIMTC Sub-Committee XV on winterization of construction equipment for subzero operation has been closely followed since its inception. Our chief interest in this project has been in the use of priming fluid application for starting equipment at these low temperatures. The work of the committee tends to confirm our own findings that in starting engines at subzero temperatures with priming fluids, method and rate of fluid introduction is important. It is also essential that an adequate supply of priming fluid be made readily available in order to keep the engine running after it has been started. The quantity of fluid required to keep an engine running until it is warmed up sufficiently to operate on regular fuel alone varies with the design and size of the engine as well as temperature level and is usually several times that needed to achieve a start.

Closely allied with our studies on priming fluids and their application has been a related interest in engine fixes necessary to improve starting characteristics at subzero temperatures. Studies in this direction have included the effect of changes in valve timing, valve and valve guide clearances, injection pressure and spray pattern, cranking speeds, fuel transfer pump characteristics at low speeds, and exhaust back pressure. Based on these studies it is felt that a great deal can be done towards improving the starting of engines at low temperatures through engine modification.

Many of these modifications become obvious at the temperatures at which the engines may be required to operate. Consequently, cooperative programs such as the one carried out by SAE CIMTC Sub-Committee XV are of great value in bringing into focus some of the problems involved. The Committee is to be commended for their efforts in this direction, and the authors congratulated for their able handling of a difficult subject.

Up to the time of the committee's active interest in this subject, very little cooperative effort had been expended in this direction. There is probably a very real reason for this, namely, the attendant difficulties, and expense involved in carrying out such a program.

Engine performance and equipment studies at subzero temperatures are probably among the most difficult to carry out. Numerous test variables are involved. Available facilities under which studies must be conducted have, until recently, been sharply limited. Instrumentation can be a problem. The work is costly. Lack of personnel enthusiasm for carrying out reliable studies at these low temperatures is frequently a factor to be considered. This is understandable in view of the difficult environment under which such studies must be conducted.

The subzero testing of engines, fuels, and lubricants is of vital importance from a military standpoint. Likewise, both the engine and petroleum industries stand to gain from such testing. Not only does it result in improved products for subzero use but the findings also may be applicable in the development of better products for use at higher temperatures. Considerable thought should therefore be given as to how such work can be more effectively executed. Cooperative effort such as undertaken by SAE CIMTC Sub-Commitee XV seems to be a step in the right direction. However, considerable evangelism is still necessary to convince many people that cooperative work along this line has important payouts for everyone and that this method of accumulating the necessary information is the quickest and most economical way of obtaining it.

ORAL DISCUSSION

Reported by D. Rinkema
Allis-Chalmers Mfg. Co.

Question: Could you give further information on battery performance and difficulties?

Answer: The lead-acid batteries operated satisfactorily while the nickel-cadmium battery was not satisfactory, probably due to the fact that the battery used was too small for the intended use in this test.

Question: Were engines protected on lower side as well as other sides?

Answer: No bottom guard equipment was provided. This was left open for air circulation to reduce the possibility of engine overheating during wide variations of ambient temperatures.

Question: Was an analysis made of any of the seals, metals, or components used on any of the vehicles?

Answer: There were no specific tests made on these items.

Question: Were any peculiarities noted in the preformance of the hydraulic systems at low temperatures?

Answer: There were none when arctic oil and the proper hose materials for −65 F were used.

THE GMR 4-4 "HYPREX" ENGINE

A CONCEPT OF THE FREE-PISTON ENGINE

FOR AUTOMOTIVE USE

A. F. Underwood, General Motors Corp.

This paper was presented at the SAE Summer Meeting, Atlantic City, June 5, 1956.

THE free-piston engine combined with a gas turbine has many attractive features for automotive use. Such an engine (Fig. 1) has been designed and installed in an automobile (Fig. 2) for evaluation. Our analysis and results indicate definitely that the engine will be a serious contender for powering automotive vehicles. Moreover the evidence obtained to date suggests that many engineering organizations will follow in this field.

The basic engine is over 30 years old and has been developed to the position of having hundreds of machines in operation throughout the world. All of these engines are either a unit of approximately 1000 hp installed in ships, locomotives, and power stations or are portable air compressors. As is customary in these fields, the engines are rather large and heavy in relation to their output. There have been other designs somewhat smaller and larger than this 1000-hp unit, but none of them has reached a state of commercial application. In 1952, R. Huber proposed a scaled-down version of the large unit for installation in a truck.[1]

The project reported in this paper presents a completely new design which has been developed for

[1] *Journal* of Société des Ingénieurs de l'Automobile, Vol. 24, 1951, pp. 267–274: "Les Générateurs à Gaz à Pistons Libres," by R. Huber.

DESCRIBED here is a 250-hp free-piston gasifier-turbine engine that has actually been installed in an automobile.

A unique feature of this Hyprex engine is that it is a siamesed unit. The overall design has been selected, according to the author, to secure a compact, light-weight machine with improved thermal efficiency and with a reduction in general noise.

Although the engine is still in the experimental stage, the author reports that analysis and results indicate it will be a serious contender for powering automotive vehicles.

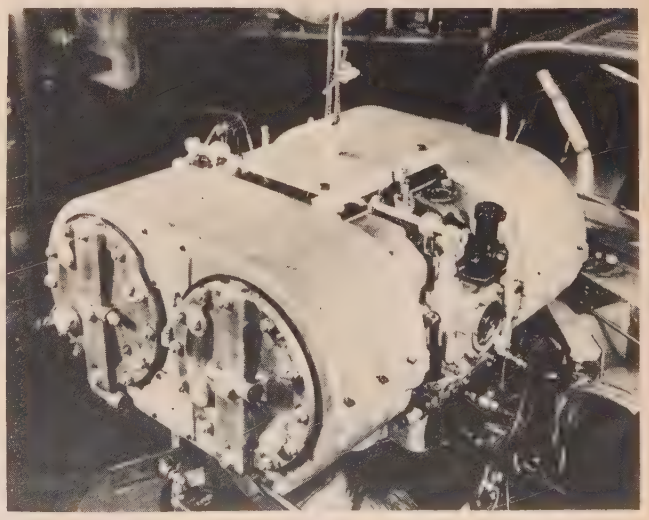

Fig. 1—GMR 4-4 "Hyprex" engine

Fig. 2—XP-500 experimental car in which free-piston engine has been installed

automotive applications. It is based on the concept of a small light-weight engine of 250 hp and is the first siamesed unit ever to be made. Such a configuration was selected to secure a compact machine of light weight with improved thermal efficiency and reduction in general noise.

This new engine offers two other unique advantages for automotive use: first, no tunnel is required in the body and, second, this engine can definitely be said to be the first one which allows the engineer to secure the desirable weight distribution by putting the engine in the front of the car, with the transmission in the rear and yet having no humps for the transmission or propeller shaft, to interfere with the passengers' feet.

If any new powerplant is to be accepted for mass production, it is necessary that there be good, sound engineering reasons for its use. It is not sufficient that a new engine be made to run or that it have most of the advantages of presently available engines. In addition to the performance of current machines, it must have additional features which make it attractive to both the engineer and the customer.

Advantages of Free-Piston Engine

The free-piston engine when combined with a gas turbine has the following interesting advantages as compared to other powerplants. These are not necessarily listed in the order of importance; however, in most applications it will be found that each one of these items is necessary for the installation to be a success. Of course, in certain situations it may be of the utmost importance to use one particular advantage over all of the others. As an example, the question of tolerance to various kinds of fuels could well be a determining factor in the installation for a military vehicle.

Insensitivity to Fuel—Over a period of many years it has been demonstrated that the free-piston engine is not sensitive to whether it is operating on kerosene, No. 2 diesel oil, or Bunker "C." Tens of thousands of hours of satisfactory operation have been run with such fuels having a sulphur content as high as 4%. However, we have extended the range of fuel testing and have demonstrated that the engine can operate successfully on gasolines of varying octane up to 100 octane and equally as well on certain crude oil which has not had any refining. We have even demonstrated that the engine will run satisfactorily on such combustible materials as whale oil, peanut oil, and cottonseed oil, although operation has not been long enough to determine what effect such oils would have ultimately on the operation or condition of the engine. As would be expected, the change in horsepower output is proportional to the heating value of the fuel.

If continued testing consistently demonstrates the insensitivity of the smaller-sized free-piston engine to the type of fuel, then it is obviously of considerable importance to the supply and use of fuel in our country. While it may be premature to predict that the fuel will not have to have any specifications, it is certain that the specification can be held to a minimum compared to the requirements of other engines. As one example it might be desirable to have a minimum viscosity for automobile use so that the fuel will not have to be kept heated so as to have a low enough viscosity for the injection pumps. Yet in a locomotive application it is conceivable that the complication required to keep the fuel hot would be offset by the saving in the use of cheap fuels. It will be interesting to have an engine which is not definitely designed and limited by many requirements of fuel quality.

High Efficiency—The overall efficiency of the engine is the product of the gas efficiencies of the gasifier itself and of the turbine. The gasifier operates at a compression ratio between 30 to 1 and 50 to 1 and, therefore, has extremely high thermal efficiency. The gas is generated at an efficiency up to 40–45% at full load and with a turbine efficiency of 80% this gives an overall thermal efficiency of 32–36%. This makes it competitive with high compression-ratio gasoline engines and diesel engines which we have today. In the part-load range, the overall efficiency can be comparable to the current engines.

As far as the customer is concerned, his desire is to obtain the most horsepower-hours for a given amount of money. If now, the free-piston engine uses a cheaper fuel at the same thermal efficiency as a crankshaft engine, then as far as the customer is concerned the efficiency of his fuel dollars is higher when he uses a free-piston engine. It is interesting that up to 60% of the operating cost of a locomotive is for fuel.

Smoothness—All free-piston engines are inherently balanced and require no special design or attention except that all of the similar oscillating or reciprocating parts must have exactly the same weight. This means that the motor supports can be located without regard to torsional movement, since there is no such reaction as in the crankshaft engine. Motor mounts

are located to give proper support to the engine without strain from expansion or distortion of the chassis frame. The smoothness is frequently demonstrated by balancing a nickel on a horizontal machine surface.

An example of the inherent balance of a free-piston engine is demonstrated by a 1250-hp unit which we have been running for three years. This gasifier is mounted several feet above the floor on a supporting frame of I-beams. Each piston assembly weighs 1145 lb and reciprocates at approximately 600 cycles per minute. Although this engine and frame have never been bolted to the floor, there has never been the slightest movement of the unit.

Torque Multiplication—One of the attractive advantages of any cycle which uses a gas turbine is the ability to produce a high torque because of the stall characteristics of the turbine. In most present-day automotive applications, it is necessary to have good performance which means high low-speed torques. It is believed that a simple 2-step transmission may be satisfactory to match the performance of current transmission drives.

Low Gas-Turbine Temperature—The outstanding advantage of the free-piston-engine cycle over the combustion turbine cycle, whether regenerative or nonregenerative, is the very low temperature of the exhaust entering the power turbine. The temperature is in the range of 450 to 900 F. This is due to the fact that the energy for compressing inlet air has already been extracted from the gas before the exhaust gas plus the compressed air, is piped to the power turbine. On the other hand, as is well known, the combustion turbine must extract the power energy and the air compressor energy from the power turbine, thereby requiring high gas temperatures into the power turbine.

The low gas-turbine temperature results in a number of real and definite advantages.

Of prime importance is the fact that noncritical materials are used in the power turbine. If it is desired to construct large numbers of turbines, then it is obviously necessary to make them from alloys which are readily available and which have moderate cost. Alloys such as silchrome No. 1 containing only chromium and silicon will be used. The low turbine temperature means that elements in the fuel such as vanadium compounds will not cause harmful corrosion of the blades. If the turbine blade temperatures are kept below 1100 F, then there is no difficulty with such elements.

A second advantage is the relative ease with which the gasifier can be connected to the turbine. The power gas is at a lower temperature than the exhaust gas of a gasoline engine, even at full load. The low-temperature gas can be readily piped from one part of the vehicle to another. This gives the designer an independence of location of the gasifier and the turbine. The gasifier can be placed where it is most convenient for weight or space consideration, and the turbine can then be located where the power is to be applied to the wheels. The advantages over a gasoline or diesel engine are obvious, because in such cases it is necessary to have a propeller shaft or other mechanical connection between the engine itself and the final drive.

Power-to-Weight Ratio—Our analysis and design leads us to the conclusion that a free-piston-engine and turbine combination can approach the ratio of the present highly developed gasoline engine. Depending on how one rates the power output of such an engine, the present ratio will be 3 to 5 lb per hp. A fair comparison would include all the components of each system, up to the final power drive.

Throttle Response—For vehicle use it is highly desirable that the engine respond quickly to any change in throttle setting. Thus, if the engine is idling and it is desired to quickly accelerate the vehicle, the power must be made quickly available to the output shaft. One of the remarkable features of a free-piston engine is the rapidity with which the gasifier can increase its output of gas. There is no question of accelerating moving parts over a period of time. The pistons in the gasifier must stop and start at the end of each stroke. Therefore the speed with which the free-piston engine can be made to produce more gas is entirely dependent upon the speed controls. These can be made to operate quickly, and therefore, the response is prompt.

Consider next the turbine which consists only of the power wheel and reduction gears. For bus and truck use, the inertia can approximate that of the rotating parts in a present-day automatic transmission.

Deceleration is just as rapid because the gasifier will slow down in its production of gas directly as the fuel flow is reduced. In fact, if the fuel is cut off, the engine will stop on the next stroke of its pistons. Naturally, the gas in the pipe connecting the gasifier to the turbine must be expanded before there is zero torque on the turbine, or a bypass valve may be opened to dump the gas in an emergency.

Use of Afterburner—The gas from the gasifier still has 80% of the original oxygen; in other words, it still has an oxygen content of about 16% at full load. As has been pointed out previously, the temperature is on the order of 850 F. By placing an afterburner between the gasifier and the power turbine, additional fuel can be burned to supply an augmented power to the turbine. This feature is particularly desirable in a vehicle where acceleration and peak power may be needed for short periods of time. The power can be raised for top performance with little additional weight and at a very low loss of overall thermal efficiency. In fact a temperature increase up to 250 F could be used for continuous operation. Even higher temperatures could be utilized for short periods which would not last long enough to overheat the turbine blades and parts. As a first approximation, the power is increased in proportion to the absolute temperature of the gas entering the power turbine. A particularly important feature of this

method of obtaining increased power is that it is almost instantaneously available.

Wide Application—From the above items one can conclude that such an engine will have a wide application to all types of vehicles. Appropriate units can be designed to meet best the needs of installations in automobiles, buses, trucks, and military vehicles. As with current automotive engine design, the best configuration will depend on the required power-to-weight ratio, endurance requirements, possible use of afterburner, and the need for minimum fuel limitations.

An attractive feature of this free-piston gasifier is the basic "building block" which it gives the engineer. A family of engines can be built with practically the same components. Thus a single-cyl engine would be the smallest engine. Next, would be a siamesed unit. Higher horsepower would be developed by two or more of these units feeding into a common header to the power turbine. Built up in this manner, all engines would be serviceable from one set of parts, especially those parts which would need replacement. Obviously, there would be two engine cases (a single and a siamesed) and different power turbines. This feature brings the free-piston engine into the situation enjoyed by gasoline and diesel engines where the same components can be used for several engines by a change in block configuration. Producing such a family of engines is attractive because it can be done on tools similar to present crankshaft engine tools.

Free-Piston-Engine Cycle Wtih Gas Turbine

The engineering literature contains many papers on this type of engine. Particular mention should be made of the numerous papers by Huber,[1,2] Muntz,[3] and London.[4] As has previously been pointed out, these refer to engines which are commercially available and in general are of higher horsepowers and not designed for vehicle use. Nevertheless, the basic principle is the same for these engines and the engine described in this paper.

As shown in Fig. 3, the engine is basically a uniflow, opposed piston, 2-cycle diesel engine with a built-in air piston compressor for scavenging purposes. Considering only the two power pistons, they function in the same manner as the opposed piston type of 2-cycle diesel engine. The exhaust and intake ports are uncovered at the end of each stroke. At this time the compressed air from the air-box case blows through the diesel cylinder, reducing the temperature of the engine parts and scavenging the combustion chamber for the next portion of the cycle. Fuel is injected by a standard type of injector when the pistons are approximately at their inner dead point. The explosion forces the two pistons apart in the usual manner. All of the mechanical energy for the cycle is developed during the expansion of the gases between the pistons. A simple linkage, usually consisting of a parallelogram, keeps the two power pistons in phase. This linkage has very little force to transmit because it is required only to take care of differences in friction of the pistons, and such components as the fuel injector and controls.

In the usual diesel engine, the air for scavenging is supplied by a mechanically driven blower or by a turbocharger. In the case of a free-piston engine, the scavenging air is obtained by the air compressor pistons which are directly attached to each of the power pistons. The reciprocating motion of the air pistons draws air in through the intake valves on the outward motion of the pistons and forces the air into the air box through the delivery valves during the inward movement of the pistons.

It has been pointed out that this feature of the

[2] Application Marines et Stationnaires de Groupes Générateurs à Pistons Libres," by R. Huber. Presented at Congress International des Moteurs à Combustion Interne, The Hague, 1955.

[3] Transactions of Institute of Marine Engineers, Vol. 66, 1954, pp. 201–215: "Free Piston Gas-Generator Turbine as Powerplant for Ship Propulsion," by F. A. I. Muntz and R. Huber.

[4] ASME Transactions, Vol. 77, 1955, pp. 197–207: "Free Piston Engine and Turbine Compound Engine—Cycle Analysis," by A. L. London.

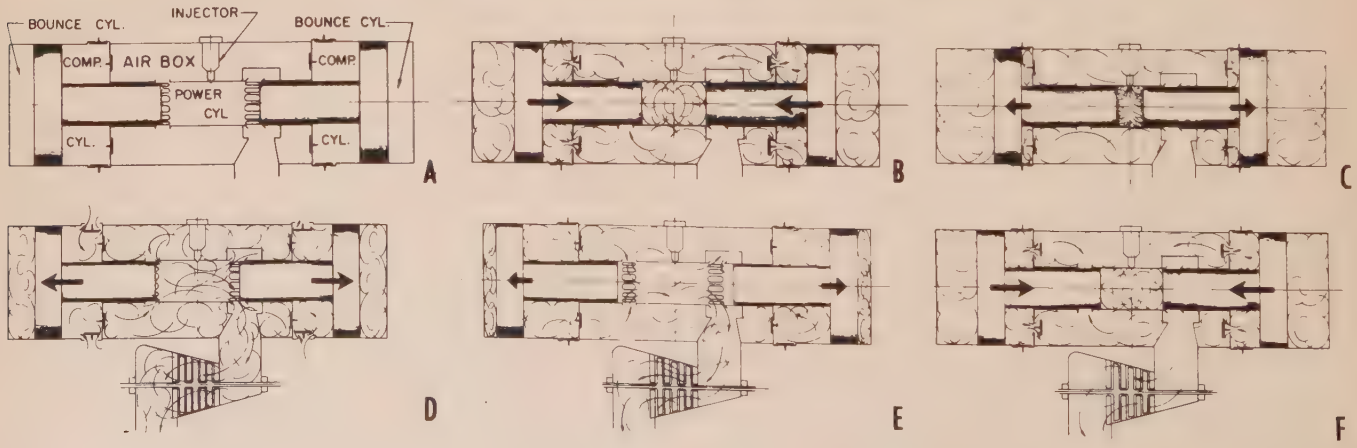

Fig. 3—Operating sequence of inward-compression free-piston engine

free-piston engine results in one of its principal advantages. To understand why this is so, let us start with the burning of the fuel in the highly compressed air of the combustion chamber. As it expands, energy is used (1) to compress the air in the bounce chamber and (2) to draw air into the air compressor cylinder. These energies are transmitted by the piston assembly. Now, when the bounce-chamber air returns the piston assembly, the energy for delivering the compressed air to the air box comes from the energy which was stored in the bounce chamber and which in turn comes directly from the expanding gases after combustion. Therefore, the energy for compressing the scavenging air into the air box is extracted mechanically from the gases within the combustion chamber. This transfer of energy means that exhaust gases coming from the exhaust ports when they are uncovered are reduced in temperature. In addition there is mixing with the excess scavenge air blowing through the combustion chamber, causing a further lowering of exhaust temperature. Since these exhaust gases are what drive the power turbine it can be seen why the turbine operating temperature is low.

It might be helpful to the understanding of the importance of this feature, to compare the energy relationships in a combustion turbine. In this engine, the energy is also produced in a combustion chamber. However all of the energy is directly transferred to the power turbine by high-temperature gas. Here, the energy required for the air compressor is extracted and fed back to the compressor turbine. If separate compressor and power turbines are used, the compressor turbine is subjected to the temperature of the high energy gas, even though only the compressor energy is extracted.

The important conclusion is that (1) in the free-piston engine-gas turbine, the power turbine receives low-temperature gas because the work of compressing the air has been removed, whereas (2) in the combustion turbine, the power turbine receives high-temperature gas because it must produce the added energy to compress the air.

The air intake and delivery valves are of the reed type. Mechanically operated valves cannot be used in a free-piston engine because of the variable stroke, and their added complication would add nothing to the efficiency of the cycle.

The bounce chambers at the outer ends of the compressor pistons, absorb energy from the expanding gas in the power cylinder while the pistons are moving toward their outer dead points. In effect, the bounce chambers can be thought of as air springs. The rate of the "air spring" in the bounce chambers is effectively controlled by the amount of air which is in the bounce chamber. Now the speed at which a free-piston engine operates is determined by the mass of the moving parts (primarily the air pistons and the power pistons) and the spring rate of the bounce chambers. Therefore, to change the speed of a free-piston engine, it is necessary to adjust the amount of air in the bounce chamber. This is accomplished by a device which has become known as the stabilizer.

The stabilizer is essentially a 2-way air valve which allows air to be transferred from the bounce chambers to the air box. During one complete cycle there are times when the air-box pressure is higher than the instantaneous bounce-chamber pressure, and there are times when the reverse is true. If it is desired to increase the speed of the engine, then the stabilizer allows an appropriate amount of air to enter the bounce chamber from the air box. This increases the spring rate and causes the mass of the pistons to operate at a higher frequency. By removing air from the bounce chamber, the engine is slowed down.

A free-piston engine such as shown in Fig. 3 will not produce less than about 25% of full load. Until recently it was necessary to discharge part of the exhaust gas to the atmosphere if it was desired to operate the power turbine at less than 25% load. Obviously this was a very inefficient situation and resulted in extremely poor part-load economy. As an example, the early models of free-piston engines required about one-fourth of full-load fuel rate for idle. In other words, it was as inefficient as the non-regenerative combustion gas turbine.

Today a method known as "recirculation" is employed to improve the part throttle and idle efficiency. This is simply the action of bypassing part of the air in the air box back to the air intake of the compressors. The effect of such recirculation is to heat up the incoming air resulting in two improvements. First, it raises the temperature of the air in the combustion chamber at part throttle and idle, thereby allowing combustion at lower compression ratio. The lower compression ratio (10 to 1) is the result of reduced stroke, primarily a larger inner dead point. A softer cushion under these conditions, in turn reduces the speed and, consequently, the power output with reduced fuel consumption. Secondly, it reduces the pounds of air being pumped by the engine and, therefore, a dump valve to the atmosphere at low loads is not required. On certain free-piston engines now in use, fuel consumption at idle has been reduced at 8% of full-throttle fuel requirements.

The quantity of air pumped by the gasifier to the turbine is determined by several factors. The first factor is the speed of the engine which is determined primarily by the mass of the parts and the spring rate in the bounce chambers. The second is the variable stroke. A relatively small change in the effective stroke of the pistons produces a large difference in the pounds of air pumped by the gasifier. The difference between minimum outer dead point and maximum outer dead point can make a difference of 45% in pounds of air pumped per minute. The rate of fuel injection is the primary stroke regulator, especially in its effect on outer dead point. Bounce pressure is the secondary stroke regulator, primarily on inner dead point but also on outer dead point.

One of the functions of the stabilizer is to insure

that the air pressure in the bounce chambers is at the proper value consistent with the amount of fuel injected for each stroke, to make the engine operate within the desired stroke limits. If, for example, the air pressure in the cushion is too low compared to the amount of fuel which is injected, then the engine will overstroke, and the pistons will hit against some part of the engine. If the air bounce pressure is too high for the amount of fuel which is injected, the engine will understroke and stop running.

Since there are no rotating parts as in a crankshaft or a gas-turbine engine, the fuel injector is operated by the linkage which connects the two piston assemblies. Because the linkage is not moving at the inner dead point, which is approximately when injection should occur, it is necessary to employ either a high-rate cam if mechanical injection is to be used, or to utilize an accumulator system which can be tripped at the proper time.

In actuality the engine operates at a variable compression ratio. For example, when the air-box pressure is one-half atmosphere, the compression ratio is in the neighborhood of 25 to 1. At full power when the air-box pressure is 3 atmospheres, the compression ratio approaches 50 to 1. Under the latter conditions the compression pressure is the order of 1200 psi and the firing pressure is less than 2000 psi. As would be expected, the temperature and pressure prevailing in the combustion chamber under these conditions are so intense as to cause effective and efficient combustion. The result is high efficiency in the burning of fuel. Also these conditions are the reasons for the relative insensitivity of the free piston engine to the quality and character of the fuel. The completeness of the combustion has been demonstrated by the tests which we have made for thousands of hours on one engine with no appreciable deposits in the ports or exhaust pipes even on Bunker "C" fuel.

The turbine in this cycle acts as a fixed orifice. At least this is true for the first approximation. Therefore, the air-box pressure of the gasifier is effectively equal to the pressure drop through the turbine plus the pressure drop through the power cylinder (including intake and exhaust ports) plus the loss in the piping connecting the gasifier to the turbine. The turbine recovers a large percentage of the energy put into the compressing of the scavenge air, which is another factor for the high efficiency.

Reasons for a Siamesed Gasifier

The most novel feature of the concept of the Hyprex engine is that it is a siamesed unit. By siamesed is meant a common case for two complete cylinder assemblies, a common air box, a common air-inlet housing and a common exhaust. This configuration was selected after careful consideration of the many types of possible designs. It should be pointed out that gasifiers have been "twinned." Twinning is accomplished merely by running the exhaust from two gasifiers into a common header and dephasing the two machines by an appropriate pneumatic device.

One of the principal differences of opinion among free-piston-engine engineers is whether the engine should be an outward compression type or an inward compression type. This terminology refers to whether the air compressor pistons compress the scavenging

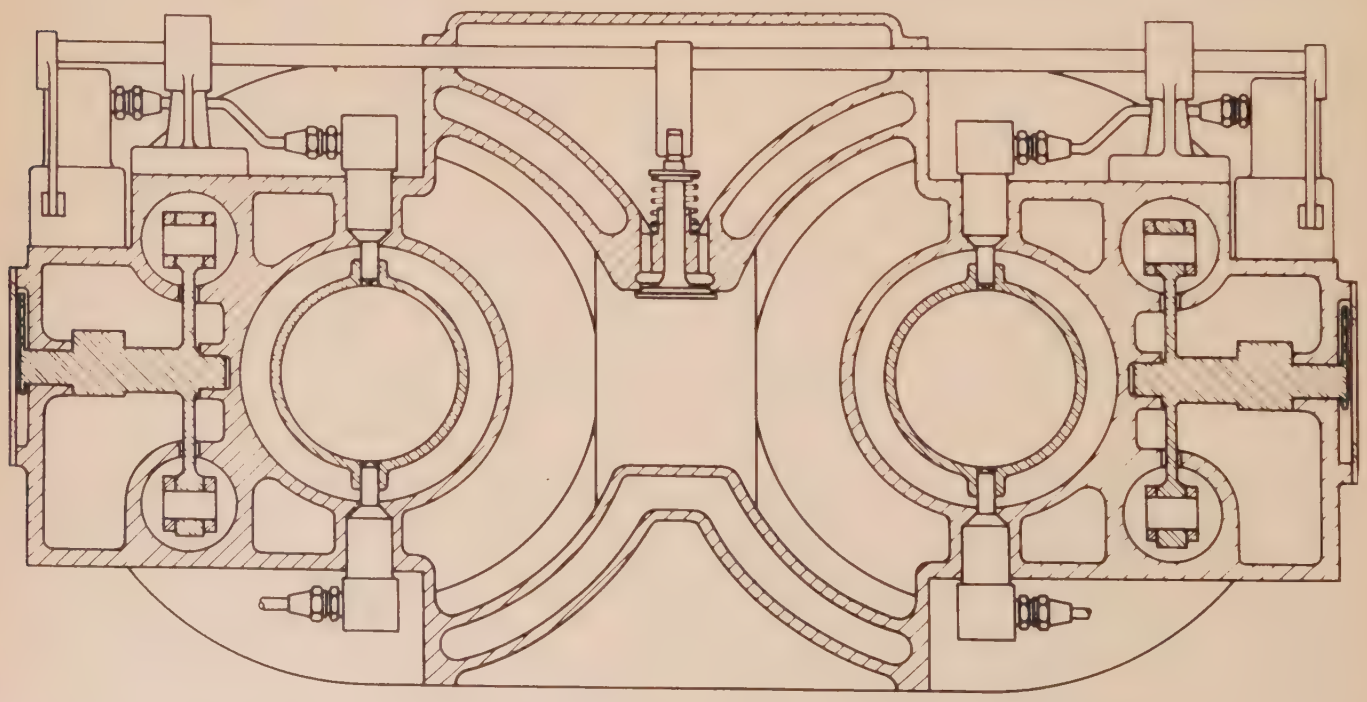

Fig. 4—Section of GMR 4-4 "Hyprex" engine

air into the air box during the inward movement of the piston assemblies or during the outward movement of the piston assemblies.

In the case of a single-cyl outward-compression gasifier, the engineer finds an improvement in efficiency over an inward compressing type. This is brought about by the fact that the outward compressing engine supplies its air to the air box during the time when the exhaust ports are open and the engine is being scavenged. The result is a lower pumping loss and, therefore, a small percentage gain in overall efficiency. The disadvantage of an outward compressing machine is that external air pipes must be added to conduct the compressed air from the ends of the machine to the air box. Also, in order to get good control of the engine speed, the bounce chamber is usually added to the end of the air compressor cylinders. This results in an overall increase in the length of the machine.

The inward compressing type has the advantage of smaller dimensions both in length and height. Another real asset is the comparative ease with which such a machine can be assembled and disassembled for inspection. Therefore it is obvious that if the inward compression engine can be made to give the slightly improved thermal efficiency of the outward compression engine, the arrangement would be one which combined the better points of each configuration. The siamesing accomplishes this because with dephased assemblies in a common case, one cylinder is being scavenged while the other power-piston assembly is compressing the scavenging air into the common air box.

Another important advantage of the siamesed design is the compactness of the engine for a given horsepower and the more economical use of space particularly if one considers the volume under the hood of a passenger car. It is obvious that a single-cylinder engine of the same horsepower as a siamesed unit would have to be considerably longer. This can be demonstrated by the preliminary design of a 250-hp engine based on a single cylinder. It was projected to be over 60 in. long. By twinning the same horsepower output could be attained in an overall length of 40 in. Less engine noise would be expected from siamesing due to more even air-intake pulsations and to having two cyinders firing at regular intervals.

While there is currently no definite information on the effect of the pulse discharges from the gasifier on the efficiency of the gas turbine, it is believed desirable to have as uniform a gas pressure as possible at the turbine inlet. The current practice with single-cyl engines is to include an exhaust-gas receiver to minimize the pulsations. A siamesed unit does this without the use of a large exhaust receiver.

Description of the Hyprex Engine

The first consideration in laying out a new engine is to determine the horsepower or performance characteristics which it must meet. The present engine was based on a nominal 250 gas hp. Because of the low-speed characteristics of the gas turbine, this would give a performance comparable to what is currently referred to as a 300-hp automobile engine. The nominal dimensions of the gasifier are 40 in. long, 34 in. wide, and 18 in. high. The model number of this engine is the GMR 4–4, denoting 2 cyl of 4-in. bore.

Using the siamesed principle the power cylinder is 4 in. in diameter, the air compressor piston is 11 in. in diameter, and the stroke is approximately 5 in. A maximum cyclic speed of 2400 strokes per min was selected. The actual cyclic speed is determined by the weights of the moving parts, the amount of recirculation, and the fuel rate. These have been varied in our testing and the minimum speed can be in the order of 100 cycles per min.

As can be seen from Fig. 4, the engine does have a common air-inlet housing, a common air box for scavenging, and a closely siamesed exhaust outlet.

The inlet valves and exhaust valves (Fig. 5) are identical. There are eight inlet valves and six delivery valves for each air compressor. Previous tests have indicated that the reed type gives the best performance with respect to highest operating speed and best volumetric efficiency. The design is such that the bodies can be die castings and the blades are simple steel stampings.

The steel pistons are of a conventional design with heat dam and oil cooling. The oil for this purpose is also used for general lubrication of the engine parts such as the connecting linkage. Lubrication for the power pistons is provided directly to the cylinder walls and indirectly from the air compressors. The wear bands on the piston are made of aluminum-coated steel. Tests on a larger engine indicated that the one component which was affected by sulphur in the fuel, was the copper-lead-coated wear bands. By the use of the special aluminum alloy during several thousands of hours of testing, it has been demonstated that there is no chemical attack.

A separate fuel-injection pump and nozzle are used for each cylinder. The pump is actuated by a cam on the linkage mechanism. As with all free-piston engines it is essential to inject rapidly at the correct

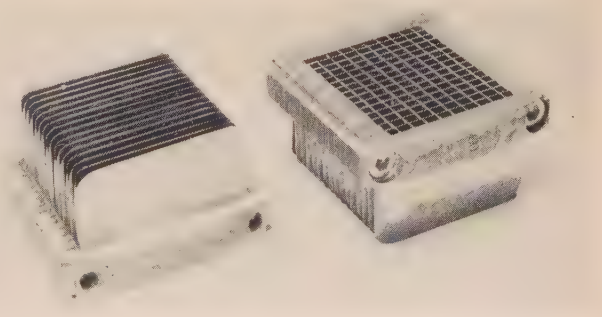

Fig. 5—Intake and delivery valves of identical reed-type construction

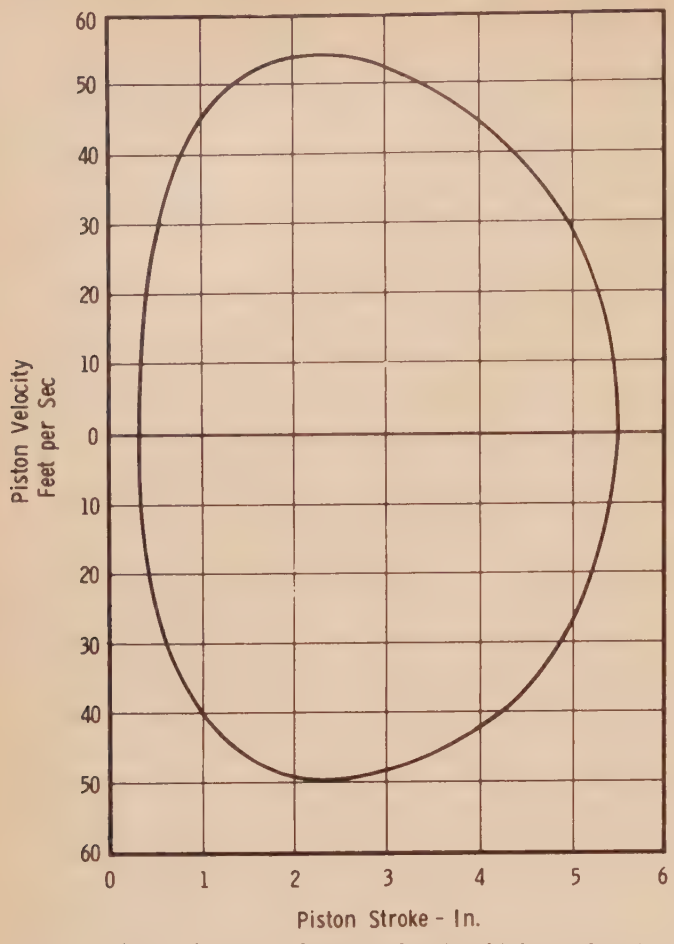

Fig. 6—Velocity diagram of piston showing high acceleration after firing

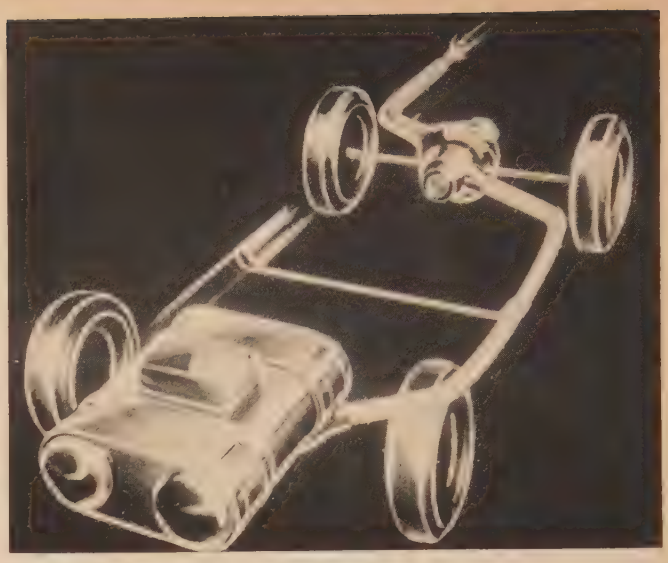

Fig. 7—Phantom view of arrangement of powerplant in XP-500

time because the pistons accelerate rapidly from their inner dead points. The high acceleration is the result of having to accelerate only the piston assemblies, whereas in a standard piston engine the inertia of the crankshaft and flywheel restrain the pistons from moving as rapidly just after top dead center. Fig. 6 illustrates by a velocity diagram the high acceleration after firing.

The successful operation of a siamesed unit is dependent on the dephaser. Its action is to keep the two piston asemblies 180 deg out of phase. The pneumatic device used on this engine obtains its signal from the bounce-chamber pressures. In general, the action is to accelerate or retard the cyclic speed of one piston assembly in relation to the other piston assembly in order to maintain the proper timing. The dephaser has functioned satisfactorily from the beginning of our siamesed-engine test and has been a strong reason for our confidence in this design of an engine. Actually, the engine starts in parallel operation and then within a few strokes is forced into dephased operation.

The starter follows the usual practice of putting the pistons into a proper position near their outer dead points. Admission of air into the bounce chambers suddenly forces all of the pistons to their inner dead points, compressing the air in the combustion space. Fuel is injected, combustion takes place, and the piston assemblies are accelerated outward. Then the control mechanisms must take over and determine promptly the correct cushion pressure for the amount of fuel which is being injected so that the engine will not understroke. As with all free-piston engines of the inward compression type it is essential that proper firing be present on the first stroke as there is no carry over of inertia for a second stroke. The most critical stroke is the second, as the controls must "take over" and adjust the bounce pressure in relation to the injected fuel or the engine will stop. In other words, if the engine does not start on the first stroke and if the controls do not properly regulate thereafter, it is again necessary to go through the starting steps. Air pressure for starting can be obtained from air bottles or from an air compressor of sufficient capacity to furnish the air directly for each start. Air pressure of 30 psi in the bounce chamber is adequate to start this engine.

From the engine the exhaust is conducted through a 3½-in.-diameter pipe which runs through the left frame of the car as shown in Fig. 7. A standard type of insulation surrounds the pipe to reduce losses and to minimize the heat which would enter the body in the summer. It has been suggested that the heat losses might be used for car heating during the winter. It should be pointed out again that the low temperature of the gas allows it to be conducted from practically one end of the car to the other with a minimum of difficulty. It would certainly be much more difficult if the gas temperatures were 1600 F or higher.

The present gas-powered turbine is a 5-stage axial-flow unit. Its stall-torque characteristics are those which are normally found in such a turbine; as would be expected, the turbine gives excellent torque multiplication in itself. The power output from the turbine

is through a gear box having a reduction ratio of approximately 7 to 1.

The gearbox is bolted directly to the front of the transmission. Since the front elements of the transmission are running whenever the gasifier is in operation, certain accessories such as the electric generator, power steering, and water pumps are driven directly from the transmission. Because of the time involved in designing a completely new transmission, an available 4-speed unit is used. In turn, the differential is bolted directly to the rear of the transmission. Naturally this means that the rear wheels are independently suspended in order to allow the mass of the turbine gearbox, transmission, and differential to be fastened solidly to the chassis.

It has been pointed out earlier that when used with a dephaser the siamese design spaces the suctions to give a more even intake situation. Without a satisfactory dephaser the two cylinders will probably run in parallel and accentuate the intake-noise problem. Or if the dephaser does not work properly there would be a hunting cycle to the intake noise. The volume between the hoods and the outside of the case proper, acts as a dampening volume between the inlet venturi and the reed intake valves.

A free-piston engine operates over a considerably narrower range of speeds, compared to a crankshaft engine. This means that the idle speed cannot be reduced to as low a value in the gasifier-turbine combination. However, by employing silencing methods plus recirculation, the intake air noise on the XP-500 has been held to a quite satisfactory level.

The exhaust noise is extremely low. The 5-stage turbine in the rear of the car smooths out the power pulses from the gasifier. No muffler is employed after the turbine. The exhaust gas from the turbine is discharged underneath the trunk of the car through a short single pipe.

Performance of the Hyprex Engine

Up to the present time the most important conclusion from the performance tests, is that it is possible to design and build a siamesed unit having a power-to-weight ratio suitable for automotive use.

At the time of writing this paper, complete engineering data were not available, but are expected to be the material for later papers. However a specific fuel value under 0.45 lb per gas hp-hr was obtained during the preliminary testing.

On the basis of 25,000 hr of testing on various free-piston engines of the same approximate bore it is expected to reach soon a value of 0.40.

As with all internal-combustion engines, the specific fuel consumption increases at part load. Our analysis indicates a value comparable to those for other automotive engines.

The efficiency of the turbine is as important as the efficiency of the gasifier. Because of the desire to obtain immediate preliminary data, a complete map of turbine performance has not been completed. The turbine on the XP-500 is the first one we have ever made for a gasifier. However, it has shown an efficiency of about 70%, and very fortunately this efficiency is maintained over the speed range for the road-load requirements of the automobile. That is to say, the efficiency when running at light load and low speed is the same order as when running at high load and high speed.

As with all turbine installations the torque multiplication gives increased performance over the crankshaft engine of the same nominal horsepower. Another way of expressing this is to say that the transmission engineer has a new opportunity to develop a simpler transmission which utilizes the special characteristics of the gas turbine.

An appropriate question is, "How should the power and performance of a gas turbine and gasifier be rated?" Last year we proposed to the SAE that a special committee give this matter consideration at an early date so that a code could be established before various companies select their individual choices. In addition to the important question of nomenclature, the test code should answer such questions as:

1. How is horsepower of the gas measured?
2. What determines the maximum horsepower rating?
3. How should the efficiency of the turbine be rated?
4. How should the torque multiplication of the turbine be determined?
5. What temperature and barometric corrections should be made to the observed horsepower values?

Chassis and Body

The chassis was especially designed for the XP-500. (See Fig. 7.) The side frames are large enough to house the insulated pipe connecting the gasifier to the turbine on the left-hand side. The right-hand frame is used to conduct hydraulic-power tubing and other lines from the rear to the front of the car. Because the engine has a small overall height, cross members can be placed across the engine compartment to tie the side frames together, thereby stiffening the frame. A standard automobile fuel tank is contained in the rear trunk of the car.

The body was designed by the General Motors styling section specifically to take advantage of the fact that there is no requirement for a hump on the front-seat toe boards or, for a tunnel to the rear of the car. It is the first car in many many years in which the middle passenger in the front seat has as comfortable a space to place his feet as the other passengers. Rear-seat passengers find a similar advantage.

Other advanced features of the suspension, steering, and rear-wheel drive have also been described;[5]

[5] See pp. 471–477 of this issue, "Experimental Chassis for Firebird II," by J. B. Bidwell and R. E. Owen.

this has been done to allow further evaluation of these advanced concepts in automotive engineering.

Future Programs

As with all new engines, it is easy to predict a continued improvement in the power-to-weight ratio by the use of lighter metals and better utilization of all metals as experience demonstrates how to control the stresses and deflections. Further control and study of the combustion process could well bring about a continued decrease in fuel rate to a value better than 0.36 lb per gas hp-hr.

With the tremendous advantage of low turbine temperatures, continued engineering on turbines for this particular application will produce both more efficient and cheaper turbines. Special consideration will be given to radial in-flow turbines and turbines of the positive displacement type.

To better meet the turbine characteristics and the vehicle requirements, the gasifiers will have a higher speed and a greater speed range. Currently the maximum pressure of the exhaust gas to the turbine is about 3 atmospheres, it is reasonable to expect that the power output will be increased by some elevation of this value.

For some applications it will be best to use a single-cyl gasifier with an afterburner. If the vehicle is to be used most of the time at some low horsepower and only requires peak horsepower for limited periods such as for acceleration or for short periods of sustained top speed, then the use of the afterburner appears to be most promising.

While the intake and delivery air valves are completely satisfactory at the present time, continued advances will be made to allow increasing cyclic speeds, better efficiency, lower production cost, and optimum numbers of reeds. It is safe to predict developments in the method of driving the accessories and the simplification of starting systems.

It should not be concluded from this paper that there are no problems with this type of internal-combustion engine. This engine is still in the early experimental stage of evaluation and development. Just as there are today more engineers than ever before working on the gasoline engine after 60 years of development, there are and will continue to be problems both of improved durability and better performance.

Acknowledgment

To carry such a conceptual design to an evaluation in a few months, required the experience, skill, and continued interest of a large number of persons. Invaluable consultation was had from R. Huber, S.E.M.E., who made the basic engine design to conform to our requirements. The design of the gasifier and the reduction gears was under the supervision of W. H. Smith. The turbine was furnished by W. A. Turunen, gas turbines department.

Especial acknowledgment is made of the effort and contributions of G. Flynn, Jr. who has been associated with the project during its entire development and who has coordinated the project work. General supervision was the responsibility of W. H. Percival, with particular reference to construction and testing.

Each of the many engineers, designers, and mechanics who participated in this program deserve credit for their interest and their work.

DISCUSSION

Design Limitations Require Altering Lubrication System

—C. E. Habermann

Socony Mobil Oil Co., Inc.

WE have had a long-standing interest in the development of free-piston engines. Our contact has been close over the 30 odd years that it has been in existence in Europe, dealing with industrial generators and air compressors. In addition to our interest from a lubrication standpoint, we are also a big purchaser of power equipment of all types and are therefore interested in any developments which may offer new possibilities.

In discussing the free-piston engine with our people, they confirm my observations that the various design units have shown a common limitation which, I believe, has in large part restricted the widespread adoption of this type of power.

In short, high piston-ring wear, scuffing, and ring-belt and port deposits appear to be a prime limitation leading to short overhaul life by comparison with conventional engines. I am, of course, referring to industrial use where an engine is expected to run for thousands of hours before overhaul. This limitation can undoubtedly be raised a long way through metallurgy and advanced piston and ring design. I would like to make a plea also that more thought be given to lubrication, that is, the type of lubricant and the way it is introduced to the power cylinders.

For example, in one design the oil is fed to the power cylinders by force-feed lubricators in such a way that the cylinder gets one squirt of oil for every 100 strokes of the piston. Furthermore, the oil has traveled through lines 3 to 4 ft long before reaching the cylinders, and these lines are in the air box where the temperatures are in the order of 400 F. A considerable number of these units are in commercial service, but I am strongly of the opinion that the relatively short overhaul life obtained could be greatly improved by altering this lubrication system.

It may be that even with a revised system, further help will be needed from the lubricant itself. Certainly with the advances made in the additive field, considerable help can be expected. There are other items which could be raised in connection with lubrication of the existing engines, but I believe this brief example illustrates the main point that lubrication may well be the key toward providing desired operating life.

With this thought in mind, I would like to ask Mr. Underwood the following questions:

1. Could he describe the manner in which lubricant is applied to the cylinder wall, as well as the system feeding the piston cooling oil and compressor cylinders?

2. It seems obvious that the ring-loading conditions are severe and that temperatures both of the liner and ring belt are high. Could he give us some information as to the pressures and temperatures which exist in the Hyprex engine by comparison to conventional diesel and automotive engines?

Concluding, I think that the free-piston engine appears to offer

definite possibilities in various industrial and marine applications, and based on Mr. Underwood's presentation it seems that there are attractive possibilities also for automotive use.

Data Indicate Further Development Of Hyprex Engine Still Needed

—H. V. Nutt
USN Engineering Experiment Station

THE Hyprex free-piston engine consists essentially of a gas producer, a gas turbine or motor, and a hydraulic torque converter. For other than vehicular drive it is possible to eliminate the torque converter. For purposes where mechanical energy is not ultimately desired, the gas turbine may be eliminated. Thus, where compressed gas is the ultimate requirement, as in a jet aircraft, the gas producer alone might be the complete powerplant. For my purposes, however, I shall consider that the powerplant includes both the gas producer and the gas turbine. Thus we have combined in a unit powerplant, a reciprocating-type air compressor, a reciprocating-type heat engine, and a rotary-type gas turbine. The heat engine converts the chemical energy of the fuel into reciprocating mechanical motion of pistons. These pistons in turn produce compressed air or pneumatic energy which is combined with the excess compressed gas from combustion of the fuel to produce the working fluid for the gas turbine.

This, of course, is not the first use of either the free-piston gas producer or the rotary gas turbine. Probably the first use of the expansive action of gases to operate a revolving wheel for the generation of power dates back many centuries. In fact, the windmill is a gas turbine since the pressure of the moving air is due to the thermals produced in the atmosphere. I would like to quote from a paper presented by R. M. Neilson, given before the British Institute of Mechanical Engineers over half a century ago in October, 1904. Mr. Neilson said at that time, "A prophecy expressed frequently in engineering circles at the present day is that turbines actuated by hot gases other than steam will eventually come to the front as prime movers. There is no doubt that many persons speak of the advantages of gas turbines without duly considering the difficulties to be encountered. Temperature of metal parts is a problem seriously limiting performance and requiring great ingenuity to overcome. If a satisfactory turbine were obtainable there is no doubt that motor car builders would eagerly buy it and install it on their cars even if the cost were greater and the efficiency less." At about the same time, shortly after the turn of the century, Dugald Clerk, the great pioneer in internal-combustion engine design, stated that he saw some future in the gas turbine in utilizing the exhaust gases from reciprocating engines. It was along about the same time that a great controversy arose in heat-power circles as to the relative advantages of using reciprocating compressors versus the use of rotary compressors in conjunction with gas turbines. I have sought diligently to find in the literature any indication that reciprocating gas compressors can be produced to compete in efficiency with rotary compressors. I have not found it. It would be interesting if Mr. Underwood would indicate what efficiency is obtainable with his reciprocating compressor.

Mr. Underwood has raised the question of how best to report the results of tests on free-piston gasifier-type powerplants. He further makes use of the term gas horsepower. He then mentions performance and economy in terms of pounds of fuel per gas horsepower-hour output. I do not disagree with this method of presenting performance for purposes of studying, analyzing, and developing various components of the overall powerplant. Neither do I disagree with the method of reporting if the ultimate requirement is compressed air, as in an air compressor or a jet engine. However, when the ultimate requirement is either mechanical energy as produced by a rotating turbine shaft or electrical energy, then it seems more fitting and understandable to present economy data in terms of pounds or gallons of fuel per shaft horsepower-hour or kilowatt-hour, respectively. It would have been very interesting if the author had presented actual test data showing the performance of his gasifier-turbine power plant in terms of fuel consumption per output shaft horsepower-hour. I have seen little data of this type. The data I have seen have come entirely from our own experiments on gasifier turbine units at the Naval Engineering Experiment Station. These data indicate that considerable development of this type of powerplant is still required in order to allow it to perform as fondly hoped by its proponents. Although there are logical reasons for the belief that the free-piston gasifier-turbine combination can perform in a competitive manner with diesel engines it would appear that the turbo-supercharged diesel engine is now a practical piece of equipment which produces energy with an economy which the gasifier may never surpass.

Mr. Underwood has very ably discussed the many advantages of a free-piston gasifier. I would like to discuss one claimed advantage from a slightly different viewpoint. I refer to the claim that marginal-type fuels can be burned satisfactorily. Saying it another way, the gasifier has been claimed to have the advantage of being noncritical to a wide range of fuel properties. Such a characteristic on the part of a heat engine is a very definite military advantage. The large amounts of excess air used in the free-piston gasifier probably are the explanation for its ability to ignite and burn cleanly and completely those fuels having high viscosity, high boiling points, high carbon-to-hydrogen ratios, and relatively high ignition temperatures. The ability to carry on a war in a successful manner today is believed to depend greatly on the industrial potential of a nation. If a nation develops suitable munitions of war and then puts the designs on the shelf ready for use if an emergency occurs, it will find itself with an almost insurmountable time handicap when an emergency occurs as compared to that nation which not only designs, but actively develops the manufacturing means and continuously produces the product during peacetime. This latter nation has reduced its mobilization problems enormously by being able to continue producing at a high rate its normal peacetime equipment which is immediately utilized as a satisfactory war munition. I am highly gratified, therefore, that the General Motors Corp. and certain other companies in this country are actively exploiting a prime mover such as the free-piston gasifier which has the excellent military performance characteristic of insensitivity to fuel quality.

The USN has for many years been testing free-piston engines. The first such engine in the form of a free-piston air compressor was brought to the USN Engineering Experiment Station in 1946 from Germany. This engine was tested, and the data obtained were later published for the information of American industry. Later a similar piece of equipment produced entirely in America by an American company was evaluated. Still later the Navy obtained on contract a gasifier-turbine powerplant which was tested at the Experiment Station in Annapolis. As a result of this work, a new gasifier-turbine unit is now being constructed. It is also gratifying to know that the General Motors Corp. is building for the Maritime Commission a free-piston gasifier-turbine powerplant for installation and trial in a World War II noncombat-type cargo vessel. These projects indicate the high hopes the Navy has for the successful performance of a free-piston gasifier. As Mr. Underwood has indicated, not all of the problems have been overcome. We will be working for some years "to iron out the bugs." Although Mr. Underwood has expressed an optimistic view with regard to starting, we recognize that the units do not start 100% of the time. Mr. Underwood indicated that he does not obtain a record of cylinder pressures during his test development work. He probably knows, however, that maximum cylinder pressures rise to as much as 2000 lb per sq in. during full-load operation in some gasifiers. Such high pressures undoubtedly are the reasons for the difficult sealing problem in the piston-ring area. It is probable that future designs will deal more effectively with the sealing of these high-pressure gases than has been done to date.

The use of a power augmentation device in connection with the free-piston gasifier was mentioned by Mr. Underwood. I believe this is a very worth-while suggestion and should be actively studied by all people interested in making use of the free-piston gasifier. In naval circles, surveys have indicated that many ships use half of their total installed horsepower only about 2% of the total time. If this 50% little-used horsepower is produced with heavy long-life efficient and reliable equipment of the nature used to supply the base-load horsepower, then each such horsepower developed is obtained at extremely high first cost. On the other hand, many powerplants can withstand much higher loads for short bursts or intervals of time. The use of an afterburner or continuous flow combuster in the exhaust stream of a gasifier-

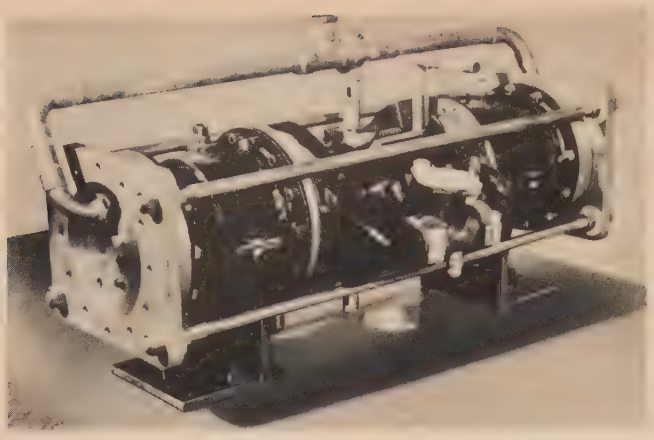

Fig. A—Experimental free-piston gasifier under 50 hp

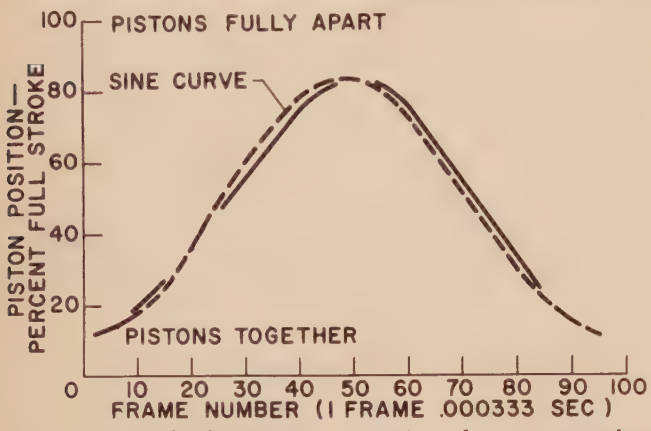

Fig. B—Piston displacement versus time for one complete cycle of operation

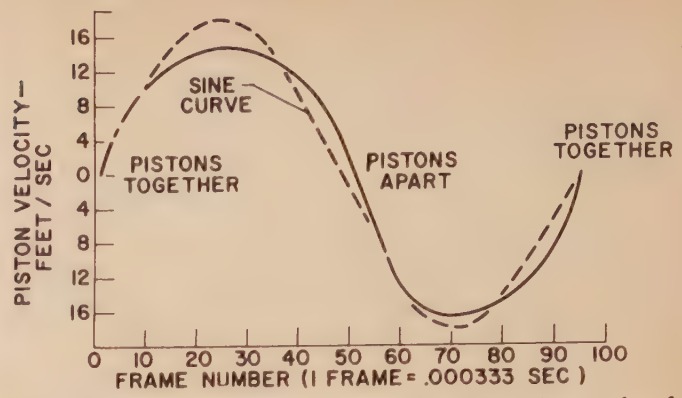

Fig. C—Piston velocity versus time for one complete cycle of operation

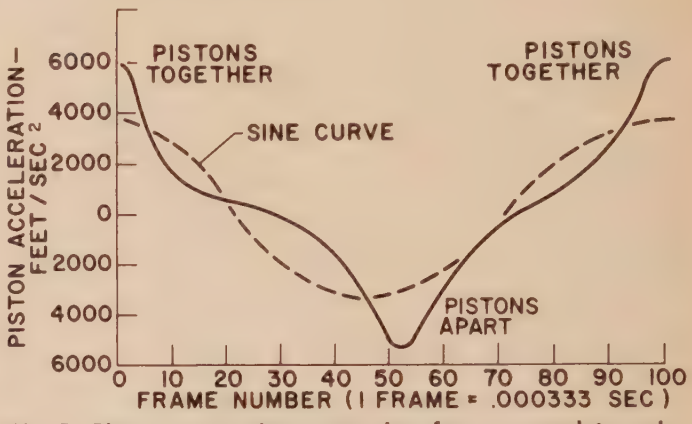

Fig. D—Piston acceleration versus time for one complete cycle of operation

turbine combination or a turbocharged diesel engine is one which should be studied intensively. It offers a real advantage to vehicular drive powerplants whether they are located on ships, in locomotives, or in road vehicles.

Sees Advantage in Centralizing Heat Power

—A. H. Glasenapp
International Harvester Co.

THIS engine or gasifier can be likened to a set of store teeth, in that it masticates what the turbine digests. Like mechanical teeth, its apparent simplicity may get rather complicated particularly with regard to controls. Its comparative lack of moving parts may not be synonymous with endurance. Its probable low cost may not stay so low as frequencies of speeds increase and temperatures rise in order to get more chewing power in a smaller mouth and make for more complex construction and the use of rarer materials. Its service and maintenance are probably very simple if handled as a unit burner and washed and sandpapered under the faucet. However, complications may arise if the servicing is of necessity carried out in the chassis of the vehicle (the removal of raspberry seeds gets to be quite a problem if carried out in the mouth of the victim).

Nevertheless, on the surface, it appears as though here is a pressure cooker which is, at the moment, the most practical link between the pure reciprocating and the pure rotating types.

To me, the biggest advantage of the engine is the centralization of heat power which can be furnished to a multiplicity of turbines at semiremote locations for turning the wheels. It's the difference between your central heating and air conditioning plant and the old pot-bellied stove in the living room.

We've all seen artists' conceptions, or had our own dreams of machines of the future. Let us imagine, for the moment, a *super* super highway constructing machine which begins on one coast and proceeds, at a fairly rapid pace to other; all the while chewing through the hills, depositing in the valleys, and making and laying a rapidly usable roadway as it goes. This machine will be somewhat of a behemoth with many wheels that turn; the configuration can be as infinite as the number of imaginative brain cells; but can you imagine any better way to power this monster than with a central power-maker masticating the food (perhaps atomic) and feeding it to its fledglings which are in turn doing their own individual job?

Studies Operation of Free-Piston Machinery

—P. Klotsch and G. H. Millar
Ford Motor Co.

THE free-piston turbocompound engine has held the interest and fascination of engineers for almost three decades. Its lack of popularity as an automotive propulsion plant in all probability is due to the complicated thermodynamic and dynamic characteristics brought on by the obvious advantage of eliminating the crankshaft.

For some time a free-piston turbocompound engine development program has been in existence in the scientific laboratory of the Ford Motor Co. The objective of this program has been to take a good look at the possible advantages offered by the free-piston turbocompound engine.

To implement this process, several experimental free-piston gasifiers have been designed and built. These machines are relatively small in size and operate at frequencies up to 3600 cycles per min. Fig. A is a photograph of one experimental machine which is under 50 hp.

Laboratory work with these machines has been directed to-

wards obtaining a better understanding of the operation of free-piston machinery above the usual 1000-cycle per min range. It is further desired to evaluate on a laboratory basis the piston motion-combustion process relationship under as many and as varied conditions of operation as can be achieved.

The results of this program have been surprising and significantly interesting. It is advantageous to itemize the results in the following broad categories:

1. Mechanical design.
2. Thermodynamic performance.
3. Stability and control.

With regard to mechanical design it has been apparent from the results of engine tests that one pair of synchronizing rods is entirely adequate to keep the pistons in phase. The second set of rod are excess baggage, and the attendant difficulties with alignment and fastening do not justify their existence.

With reference to Fig. 6 in Mr. Underwood's paper, motion studies by means of high-speed photography have revealed piston-motion characteristics unlike any known engines. Fig. B is a displacement-time curve obtained from an engine running at 1900 cycles per min. On the same grapl is plotted a sine curve for comparison. Although the displacement curve is not appreciably different from a sine curve, Fig. C shows what happens when this is converted to a velocity-time curve, and further, Fig. D shows the marked difference in accelerations. Acceleration is the important factor in keeping free-piston machines in one piece.

During the course of operation the effect of the combustion process on the piston motion, and the converse, has been studied. There is no question but what the variable stroke and lack of a fixed volume-time path are important influences on combustion in free-piston machinery.

Gasifiers have been operated in the scientific laboratory with both gaseous and liquid fuel on both spark ignition and compression ignition at practically all conceivable air/fuel ratios. Carbureted- and injection-type fuel systems have been used interchangeably. It has been found possible to run at low power levels on spark ignition and as the pressure ratio is increased to change automatically to compression ignition. This change is operation on gasoline-type fuels is practically undetectable except on a pressure-time card.

Propane has been also used as a fuel with spark ignition and direct solid fuel injection was operated both with and without spark ignition. The completely different response of these experimental gasifiers to the normal parameters of fuel quality remains a major research effort.

Control of the free-piston gasifier requires a basic knowledge of many parameters. Under some conditions of operation instantaneous starting and stopping at will has been achieved. Under some circumstances starting and stopping has not been so successful.

Complete control of the gasifier cannot be achieved by manipulation of the fuel alone. The fuel must be regulated along with other parameters to achieve stable and smooth control.

Suggests Semifree-Piston Engine Has Greater Advantages

—P. H. Schweitzer
Pennsylvania State University

THE compactness of the Hyprex engine is truly remarkable and was accomplished through a great deal of ingenuity in the siamese gasifier and the pneumatic dephaser. On the latter, more information would be welcome.

Even if we concede all the advantages of the free-piston engine enumerated by the author, we should not close our eyes to its several disadvantages, as:

1. Absence of a rotating shaft makes the drive of the injection pump and, to a lesser extent, that of the water pump, cooling fan, lube oil pump, and the like awkward.

2. If the control fails because of loss of compression or some other reason, the engine may stall.

3. The starting is awkward as the engine must start at the first stroke. It requires a supply of vacuum, of compressed air, and a thorough venting of the fuel lines before starting. If we insist on pushbutton starting, that pushbutton will have a lot of work to do. One job is venting of the fuel lines. When an engine is not running, the fuel lines become partially airbound. In a conventional engine this small amount of air is usually driven out during 10 or 20 revolutions of cranking. If a free-piston-engine injection system is airbound, the engine simply will not start.

4. The oil cooling of the free pistons is difficult, and the pistons are prone to overheating.

5. Unsymmetrical porting, with a desirable exhaust lead on one hand and inlet ports closure after the exhaust ports on the other hand is, with 180-deg phase angle between the opposed pistons, impossible.

6. The size of air cleaner needed by an engine like the Hyprex is enormous. According to the author the engine runs with 400%

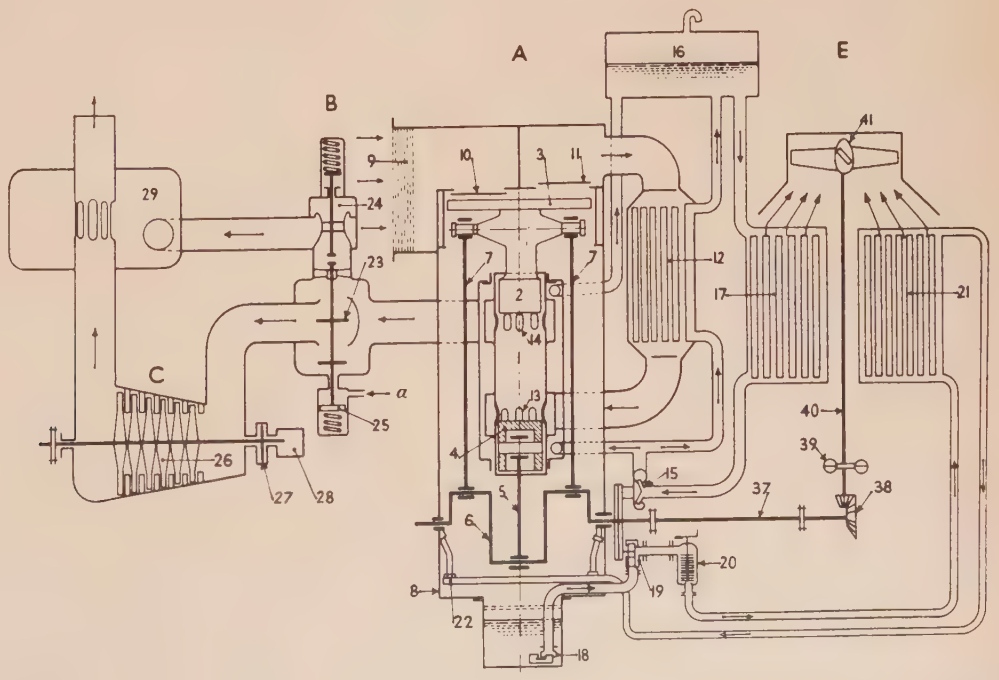

Fig. E—Götaverken compound powerplant for locomotives

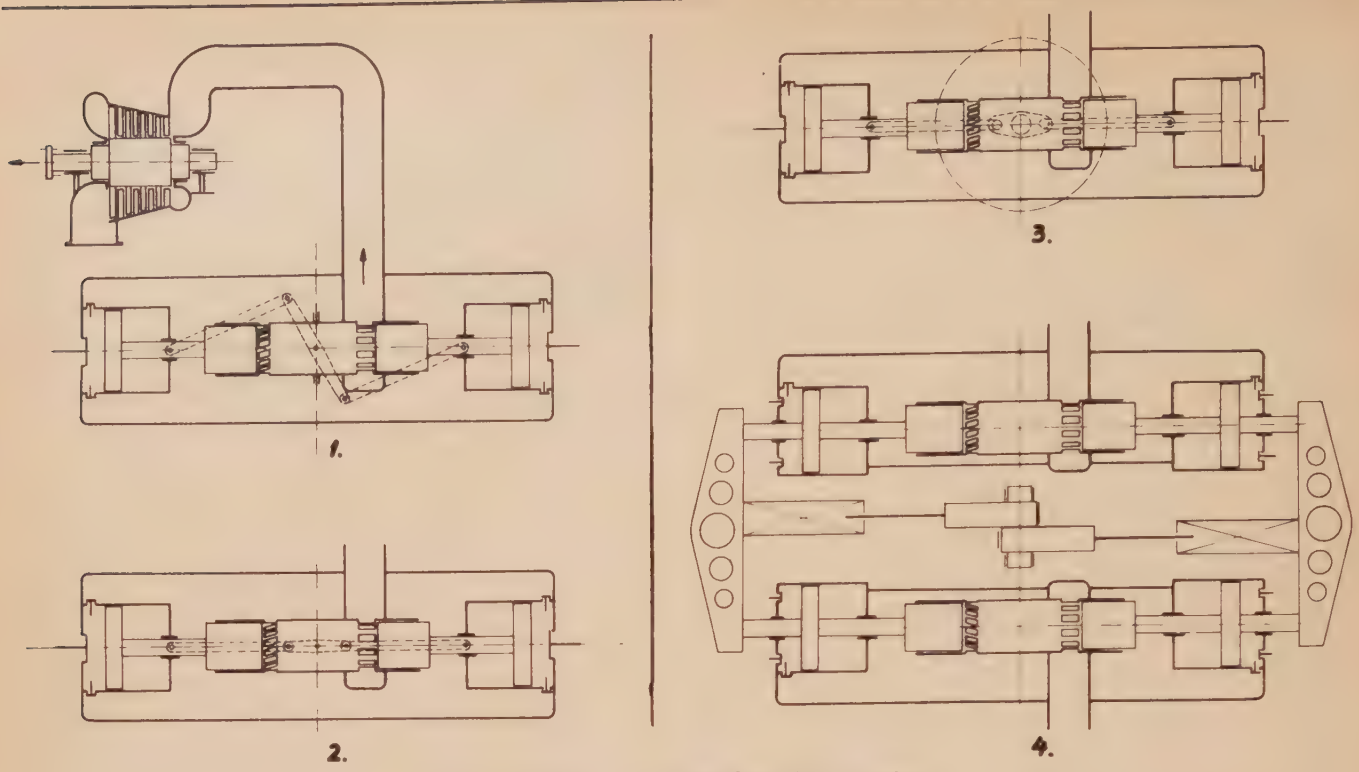

Fig. F—Evolution of the semifree-piston engine

excess air. Because of this it would require an air cleaner of 5 times the capacity of that of a gasoline engine with equal fuel consumption, or say 2½ times with equal horsepower. However, a 200-hp gasoline passenger-car engine uses less than 10% of its power in average (15–20 hp at 60 mph). The air consumption of a carburetor engine is roughly proportional to its horsepower output. The air consumption of a conventinal diesel engine is independent of the load. In the Hprex with "recirculation," it may decrease slightly with the load, but surely not to the same extent as that of a carburetor engine. So the air-cleaner capacity required by a Hyprex engine would be many times larger than that of a conventional gasoline engine if we want the same replacement period, because of the greatly increased hourly air consumption of the free-piston engine.

A straight turbine engine may get along without any air cleaner but a free-piston engine needs, if anything, cleaner air than the gasoline engine. We not only have rubbing surfaces under severe conditions, but also reed valves which lose their seal when dust gets on their seats.

Some of the advantages of the free-piston powerplant without some of its disadvantages are being achieved by the gasifier turbine system used in a locomotive by Götaverken in Sweden. Fig. E shows the arrangement which consists of an opposed piston gasifier with a crankshaft, A; a change valve to bypass the turbine during starting, B; a power turbine, C; and water cooler, E. According to information, 27,000 miles have been accumulated with locomotive in experimental service. The powerplant delivers 1300 gas hp, and its specific fuel consumption is given as 0.43 lb per hp-hr. The torque curve appears satisfactory.

More of the free-piston advantages with less of its disadvantages can be achieved by a "semifree-piston engine." The principle of the semi-free engine can best be understood from the evolution diagrams shown on Fig. F. Diagram 1 shows a conventional free-piston powerplant with a synchronizing linkage. By shortening the center link the stroke can be limited as in Diagram 2. In Diagram 3 a flywheel is added to the linkage and the center link which oscillated before now rotates. In Diagram 4 we have two cylinders in tandem, the pistons of which are connected by a rigid yoke to each of which a crosshead is attached. The crossheads are connected to cranks and the mechanism can be considered either as a crank mechanism or a synchronizing linkage, depending on the size of the flywheel attached to the shaft.

The semifree-piston engine shares with the full free-piston engine the following advantages:

1. Simplicity with few moving parts.
2. Direct transmission of forces from the power pistons to the compressor pistons without rotating or oscillating components between the two.
3. No side thrust on the pistons.
4. Perfect balance and absence of vibrations.
5. Small space.
6. High thermal efficiency.

It is free, however, from five of the six disadvantages listed for the free-piston engine and even the sixth (air-cleaner size) is mitigated. The oil cooling of the piston is facilitated by the presence of a piston rod. With proper piston cooling 400% excess air is no longer necessary. If we can get along with 150% excess air, the size of the air cleaner can be reduced to one half.

The apparent disadvantage of the semifree-piston machine is the continued use of crankshaft and connecting rods. These parts, however, are lightly stressed because they do not transmit much force. All bearings are away from the engine at a cool and accessible place where they are easy to lubricate.

Fig. G shows the schematic arrangement for a semi-free-piston engine compressor. If all the air delivery of the compressor is ducted to the power cylinders, the machine can serve as a gas generator to the turbine.

ORAL DISCUSSION

Reported by R. B. Bicknell

General Motors Corp.

J. Thorpe, Allegheny Ludlum Steel Corp.: If an afterburner were used in the system, would there be any problem with materials?

Mr. Underwood: This depends upon the exhaust-gas temperature allowed. If the temperature of the gases in the afterburner section were allowed to go as high as 1500 F, then high-temperature alloy steels would be necessary.

G. Grubeck, American Machine and Foundry Co.: Is cutting out cylinders a practical answer for part-load operation?

Mr. Underwood: Yes.

Mr. Grubeck: How are the pistons pulled apart prior to starting?

Mr. Underwood: This is done by pulling a vacuum in the bounce chamber.

Authors' Closure To Discussion

IN answer to Mr. Habermann's question regarding the type of lubrication, I can agree with him that the original method of lubrication can be improved. As he points out, the lubrication on the first SIGMA engines was rather intermittent. However, all of the advanced designs introduce a smaller total quantity of oil and, at the same time, feed it more frequently to the cylinder liners. Our tests have shown considerable improvement in cylinder and ring wear. The compressor cylinders are a cause of no concern, as demonstrated by the fact that our original compressor cylinders and pistons have now run 10,000 hr and are still in good running condition. The pressures in our free-piston engines run as high as 1800 psi, which are obviously considerably higher than in a conventional diesel or automotive engine. We do not know the combustion temperatures. It is true that many investigators have had difficulty with piston rings, and until rather recently this has continued to be true. However, during the past year our developments have made considerable gains in this direction, and our latest tests demonstrate that piston rings can be made to operate for many thousands of hours.

Mr. Nutt asked the question regarding the efficiency of the reciprocating compressor. We have not made a complete enough study of this part of the gasifier to have an answer to his question. Data on the performance in terms of fuel consumption per shaft horsepower hour or kilowatt-hour were not included because this has been thoroughly covered in papers by the SIGMA and SEME organizations. As an example one could refer to "Développement récent des groupes à générateurs à pistons libres et turbines à gaz," by D. Coste and R. Huber, published in the October, 1956, issue of "L'Industrie du Pétrole." Reporting on the Cherbourg installation, the figure of 0.54 lb of fuel per kilowatt-hour at the generator terminals is given.

The comments by Dr. Schweitzer on the semifree-piston engine are interesting, but it is difficult to see how the engine which he proposes can be termed a type of free-piston engine. The phrase "free piston" signifies that the piston stroke is variable depending upon the operating conditions of the engine. If the piston stroke is crank-restricted, then the engine is not a free-piston engine, nor a semifree-piston engine.

Dr. Schweitzer's proposal appears to be a different configuration of an ordinary diesel using a piston air compressor, except that it is used as a gasifier for a compound engine.

The absence of a rotating shaft does complicate the drive of pumps, fans, and other accessories. It does not necessarily complicate the injection pump drive.

We are not familiar with Dr. Schweitzer's operating experience with free-piston engines, but our experience indicates that several disadvantages which he enumerates are not troublesome. When a free-piston engine is in satisfactory condition, it is easy to start and continues running without interruption. On the other side of the picture, when a diesel engine is not in good operating condition, it can be difficult to start and to keep running. We do not understand why the pistons of the semifree-piston engine are less difficult to cool. Seals are needed, and it would appear to require more complication than the usual design.

If less air were used in order to employ a smaller air cleaner, then the exhaust temperature would be higher. An increase of even as little as 200 F would result in turbine problems which we do not feel are worth the advantage of a somewhat smaller air cleaner.

The statement is made that the crankshaft and connecting rods do not transmit much force. If the "semifree-piston engine" is designed to store and feed back enough energy to keep the engine running for several cycles when there might be a temporary loss of power, then it would appear that the connecting rods and crankshaft would have to be rather substantial. It should be remembered that the starting forces can be substantial and, if the engine is to be started through the linkage, it is not lightly stressed.

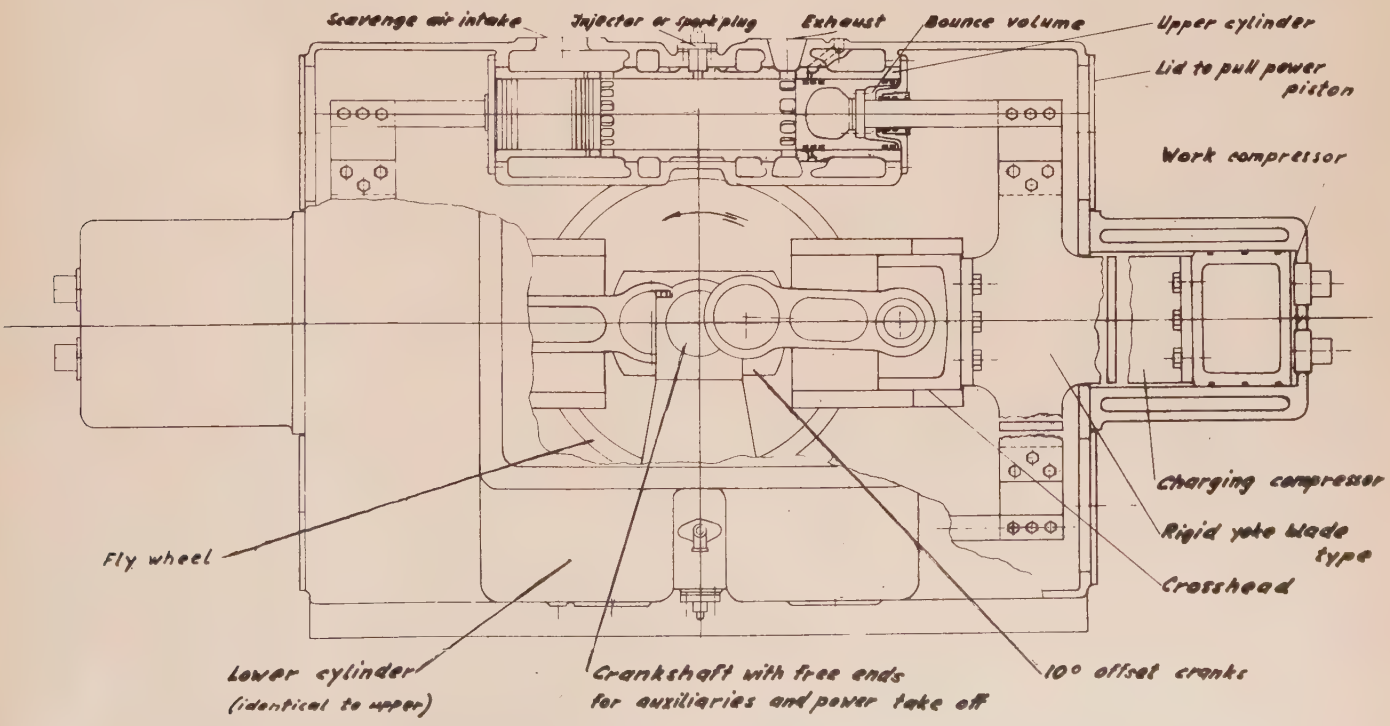

Fig. G—Schematic arrangement for a semifree-piston-engine compressor

WHAT DO CALCULATED GEAR

THIS paper will indicate the major considerations included in a modern equation for calculating the stress and for rating the strength capacity of gear teeth. Particular emphasis will be placed upon the contribution of "tests" towards the development of a formula.

THIS paper utilizes the proposed formula for "the strength of gear teeth" developed by the Gear Rating Coordinating Committee of the American Gear Manufacturers Association to answer the question "What do calculated gear stresses mean?"

It is shown that meaningful gear-stress calculations use a formula developed upon a good basic theoretical foundation and used within the limitations of validity. The basic formula must be adjusted by measurable modifying factors evaluating load, accuracy, and materials. These modifying factors depend upon data secured by field experience and laboratory tests.

It can be concluded that gear testing and theoretical analysis are complementary functions necessary for accurately calculating the stress in gear teeth.

What do calculated gear stresses mean? This question is often asked particularly by engineers concerned with small mass-produced transmissions which can be easily tested in dynamometer stands. In answering, the discussion centers on "basic theory," "assumptions," "boundary of knowledge," "validity of application," or similar factors. Those of us associated with gear trains which can carry tens of thousands of horsepower or millions of pound-feet of torque cannot readily make simple laboratory load tests. When we do, we naturally ask *"What do laboratory gear tests mean?"* The answer usually involves discussions of "ideal environment," "perfect mounting," "controlled loads," and similar modifying notions.

A calculation assumes the existence of a formula. A formula presumes a theoretical basis whether this be a complex mathematical analysis or the "best" fit of a line to plotted test results. In either case the meaning of the answer obtained by computing the formula must be "judged" upon the basic theory or test, the boundary of knowledge associated with the theory or test, and the validity of its application to the particular problem. Secondly, the data used for computation must be checked to determine whether the dimensions or specifications used are precisely those which will be obtained and maintained in actual service. Further, good engineering practice demands that

STRESSES MEAN?

E. J. Wellauer, Falk Corp.

This paper was presented at the SAE National Tractor Meeting, Milwaukee, Sept. 12, 1956.

the computed answer be compared with service experience and available test results. Finally, after serious consideration of all these items, the degree of risk which must be assumed for either commercial or technical reasons is determined and a proper factor of safety applied.

Gear tests, either in the laboratory or in the field, furnish substantiating data, and when made the chief criterion they must be considered valid only within the limited range of the test variables. It is therefore suggested that perhaps both calculations and testing are not in themselves final evaluations but rather complementary. It is believed that this concept can be illustrated best by the general equation for gear-tooth strength proposed by the Gear Rating Coordinating Committee of the American Gear Manufacturers Association. The basic background for the development of the formula is recorded in the symposium by Wellauer, Dudley, and Coleman.[1] This recently proposed tentative practice for the coordinated rating for the strength of gear teeth is readily adapted to specific rating of spur, helical, herringbone, straight bevel, spiral bevel, and hypoid gears. A historical and derivation study of the factors used in the Proposed Tentative AGMA Standard Rating (225.01) for the "strength of gear teeth" will indicate the complementary function of theoretical analysis and testing evaluation required to impart true meaning to calculated gear stresses.

Gear Tooth Stress Formula—Historical

W. Lewis [2] introduced, over 60 years ago, one of first rational methods of computing the tooth bending stress and load carrying capacity of gearing. Numerous investigators since that time have made definite improvements for more closely evaluating gear-tooth stress and strength capacity. Now high speeds, light weights, immense powers, and economical considerations demand accelerated developments to determine more precisely the strength capacity of gears under load.

The original Lewis formula was quite simply stated as:

$$W = SFpy \qquad (1)$$

where:

S = Bending stress
F = Face width
p = Circular pitch
y = Lewis factor

The derivation of this formula was based upon

1 *AGMA*, 1956: "Coordinated Rating for Strength of Gear Teeth," by E. J. Wellauer, D. W. Dudley, and W. Coleman.
2 *Proceedings of Engineers Club*, Vol. 10, 1893, p. 16: "Investigation of Strength of Gear Teeth," by W. Lewis.

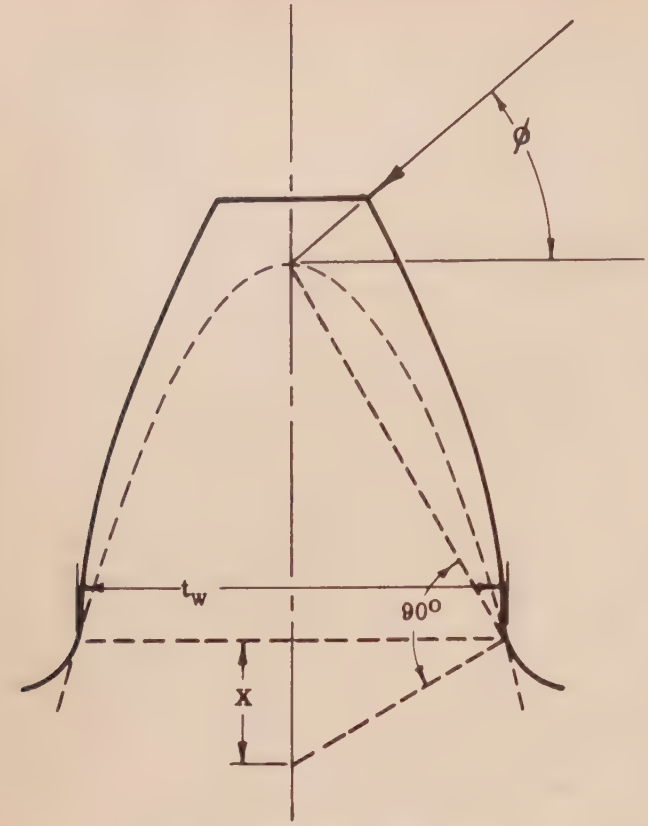

Fig. 1—Lewis form factor layout

Table 1—Gear Test Variations

Dimension	
Center Distance	4.688 to 13.455 in.
Gross Face Width	2.25 to 6.25 in.
Pitch Diameter, Pinion	1.205 to 6.679 in.
Pitch Diameter, Gear	8.17 to 23.956 in.
Gear Data	
Type of Gears	Single and double helical
Helical Angle	8 to 40 deg
Pressure Angle	19 to 37 deg transverse
Operation Details, rpm	1500 to 5000 (high-speed tester 32,000 rpm max.)
Pitch Line Velocities	350 to 8750 fpm (high-speed tester 25,000 fpm)
Tooth Contact Cycles (pinion)	147,000 to 240,000,000
Tooth Loadings	
Pressure per in. of Face, P/F	612 to 3790
Pressure per in. of Face per in. of Pitch Diameter.	217 to 1873
K Factor	244 to 2250
Materials	
Steel	160 Brinell to carburized 60 R_C
Bronze	
Aluminum	

a cantilever beam and a load applied as shown in Fig. 1. Lewis assumed that the toothshaped beam could be simulated by a parabola which was tangent to the tooth profile at the most highly stressed section. A parabola is a beam of "constant stress" when loaded with a tangential force applied at the vertex. Such a parabola has the same stress in the outer fibers at all the transverse sections. Actually, the gear-tooth-shaped beam is stressed less than the parabola except at the point of tangency where the stresses are theoretically identical.

The stress in a cantilever beam is given by the formula:

$$S_b = \frac{6Wh}{Ft^2} \tag{2}$$

where:

S_b = Bending stress
W = Tangential component of load applied at vertex
F = Face width
h = Height of vertex above cantilever base
t = Thickness at cantilever base

As shown in Fig. 1, Lewis drew a line from the vertex to the point of tangency and from this point constructed a second line at right angles so as to intersect the center line. Then:

$$X = \frac{t^2}{4h} \tag{3}$$

Using p = circular pitch, Lewis developed a factor:

$$y = 2/3\,\frac{X}{p}$$

$$y = \frac{t^2}{6ph} \tag{4}$$

Hence, by manipulation:

$$W = SFpy$$

which is the Lewis equation.

Modifications of Lewis Equation

Through the years, designers have added modifications to the fundamental Lewis bending stress equation as required by increasing speeds, higher loadings, increased strength of materials, improved tooth accuracies, and better developed tooth shapes. Some of the more important introductions have been the use of factors compensating for velocity, load sharing, stress concentration, and allowable stresses varying with the strength of the material. Many investigators [3-10] have suggested modifications to evaluate more accurately the stress system existing in loaded gear teeth. Some modifications involve the selection of a different point of load application on the profile in order to evaluate more precisely the condition of maximum loading. Others change the direction or angle of load application using working pressure angles and the worst position of load sharing during

[3] Carl Barth.
[4] Wissman, "Doctoral Dissertation." Presented at Berlin Technical College, 1930. See also footnote 8.
[5] *Engineer*, July 8, 1932: "Gear Performance II," by H. E. Merritt.
[6] *Automotive Industries*, Oct. 15, 1939, pp. 56–62: "Suggested Modifications in Gear-Teeth Formula," by P. M. Heldt.
[7] *AGMA*, 1941: "Calculated Bending Stress in Gear Teeth," by A. Candee.
[8] *AGMA*, 1944: "Combined Static Stresses in Teeth of Spur Gears," by M. Maletz.
[9] *AGMA*, 1922: "Gleason Works System of Bevel Gears," by F. E. McMullen and T. M. Durkan.
[10] *Institute of Mechanical Engineers*, Vol. 159, 1948, pp. 384–391: "Tensile Fillet Stresses in Load Projections," by R. B. Heywood.

the contact cycle. Others recognize that tangential, radial, and frictional components of the transmitted load exist, and attempt to compensate for their effect.

The compressive stress developed by the radial component of the load has been used to modify the tensile bending stress. Carried to extremes, when the load is applied very low on the profile, the compressive stress developed by the radial component can be greater than the bending moment tensile stress. Hence, the resultant stress can be in compression, and failure could not develop. Many of these uncertainties have been more clearly evaluated by photoelastic and laboratory load tests.

Schmitter [11,12] extended the analysis for application to the unique oblique line of contact of helical gears, and was also one of the first to recognize the point of maximum loading for spur gears.

AGMA Formula for Gear-Tooth Strength

The Gear Rating Coordinating Committee studied the effects of the tensile stress caused by the bending moment of the tangential load, the compressive stress caused by the radial load component, and the combined stress formed by the shear and the tensile stresses. These studies indicate that the bending tensile stress is the maximum for almost all practical combinations of gear teeth and, therefore, was adopted for the standard formula. Maximum loadings applied low on the tooth profile in conjunction with high-pressure angles need special design analysis and consideration of the combined shear and tensile stresses.

Since the parabolic beam concept of Lewis apparently is still the best available basis for computation, the proposed fundamental formula was derived accordingly to produce the equation:

$$s_t = \frac{W_t K_o}{K_v} \frac{P_d}{F} \frac{K_s K_m}{J} \tag{5}$$

$$W_a = s_a \frac{K_v}{K_o} \frac{F}{P_d} \frac{J}{K_s K_m} \frac{K_L}{K_T K_R} \tag{6}$$

Note that equation 5 is divided into three groups of terms, the first of which is concerned with the load, the second is concerned with the tooth size, and the third is concerned with the stress distribution.

Load:

W_a = Allowable tangential load
W_t = Transmitted tangential load
K_o = Overload factor
K_v = Dynamic factor

Tooth size:

P_d = Diametral pitch (Substitute P_{nd} for helicals. Use P_d at large end for bevels.)

F = Net face width

Stress Distribution:

K_s = Size factor
K_m = Load distribution
J = Geometry factor

The relation of calculated to allowable stress is:

$$s_t \leqq \frac{s_a K_L}{K_T K_R} \tag{7}$$

where:

s_a = Allowable stress
K_L = Life factor
K_T = Temperature factor
K_R = Factor of safety

The similarity between the allowable load equations for the Lewis and AGMA formulas is apparent. The terms S and s_a, F and F, P_d and $\frac{1}{p}$, and y and J have similar connotations.

$$W = SFpy \qquad \text{Lewis}$$

$$W_a = s_a \frac{K_v}{K_o} \frac{F}{P_d} \frac{J}{K_s K_m} \frac{K_L}{K_R K_T} \qquad \text{AGMA}$$

Why were an additional seven factors added? Simply because knowledge developed since the time of Lewis clearly indicated the need for these factors. Experience with gears which destroyed themselves within a few hours of operation due to loads developed by tooth inaccuracies or resonant vibrations showed the need for considering dynamic and elastic conditions. Errors caused by improper mounting or excessive deflection illustrated that load distribution must be evaluated. Sharp root fillets and premature failures emphatically demanded stress corrections due to stress concentration. The higher loads carried by very accurate spur gears demonstrated the potentials of load sharing between teeth. Coleman reports that

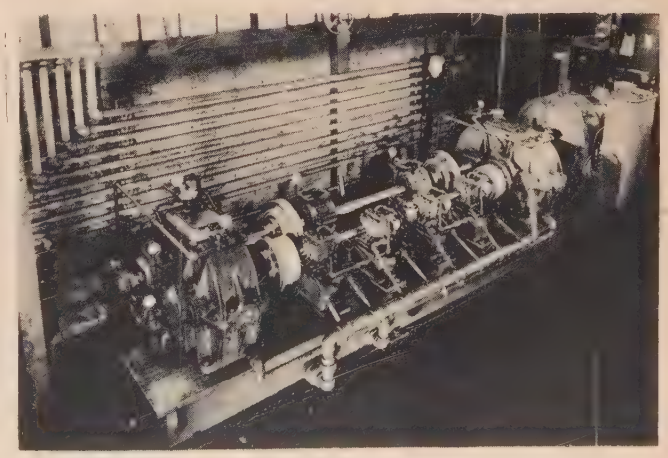

Fig. 2—Gear tester with 12.000-in. center x 7.5-in. face

11 *AGMA*, 1941: "Report of Investigation Conducted by Helical and Herringbone Gear Committee," by W. P. Schmitter.
12 *AGMA*, 1937: "Investigation of Contact Stresses at Region of Maximum Intensity," by W. P. Schmitter.

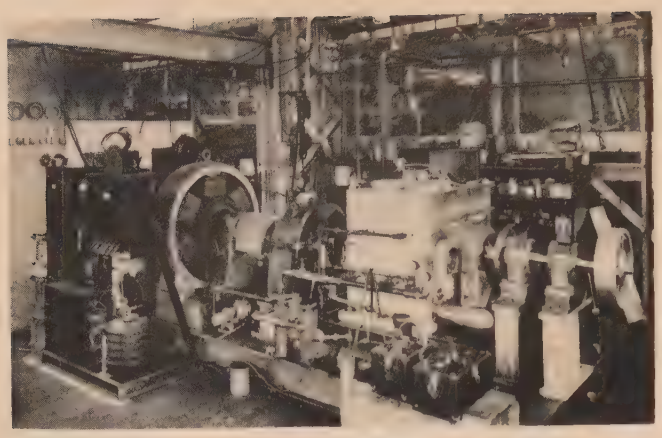

Fig. 3—Super high-speed gear tester with 13.455-in. center x 7.75-in. face

Table 2—Recommended Overload Factors—K_o

Character of Power Source	Character of Load on Driven Machine		
	Uniform	Moderate Shock	Heavy Shock
Uniform	1.00	1.25	1.75
Light Shock	1.25	1.50	2.00
Heavy Shock	1.50	1.75	2.25

Table 3—Typical Drive Combinations

Typical Examples of Power Source Character

Uniform	Moderate Shock	Heavy Shock
Electric Motor	Multicylinder Internal Combustion Engine	Single-Cylinder Internal Combustion Engine
Generator		

Typical Examples of Load Character

Uniform	Moderate Shock	Heavy Shock
Centrifugal Blower	Lobe-Type Blower	Ore Crushers
Pure Liquid Agitator	Liquid and Solid Agitator	Single-Cylinder Compressor
Belt-Conveyor-Uniform Feed	Belt-Conveyor-Nonuniform Feed	Reciprocating Conveyor

previous bevel gear formulas did not correlate with hypoid experience which led to improved analyses having better adaptability. All these problems usually demanded laboratory tests and field experiences to provide answers or at least fundamental modes of reaction to the major variables.

Many of these tests are outlined in the symposium by Wellauer, Coleman, and Dudley.[13] As an example, Figs. 2, 3, and 4 show gear testing units installed in the laboratories of the Falk Corp. The range of variables tested is shown in Table 1. Combining the results from tests such as these operated by others in the gear industry, pooling common field experiences, and using a basic fundamental theory enabled the AGMA Gear Rating Coordinating Committee to develop its proposed formula and general standard for the rating of gear teeth. The proposed recommended tentative practice for the strength rating of gear teeth is believed to include all the factors which are known to affect gear-tooth strength and which can be numerically evaluated. This fundamental formula is applicable for computing the strength of any type of gear tooth.

One main virtue of the formula is the flexibility which will enable the immediate incorporation of new or revised data as determined by field service or laboratory tests.

The remaining portion of the paper will outline briefly the reasons for the adoption of the various factors and the means by which numerical values were assigned. Attempts will be made when possible to provide the basic theory, the assumptions involved, the validity and restrictions of application as well as the need for additional or more accurate data. It is not intended to furnish precise information for design or analytical purposes, but rather to use the factor applicable to either spur, helical, or bevel gearing which most clearly illustrates the principle under discussion. The explanation is divided into factors associated with load, tooth size, stress distribution, and allowable stress.

Factors Associated With Load

Transmitted gear loads cannot be accurately evaluated due to the variable characteristics of driving and driven machinery. Some prime movers can develop very high momentary overload peaks, and in combination with certain driven machines applied to specific applications these overloads are experienced with more or less regularity. The magnitude and partially the frequency of application of the overload typical for certain applications of driver-driven machine combinations are measured by the overload factor (K_o). The overload factor compensates for the roughness of the operation and can be established only after considerable field experience.

Overload Factor (K_o)

Reliable field experience extending over 20 years is available in the "service factors" used for establishing the various AGMA durability ratings for gears. The various overload factors so derived are listed in Table 2.

Some idea of the character of typical drive combinations is shown in Table 3.

In the absence of definite field experience, comparative studies or actual horsepower or torquemeter readings can act as a guide. It is imperative that all meter readings be taken for a sufficient period to insure measurement of the worst operating condition. The only question which might arise from using overload factors established from experience with durability ratings is the fact that the contact stresses measured for durability rating vary as the square root of the load whereas for strength the critical stresses vary almost proportionately to the load. It is presently believed that the service factors established for durability rating are a measure of actual horsepower

[13] *AGMA*, 1954: "Fatigue Strength Characteristics of Gear Teeth—Symposium," by E. J. Wellauer, D. W. Dudley, and W. Coleman.

peak loading rather than a measure of protection against stress. Therefore, the service factors used for durability should be used as the overload factor for strength until more precise field experience is developed.

Besides the overload correction which is actually a part of the transmitted load, it must be recognized that the rotation of the gears imposes an additional load caused by dynamic effects. The increased load is measured by the dynamic factor.

Dynamic Factor (K_v)

The dynamic factor measures the influence of:

1. Tooth spacing and profile errors
2. Pitch line speed and rpm
3. Inertia of pinion and gear
4. Transmitted load per inch of face
5. Tooth stiffness

Attempting to compensate for these variations is known to be a complex procedure. Modern instrumentation including strain gages is helping to solve the problem. Fortunately, over 20 years of experience has been obtained with velocity factors used for calculating durability ratings for various speeds. Experience has seemed to prove that very accurate gears substantially mounted and connecting light masses are only slightly affected by increasing velocities. Even in these precision gears there is a constant dynamic factor of approximately 0.70 as proven by static single-tooth strength tests. In the proposed strength formula it might be well to consider that lower than maximum allowable stresses are used for the initial 0.70 velocity correction. For bevels the effect of velocity has not been measured, and the dynamic factor is based upon contact conditions and inertia. A high size factor might also partially correct for velocity.

Typical dynamic factors for various types of gearing are shown in Fig. 5. These curves serve as guides, and individual considerations of a particular application might require more drastic devaluations.

Factors Associated With Tooth Size

The factors concerned with tooth size are the diametral pitch (P_d) and the net face with (F). These are measurable factors, but it should not be construed that the strength capacity is directly proportionate.

For bevel gears the size factor is affected by the pitch, and for helical gears either the normal or transverse diametral pitch necessarily corrected for the helix angle might be used depending upon the convenience desired for computation. The net face width must be viewed in terms of contact distribution which varies both with accuracy and load. The compensating factors involved will be discussed in appropriate sections.

Factors Associated With Stress Distribution

The factors concerned with stress distribution are

K_s = Size factor
K_m = Load distribution factor
J = Geometry factor

Size Factor (K_s)—The size factor K_s reflects nonuniformity of material properities. It depends primarily upon:

1. Pitch of teeth.
2. Diameter of parts.
3. Ratio of tooth size to diameter of part.
4. Face width.
5. Area of stress pattern.
6. Ratio of case depth to pitch.
7. Hardenability and heat-treatment of materials.

That a size factor exists in the performance of materials seems evident from service experience. As a general conclusion, Horger [14] seems to indicate that

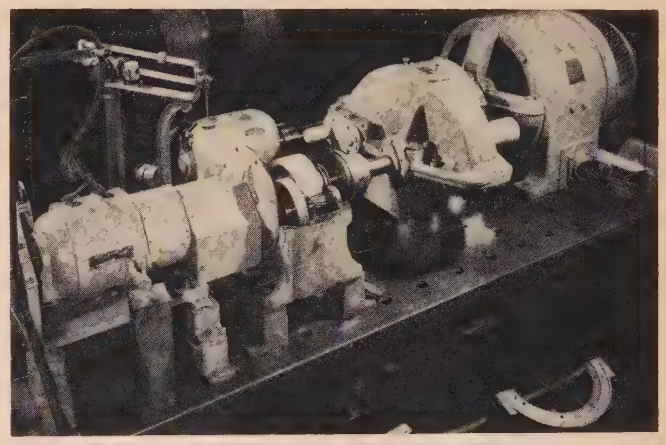

Fig. 4—Double reduction gear tester. First reduction: 4.688-in. center x 1.25-in. face; second reduction: 4.688-in. center x 3.125-in. face

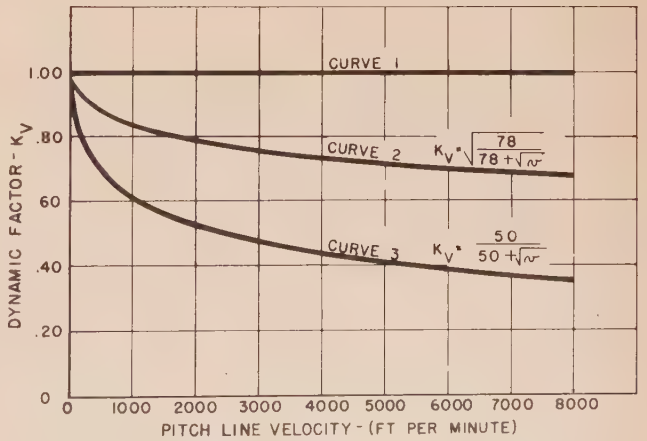

Fig. 5—Dynamic load factors for various types of gears

14 "Fatigue" by O. J. Horger, B. J. Lazan, and T. J. Dolan. Pub. by ASM, Cleveland, 1954.

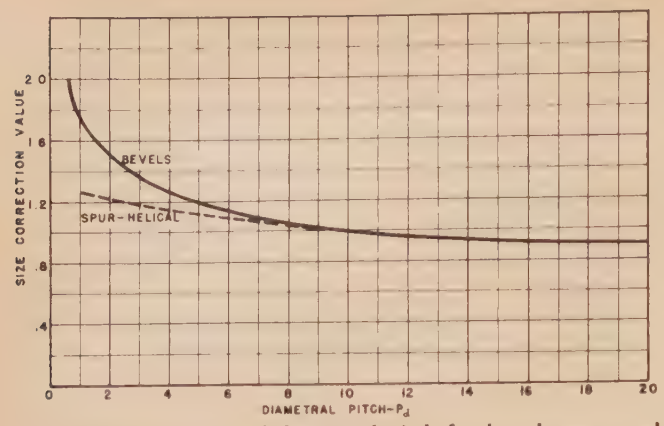

Fig. 6—Size correction of diametral pitch for bevel gears and for spur and helical gears

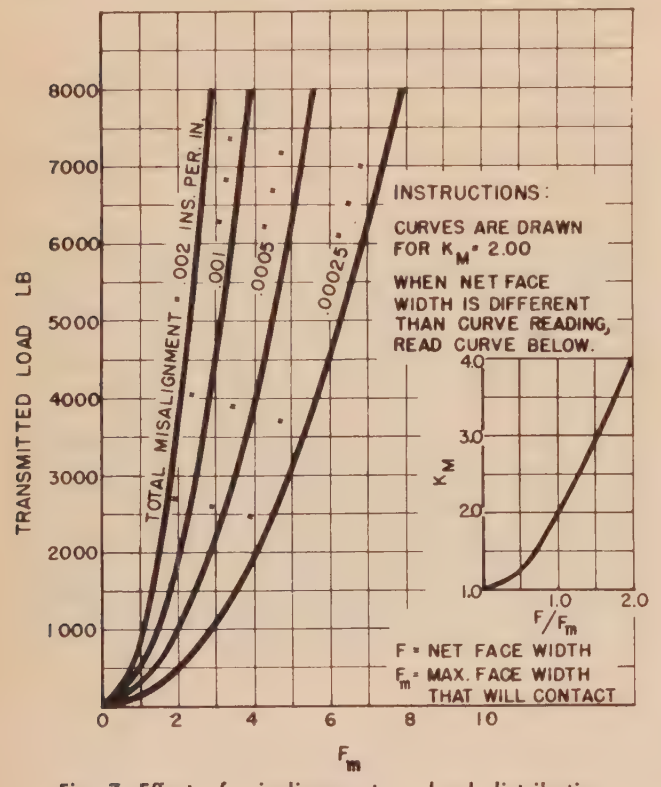

Fig. 7—Effect of misalignment on load distribution

size effect is a reflection of the statistical probabilities of a weak spot being encountered. This vulnerability is demonstrated more clearly in another report.[15]

Coleman reports the necessity of using the size correction shown in Fig. 6 for bevel gears. Dudley and the author are in agreement that for spur and helical gears the maximum size factor need not exceed the values indicated in Fig. 6. Actually, for high quality materials heat-treated with the best techniques, it appears that very little size effect is obtained. The difference between the values required for bevel and that suggested for spurs and helicals undoubtedly reflects the greater amount of experimental evidence obtained with case-carburized bevel gears of coarser pitch in which the strengthening effect of the case is not nearly as large as with finer pitches. This might be considered when analysing carburized spur and helical gears of coarse pitch.

Fig. 6 is based upon allowable stresses for 10 diametral pitch. The allowable stress for bevel gears is based upon one diametral pitch, and actual numerical values for K_s for other pitches is accordingly computed.

The present basis for size factors is largely experimental evidence. Considerable theoretical and laboratory test progress is being made as applied to materials generally. This data should prove useful for a more precise evaluation of a size factor particularly in regard to the other six variables which are believed important.

Load Distribution Factor (K_m)—The load distribution factor K_m depends upon:

1. Misalignment of axes of rotation.
2. Alignment errors due to tooth inaccuracies.
3. Elastic deflection of shafts and bearings under load.

Errors in cutting gear teeth, excessive elastic deflections, or high bearing clearances can be great enough to cause less than the full face width to contact. It can be imagined that from this extreme to a condition where full contact but nonuniform loading still occurs across the face is only a matter of degree. For large deflections it is common practice to taper the teeth so that more uniform contact is secured under loading. In addition, the deflections of the teeth themselves partially compensate for misalignment, and it is possible to calculate the amount of face width that would contact as well as the overloading due to the uneven load distribution. The development of this calculation is credited to Van Zandt [16] whose formula is

$$K_m = \frac{F_m^2 u \alpha}{2W_t} + 1 \qquad (8)$$

using F^2 instead of F_m^2 if it is smaller.
where:

u = Misalignment, in. per in.
α = Stiffness constant
F_m = Amount of face width working under different amounts of misalignment

A recommended curve for F_m and K_m factors is shown in Fig. 7. This curve seems applicable to both spur and helical gears although more precise studies are desired. It must be kept in mind that the total misalignment errors vary with a number of factors including load. Situations might occur in which the total deflection misalignment with load increases more rapidly than the compensating deflection of the

[15] WADC Technical Report 54-531, Supplement I, October, 1955: "Investigation of Materials Fatigue Problem Applicable to Propeller Design," by H. N. Cummings, F. B. Stalen, and W. C. Schulte.

[16] *AGMA*, 1951: "Proposed Practice for Strength of Spur Gears," by R. P. Van Zandt.

tooth, and a decrease rather than an increase in F_m might result with a higher transmitted load.

For the bevel gear load distribution factor, a rise in mounting displacement increases to K_m factor for a given bevel gear face width.

For face widths exceeding those equal to F_m in Fig. 7, the errors per inch of face across the face width are in some areas compensating so that a wave-like error rather than a uniformly increasing error is secured. Under these circumstances special design procedures involving bending and torsional deflections are followed and, when correlated with special manufacturing techniques, result in a satisfactory contact. In fact, the load and no-load contact can act as a guide for establishing a K_m factor. Table 4 shows typical variations.

The validity of the notions expressed in the F_m factor has been verified both in the laboratory and in the field. The numerical accuracy has been partially checked although further experience is required for many special applications.

Geometry Factor—The geometry factor is similar to the Lewis form factor because it evaluates the tooth shape. Recognition is required of the fact that the ordinary gear tooth is a short or stub beam with varying widths for which imperfect knowledge exists as to stress computations. Modifications are seemingly necessary to compensate for loadings close to the root of stub beams, which condition is not easily analyzed by existing stress formulas.

Other useful information [17-20] on this phase has also been contributed.

Along with these factors, cognizance must be given to the fact that instead of the simple bending stress system of Lewis, actually a complex combined stress system is present which has fairly high-stress concentrations and responds to dynamic effects as allowed by the elastic flexibility of the system. The geometry factor for the proposed AGMA practice evaluates the shape of the tooth, the position at which the most damaging load is applied, stress concentration effects, and the sharing of load between one or more pairs of teeth.

The geometry factor is:

for spur gears

$$J = \frac{Y}{K_F} m_N \qquad (9)$$

for helical gears

$$J = \frac{Y}{K_F} m_N \cos \psi \qquad (10)$$

where:

Y = Form factor taken at most critical position of loading

K_F = Stress correction factor which depends on
1. effective stress concentration
2. location of load
3. plasticity effects
4. residual stress effects
5. materials composition effects
6. surface finish
   a. gear production finish
   b. resulting from service
7. Hertz stress effects
8. size effect
9. end of tooth effects

m_N = Load sharing factor, composed of
1. profile contact ratio
2. face contact ratio
3. crowning effect
4. strengthening effect of unloaded ends

The assumptions for the positions of the simulated concentrated load for spur, helical, and bevel gears are shown in Figs. 8, 9, and 10.

In the proposed standard, it is to be noted particularly that accuracy limits are presented which must be adhered to before spur gear teeth can be expected to share the load. These limits indicate that exceptionally close tolerances must be met in manufacturing, held in mounting, and maintained in service if "load sharing" is to be other than an academic expression.

The stress correction factors for spurs and bevels

Table 4—Load Distribution Factor (K_m) for Wide Face Widths

Face Width Pitch Diameter	Contact	K_m
1.0 or less	95% face width obtained at 1/3 torque	1.4 at 1/3 torque
	95% face width obtained at full torque	1.1 at full torque
1.0 or less	20% face width contact at 1/3 torque	4.0 at 1/3 torque
	75% face width contact at full torque	2.5 at full torque

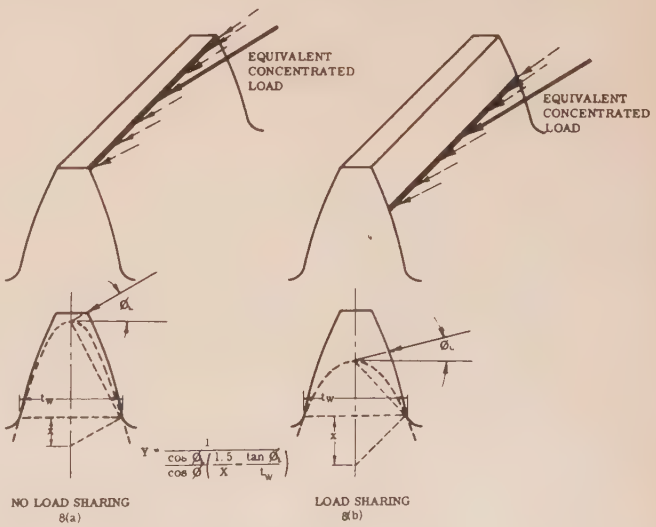

Fig. 8—Assumed concentrated load position and Y-factor determination for spur gears

17 *Mechanical Engineering*, Vol. 59, 1935, p. 225: "Deflection of Long Helical Gear Tooth," by C. W. MacGregor.
18 *Journal of Applied Mechanics*, Vol. 4, 1937: "Cantilever Plate with Concentrated Edge Load," by D. L. Holl.
19 *SESA Notes*, Vol. 1, 1953, pp. 2–5: "Subsurface Bending Stresses in a Gear Tooth," by G. E. Kochanek.
20 "Stresses and Deflections in Eccentrically Loaded Gear Teeth," by J. Marin and R. H. Shenk. Paper presented at ASME Annual Meeting, November-December, 1954.

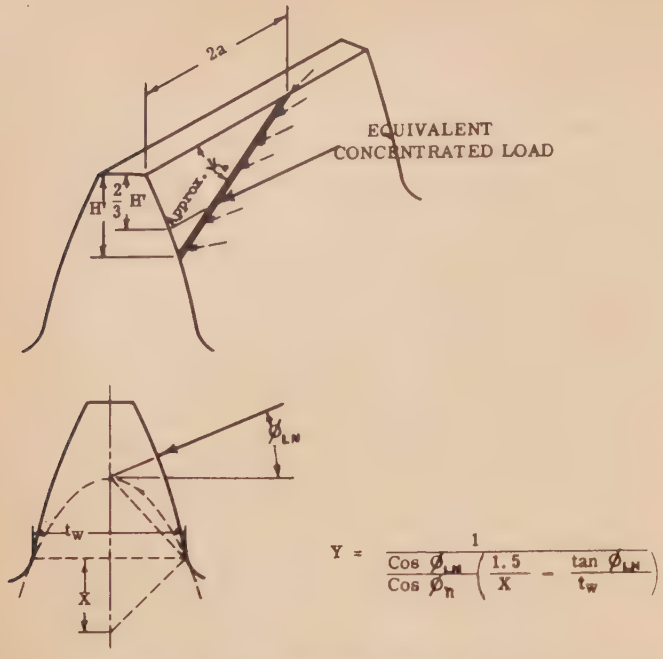

Fig. 9—Assumed concentrated load position and Y-factor determination for helical gears

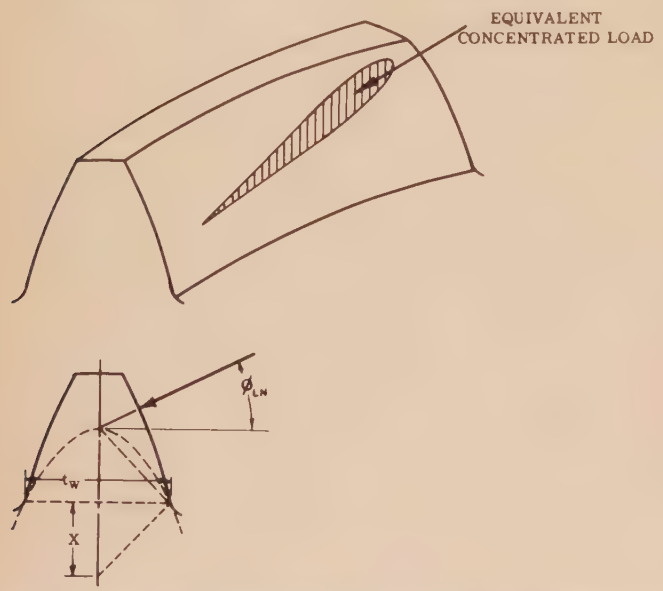

Fig. 10—Assumed concentrated load position and Y-factor determination for spiral bevel gears

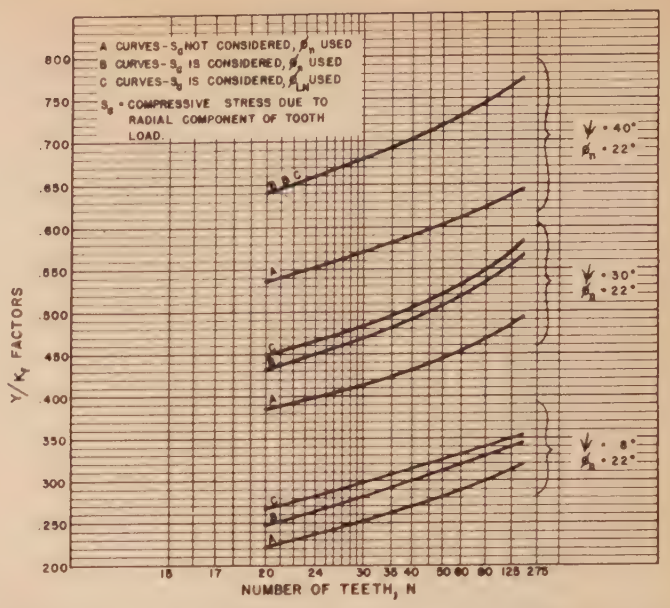

Fig. 11—Comparison of compressive stress developed by radial component of tangential load when considered at various pressure angles for helical gears

are based upon the work of Dolan and Broghamer.[21] Those selected for helicals reflect the special condition of the oblique line of contact and represent a compromise between determinations by photoelastic studies and actual laboratory tests. The classic contribution of Dolan and Broghamer illustrated that the stress concentration factor varied with the direction and location of the load on the profile. Additional useful data was furnished by Heywood,[10] Connell,[22] and Hein.[23] It must be recognized that the stress concentration varies from one at a relatively few numbers of cycles to a maximum at approximately 10^6 cycles as had been reported by Finch.[24]

Since only one pair of teeth carry the maximum load in spur gears, the m_N factor is one. Corrections for "crowning" and the strengthening effects of unloaded ends are not numerically measured at the present time.

In bevel gears, the amount of sharing of the load was determined by integrating the contact area. This procedure is not readily adaptable to helical gears since the variations in tooth proportions and angles are not as rigidly standardized. For helicals, the minimum contact length is established as a criterion. This is satisfactory because deflections at the ends relieve the load on what otherwise might be the critically loaded section. This procedure requires that helical gears with low profile or axial contact ratios be more critically analyzed or be subjected to the use of a larger factor of safety. The term cos ψ in the helical gear geometry factor compensates for the increased magnitude of the forces in the normal plane.

For helical gears the comparative effects of the compressive stress developed by the radial component of the tangential load when considered at the normal pressure angle and at the normal working pressure angle and when not considered is shown in Fig. 11. It is evident that the helical angle used has an influence. Recognition must be given the enlargement used to prevent undercutting as well as the reduction in

[21] University of Illinois Engineering Experiment Station Bulletin 335 (1942): "Photoelastic Study in Gear-Tooth Fillets," by T. J. Dolan and E. L. Broghamer.
[22] *AGMA*, 1942: "Progress Report of Investigation to Determine Rupture Factors for Various Diagonal Loading of Gear Teeth," by C. B. Connell.
[23] Masters Thesis, University of Wisconsin, 1950: "Investigation of Root Fillet Stress Concentration Factors in Herringbone Gearing," by Q. Hein.
[24] *Proceedings of ASTM*, Vol. 52, 1952, pp. 759–778: "Study of Fatigue of Steels in Finite Region of S-M Curve," by W. A. Finch.

Table 5—Life Factors K_L

Number of Cycles	Spur and Helical Gears: 300 Bhn	Spur and Helical Gears: Case-Carburized	Bevel Gears: Case-Carburized
1000	3.2-3.6	2.6	4.6
10,000	2.5-2.4	2.0	3.1
100,000	1.8-1.6	1.5	2.1
1,000,000	1.4-1.1	1.1	1.4
10,000,000	1 -1	1	1
100,000,000	0.8-0.8	0.8	1

thickness used for the purpose of obtaining backlash. Tip relief naturally has an influence which in effect reduces the height of the tooth.

Factors Associated With Allowable Stress

The relation for allowable stress, s_a, to calculated stress, s_t, is given in AGMA 225.01 as

$$s_t \leqq \frac{s_a K_L}{K_T K_R}$$

where:

s_t = Calculated stress
s_a = Allowable stress
K_L = Life factor
K_T = Temperature factor
K_R = Factor of safety

Allowable Stress—The allowable stress for gear materials varies considerably with heat-treatment, forging or casting practice, and composition. Commercial gear materials rather than being homogeneous are actually heterogeneous which implies all the concepts of spread or statistical variation modern interpretation places upon this condition. This requires that allowable stress values be modified for a given number of life cycles, operating temperature, and with a proper factor of safety.

The allowable stresses proposed are shown in Fig. 12 comparing the theroetical endurance limit (500 x ultimate tensile strength) and the values for varying percentages of survival at 10^7 cycles obtained from a large number of laboratory specimens as recorded in the WADC [15] report. It was previously reported that some conservatism is required to compensate for the initial rotation not included in the dynamic factor. It would appear that even considering the nonreversal of stress in the normal gear tooth load cycle, the allowable stresses have been set at a realistic level; where reversals occur such as in idlers, 70% of the suggested values are normally recommended. This derating might in time prove to be too high. A life factor is used to adjust the allowable stress for the required number of cycles.

Life Factor (K_L)—Typical values for life factors are shown in Table 5. When it is remembered that the stress developed in gears is not directly proportional to the load and that even the stress concentration

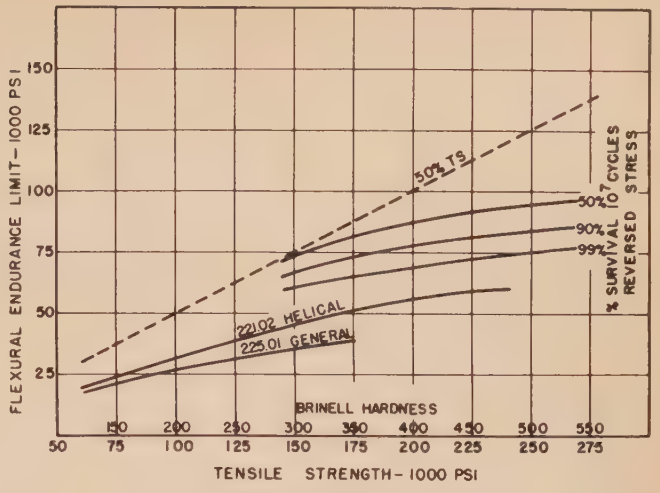

Fig. 12—Endurance limit versus tensile strength for allowable stresses

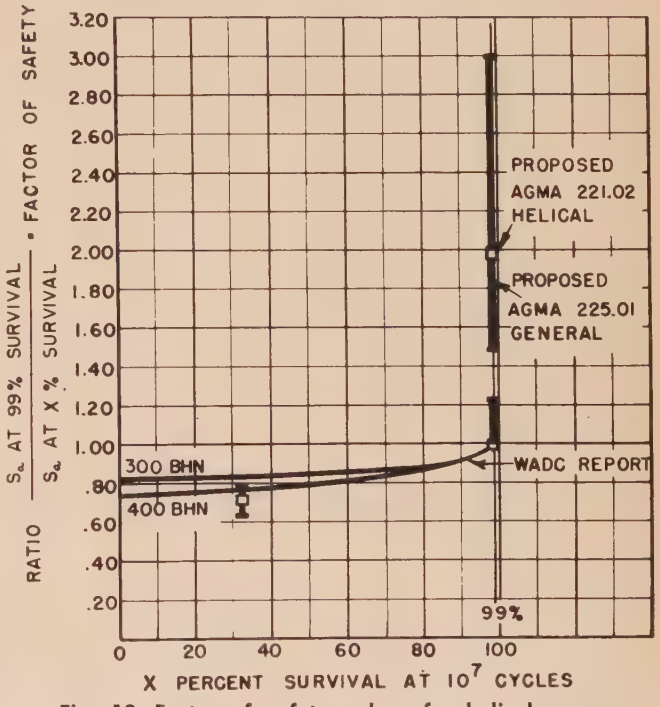

Fig. 13—Factor of safety values for helical gears

varies with the number of cycles, it is evident that setting a life factor is a difficult procedure. Fortunately, data have been published on this subject.[13,25,26]

Consideration of service damage or wear which might occur when extremely large numbers of years of service are desired might suggest an increased overload factor or more likely a higher factor of safety.

Factor of Safety (K_R)—An increasing awareness of the spread or scatter of material properties and the similar performance of actual machine elements has brought about a wider scope of meaning to the term "factor of safety." The old concept of factor of safety to guard against any possible failure or to cover

[25] *AGMA*, 1951: "Discussion of Allowable Bending Stresses for Spur Gear Teeth," by D. W. Dudley.
[26] *Proceedings of Society for Experimental Stress Analysis*, Vol. 10, 1953, pp. 37–52: "Strength Stress Life of Helical Gear Teeth," by E. J. Wellauer.

ignorance must accommodate modern notions regarding probable percentages of failures.

The factor of safety is introduced in the AGMA strength equation to give the engineer an opportunity to design for maximum safety or when required by economical or technical reasons to design for a calculated risk. Since the life factor incorporates an adjustment based upon finite fatigue life at low cycles, the suggested factors of safety augment the life data and must be so considered rather than judged with factors customarily used in other machine design problems.

It must be strongly emphasized that the term "failure" in the factors of safety does not mean an immediate failure but rather a shorter life than specified by the life curve.

The suggested degree of risk is not without a firm foundation. Using the WADC [15] data and plotting the survival at 10^7 cycles produces the factor of safety values shown in Fig. 13. A proposed general range and specific values for helical gears are shown. It is believed that these represent reasonable figures.

Temperature Factor (K_T)—Temperature can affect the capacity of gearing both by its tempering effect on the material and the dimensional heat distortions which might occur. For most spur and helical gears, if the maximum oil or gear blank temperatures do not exceed 250 F, K_T is generally taken as 1.0. In some instances, when case-carburized gears operate above 160 F, a K_T value greater than 1.0 is required. For those cases, the values presented in the bevel gear formula can be used as a guide. Based upon limited test data, the bevel gear temperature factor is given as:

$$K_T = \frac{460 + T_F}{620} \qquad (12)$$

where:

K_T = Temperature factor
T_F = Operating temperature of cooling oil, F

Summary

It is seen that careful evaluation of over seven major variables are required to estimate satisfactorily the stress in gear teeth. The following spur gear example, using only accuracy (or quality of manufacture) as a criterion, shows that these modifications factors can exert a major influence.

Gear Specifications: 20/80T, 10 DP, 3-in. FW, 4/1 GR, 5-in. centers at full depth

Nominal Loading: 3000-lb tangential load

Ratio of Factors (minimum to maximum accuracy):
Load sharing (J), 1.42/1
Load distribution (K_m), 2.56/1
Velocity (K_v) 500 rpm, 1.24/1; 10,000 rpm, 2.02/1
Material (S_a), 1.43/1

If these factors are compounded it is evident that at 500 rpm the accurate gear will have 6½ times the capacity of the less accurate gear set; at 10,000 rpm the accurate gears will have a capacity over 10 times that of the less accurate gears. This is not startling to the designer of high-speed gearing who realizes that gear precision pays and also must be "paid for." The difference between the accuracy of the two sets is only a few thousandths in the face misalignment and a few tenths of a thousandth in profile or spacing errors. Major differences in capacity can result because of small differences in accuracy normally encountered in and between production runs. This should be significant to the engineer involved in gear testing since it indicates that considerable scatter might result in gear tests unless the range of accuracy tolerances is maintained within narrow limits. The implications of the danger involved in interpreting test results to different sizes of gear sets or to less accurate production gears need no discussion. This is viewed particularly in light of the fact that overload factors can cause an additional variation of 2.25/1 for specific applications.

Conclusion

A study of the analytical derivation and modifying factors obtained by experience (field and laboratory) used to calculate gear stresses by a proposed AGMA formula shows that substantial progress in more precise evaluations has been made within the past few years. The formula certainly will prove useful in regulating gear tests because it points out the relative influence of certain variables and estimates their effect on test scatter as well as on the validity of the test result as projected into service performance.

A satisfactory formula for calculating gear stresses must depend upon a good technical base, be modified by those principal factors known to influence gear stress, be applicable to all types of gears, and have the range of validity of application clearly outlined.

The proposed AGMA strength formula for gear teeth fulfills these requirements. In addition, emphasis has been to provide easy modification of the basic equation to incorporate more precise computations, information derived from laboratory tests, field experience, advance analyses, and a better understanding of gear applications.

The ability to grow is a primary requisite of any basic formula.

Acknowledgments

The author has made use of data accumulated by the Gear Rating Coordinating committee of the American Gear Manufacturers Association. Thanks are offered to AGMA and the various members of the committee for permission to publish this information. Special acknowledgment is made to Darle Dudley of the General Electric Co. and to Wells Coleman of the Gleason Works for their extensive help.

Stopping Ability of Motor Vehicles Selected from General Traffic

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This paper was presented at the SAE National Transportation Meeting, New York, Oct. 10, 1956.

AT the request of representatives of both government and industry, the Bureau of Public Roads in 1941 initiated a rather broad program of research on motor vehicle brakes. Although the program was halted by World War II, one important phase of the work—tests of motor vehicles selected from the everyday traffic—was completed and reported in 1944. In 1947, the program was reorganized, and field work was performed from November, 1948, to September, 1951. A comprehensive report [1] on the findings of the postwar research was published in 1954.

As a result of the information published by the Bureau of Public Roads, the National Committee on Uniform Traffic Laws and Ordinances revised its Uniform Vehicle Code [2] to provide a more realistic brake performance requirement. Since that time seven states have revised their brake performance requirements, patterning them after the Uniform Vehicle Code.

This report, which covers tests conducted during 1955, will provide current information on the braking ability of various types of motor vehicles as they appear in the general traffic. It will bring up to date similar information presented in earlier reports and will compare present levels of brake performance with those determined by earlier tests.

As in previous brake studies, the Bureau of Public Roads was responsible for collection and analysis of the data, but each state in which tests were conducted provided assistance with the field work. Personnel for weighing the vehicles and recording data were furnished by the Maryland State Roads Commission, the California Division of Highways, and the Michigan State Highway Department. Uniformed personnel, who stopped the vehicles for submitting to test, and who kept traffic moving smoothly at the test sites, were provided by Maryland State Roads Commission Truck Patrol, California Highway Patrol, and Michigan State Police. Test apparatus used in the field work was furnished by General Motors Proving Ground, McCormick Engineering Co., and the Institute of Transportation and Traffic Engineering of the University of California.

Purposes of the Study

The primary purposes of the study and of this report were:

1. To show the levels of brake performance that now prevail for the various types of vehicles currently found on the highways.
2. To bring up to date, information used as a basis for formulating brake performance regulations for such vehicles.
3. To provide up-to-date motor vehicle brake performance data which may be used in establishing highway design standards.
4. To show the trend in levels of brake performance of the various types of vehicles using the highways.
5. To focus attention on the existing brake per-

1 "Braking Performance of Motor Vehicles," by Carl C. Saal and F. W. Petring. Pub. by United States Government Printing Office, Washington, 1954.
2 "Uniform Vehicle Code, Revised 1954." Pub. by National Committee on Uniform Traffic Laws and Ordinances, Washington, 1954.

THIS paper presents the results of tests conducted during 1955 on more than 1200 vehicles selected from the general traffic. It compares past and present levels of brake performance and shows that since 1949 improvements in the brake performance of most vehicle types have been small.

Current levels of brake performance are indicated with respect to vehicle type, gross weight, vehicle capacity, and axle load.

The paper also indicates the ability of different types of vehicles to comply with current Uniform Vehicle Code specifications for brake performance.

formance situation, particularly with respect to the effect of load and vehicle type.

Terminology

Certain terms used in this report must be thoroughly understood in order to have a clear understanding of the results. Definitions of these terms follow.

Brake-system application and braking time or distance—The time elapsed or distance traveled between the instant or point at which the driver starts to move the braking controls and the instant or point at which the vehicle comes to rest. (This definition is identical to the definition for "vehicle stopping time or distance" which appears both in the BPR 1954 report, "Braking Performance of Motor Vehicles," and in the SAE Handbook.)

Braking time or distance—The time elapsed or distance traveled between the instant or point of first retardation by the brakes and the instant or point at which the vehicle comes to rest.

Brake-system application time or distance—The time elapsed or distance traveled between the instant or point at which the driver starts to move the braking controls and the instant or point of first retardation by the brakes.

Brake force buildup time or distance—The time elapsed or distance traveled between the instant or point of first retardation by the brakes and the instant or point maximum braking force is attained.

Pedal reserve—The distance, in inches, between the floor board or mat and the back of the pedal at the completion of a stop.

Swerving—An uncontrollable lateral movement of the vehicle when the brakes are applied, which is involuntary on the part of the driver.

Deceleration—The rate of reduction of the vehicle speed in feet per second per second.

Maximum deceleration—The greatest deceleration measured during the stop regardless of the length of time it is sustained.

Manufacturer's gross vehicle weight rating—The weight, in pounds, of the truck chassis with lubricants, water, and full tank or tanks of fuel, plus the weights of cab or driver's compartment, body, special chassis and body equipment, and payload as authorized by the chassis manufacturerer.

Vehicle capacity—In the case of single-unit trucks, vehicle capacity is the maximum gross vehicle weight rating; in the case of combination units, it is the gross combination weight recommended by the vehicle chassis manufacturer for a given truck-tractor or truck to be used in combination with semitrailers or trailers.

Brake-system types—Hydraulic: Brake shoes are actuated by hydraulic brake cylinders operated with hydraulic line pressure developed by a pedal-operated hydraulic brake master cylinder.

Mechanical: Brake shoes are actuated by a cam operated by a cable or rod linked to the brake pedal.

Vacuum-booster hydraulic: Brake shoes are actuated by a hydraulic brake wheel cylinder operated with hydraulic line pressure developed by a vacuum-powered master cylinder or a vacuum hydraulic power unit.

Air-booster hydraulic: Brake shoes are actuated by a hydraulic brake wheel cylinder operated with hydraulic line pressure developed by an air-powered master cylinder or an air hydraulic power unit.

Vacuum-mechanical: Brake shoes are actuated by a cam operated by a vacuum brake chamber through a mechanical linkage.

Air-mechanical: Brake shoes are actuated by a cam operated by an air brake chamber through a mechanical linkage.

Vehicle Type Code

For convenience in reference, in some cases commercial vehicle types are represented by the code shown below. Each digit represents the number of axles of a vehicle or of one unit of a vehicle combination. A combination symbol consisting of two or three parts separated by hyphens indicates a vehicle combination. The first digit of a combination symbol represents the power unit. An "S" in the second part of a combination symbol indicates a semitrailer, and the power unit, of course, is a truck-tractor. A digit appearing without an "S" in either the second or third position in a combination symbol represents a full trailer.

2 = 2-axle single-unit truck
3 = 3-axle single-unit truck
2–S1 = 2-axle truck-tractor with 1-axle semitrailer
2–S2 = 2-axle truck-tractor with 2-axle semitrailer
3–S2 = 3-axle truck-tractor with 2-axle semitrailer
2–2 = 2-axle truck with 2-axle trailer
2–3 = 2-axle truck with 3-axle trailer
3–2 = 3-axle truck with 2-axle trailer
3–3 = 3-axle truck with 3-axle trailer

2–S1–2 = 2-axle truck-tractor with 1-axle semitrailer and 2-axle trailer
2–S2–2 = 2-axle truck-tractor with 2-axle semitrailer and 2-axle trailer
2–S2–3 = 2-axle truck-tractor with 2-axle semitrailer and 3-axle trailer
3–S1–2 = 3-axle truck-tractor with 1-axle semitrailer and 2-axle trailer

Illustrations of the various types of commercial vehicles tested and their respective code designations appear in Fig. 1.

Fig. 1—Commercial vehicle types as designated by code based on axle arrangement

Scope of Tests

Tests to determine the brake-system application and braking distances of a representative number of vehicles were conducted on passenger cars and commercial vehicles selected at random from the general traffic. The vehicles were stopped on the highway by a uniformed policeman, weighed, and described, and then subjected to emergency stops from a speed of 20 mph. The test was not compulsory, and each driver clearly understood that punitive action would not be taken. The tests were conducted during 1955 on heavily traveled highways in Maryland and Michigan at the same sites as were used in 1949 and in California at a site in the same general area as that used in 1949.

Table 1 shows, by type and axle arrangement, the number of vehicles tested in each state in 1949 and in 1955. A similar series of tests, which was conducted during 1941 and 1942, included 907 passenger cars, 1403 single-unit trucks, 1597 truck-tractors with semitrailers, 216 trucks with full trailers, and 36 truck-tractors with semitrailers and full trailers. This early study was conducted at 11 locations in 10 states of wide geographic distribution. In order to maintain a continuing record of the levels of brake performance of the various types of vehicles in use on the highways it is planned to conduct similar tests about every five years at locations as near as practicable to the three used in the 1949 and 1955 tests.

Test Sites

All of the tests were conducted on dry, smooth pavement which was free of foreign material. Each test section was approximately level and located on

Table 1—Number of Vehicles Tested by State, Type, and Axle Arrangement in 1955 and in 1949

Type of Vehicle [a]	Number of Vehicles Tested							
	1955				1949			
	Md.	Mich.	Calif.	Total	Md.	Mich.	Calif.	Total
Passenger Cars	104	101	105	310	99	112	100	311
Single-Unit Trucks								
2-Axle	134	138	128	400	132	148	87	367
3-Axle	20	28	25	73	19	4	23	46
Subtotal	154	166	153	473	151	152	110	413
Truck-Tractors with Semitrailers								
2-S1	69	39	21	129	132	89	25	246
2-S2	93	43	17	153	17	44	37	98
3-S2	2	29	35	66		2	55	57
Subtotal	164	111	73	348	149	135	117	401
Trucks with Trailers								
2-2		9	7	16		6	3	9
2-3			1	1			2	2
3-2		13	33	46		4	12	16
3-3			7	7			27	27
Subtotal		22	48	70		10	44	54
Truck-Tractors with Semitrailers and Trailers								
2-S1-2		9	35	44		18	32	50
2-S2-2		5	2	7		6	3	9
2-S2-3		2		2		1		1
3-S1-2			1	1				
Subtotal		16	38	54		25	35	60
Other Vehicle Combinations								
Passenger-Car Driveaway-Towaway Combinations		3		3				
Housetrailer Factory-Towaway Combinations		5		5				
Subtotal		8		8				
Grand Total	422	424	417	1263	399	434	406	1239

[a] See section entitled, "Vehicle Type Code," for an explanation of the code used.

Fig. 2—Approach to Maryland test site

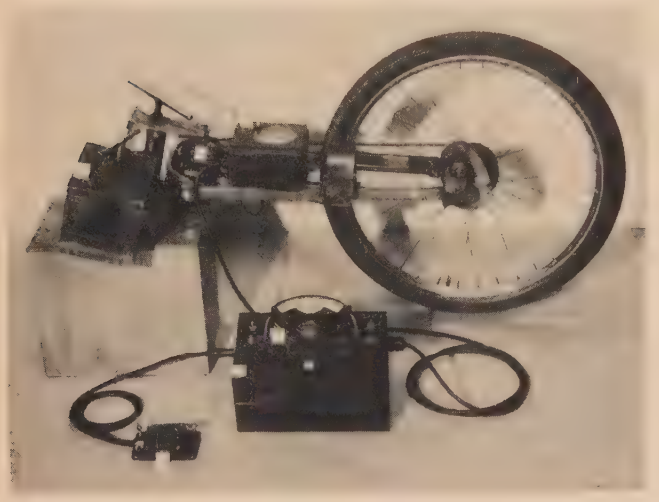


Fig. 3—Apparatus for measuring speed and brake-system application and braking distance: (1) speedometer, (2) control box, (3) pedal switch, (4) running board clamp, (5) odometer, and (6) generator

a tangent. The pavements at the Michigan and California test sites were portland cement concrete, and that at the Maryland site was portland cement concrete which had been covered with bituminous surfacing. In order to determine whether results obtained on the bituminous surface would be comparable with those obtained on portland cement concrete surfaces, a few preliminary tests were run before final selection of the Maryland test site. Locked-wheel stops were made with a passenger car both on the bituminous-covered section of pavement, which was later selected for the brake tests, and on a section of the same highway having a good portland cement concrete surface. Results of these tests of pavement surfaces showed the sliding coefficient of friction for stops made from 20 mph to be almost identical at the two sites and within the range of values normal for good portland cement concrete pavement surfaces. The site with the bituminous surfacing was selected for this study because it was the one used for the 1949 study in Maryland, which was prior to its resurfacing, and was advantageous from the viewpoint of safety.

Each of the three test sites was located on a section

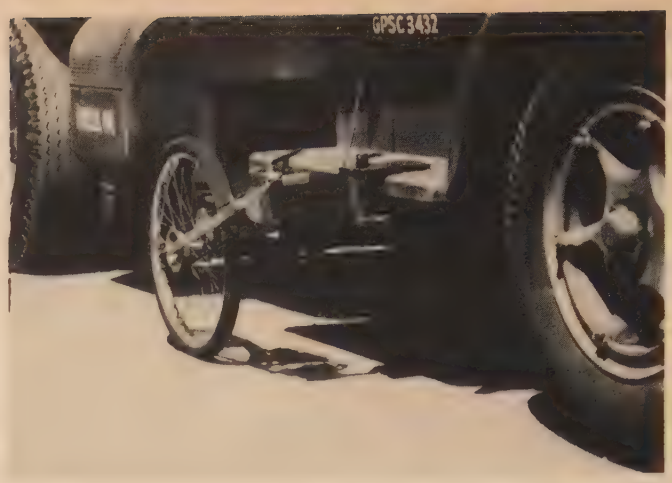


Fig. 4—Test wheel attached to truck with running board clamp

of 4-lane divided highway. The lane in which tests were conducted was closed off with rubber traffic cones placed at approximately 75-ft intervals for a distance of about one-half mile. All through-traffic was diverted to the other lane. Advance signs warned traffic to slow down, indicated that brake tests were in progress, and directed through-traffic into the open lane (Fig. 2). A policeman on foot stationed at the approach to the test lane directed vehicles to be tested into the weighing area and kept other traffic moving smoothly through the open lane.

Instrumentation

The key measurement in the study was brake-system application and braking distance. A test wheel (Fig. 3) was used to measure this distance in feet and to indicate vehicle speed in miles per hour. Methods used for attaching the test wheel to commercial vehicles and to passenger cars are illustrated in Figs. 4 and 5.

The apparatus was so arranged that the instant the driver put his foot on a switch, which was securely fastened to the face of the brake pedal on the test vehicle, an electrical circuit was completed which started the distance-measuring device. The circuit was maintained by a holding relay until released by the observer after the vehicle had come to rest. Thus, the distance was measured from the point at which the driver started to move the braking control to the point at which the vehicle came to rest. Whenever a hand-control valve was to be used during the brake test, a small finger-like switch was attached to the valve-control handle and included in the circuit in conjunction with the pedal switch. Distance was thereby measured from the point of first contact with a braking control.

Since the brake-system application and braking distance varies with speed at a significant rate, an accurate determination of speed also was essential. The vehicle speed was indicated directly by a voltmeter calibrated in miles per hour. The voltmeter, which was held in the lap of the observer, was wired

to a belt-driven generator mounted on the frame of the test wheel.

In order to insure accuracy of the data, the test wheel was calibrated at frequent intervals. To calibrate the speedometer, the time required to travel a measured mile at a constant speed of 20 mph, as indicated by the test wheel, was measured with a stop watch and compared with the computed time required to travel a mile at that speed. In every instance the actual time required to travel the mile varied from the computed time by less than 1%. The distance indicated by the test wheel was also checked on the measured mile and found to vary from the true distance by less than 0.2%.

The accuracy of the test wheel in measuring brake-system application and braking distance was also verified by means of an electric detonator mounted on the bumper of the vehicle as shown in Fig. 6. The detonator, when fired, ejected a chalk capsule which marked the pavement directly beneath it. The firing mechanism was wired to the same pedal switch used to actuate the distance-measuring device on the test wheel. The instant the driver started to move the braking control, the gun fired causing the chalk capsule to mark the pavement, and simultaneously the test wheel started to measure the distance traveled.

After the vehicle came to rest, the distance was measured from a point on the pavement directly below the detonator to the point where the gun was fired, which was clearly indicated by the mark on the pavement. This distance was compared with the distance indicated by the test wheel. Periodic tests made in this manner showed a variation of less than 2% between the two methods of measurement.

In addition to the brake-system application and braking distance, measurements were made of the maximum deceleration as indicated by a portable decelerometer. The decelerometer used was a popular make of the pendulum type. A moving scale on the instrument is graduated to read per cent braking efficiency, which is actually the per cent of 1-*g* deceleration. For example, a reading of 80% on the instrument would be a deceleration of $0.80 \times 32.2 = 25.8$ fps per sec. Movement of the scale is actuated by and is proportional to movement of a damped pendulum. When the vehicle is moving at a uniform speed, the pendulum will assume a vertical position. However, if the vehicle speed is reduced, as by the application of brakes, the pendulum tending to move on at the same speed will swing forward. The tangent of the angle through which the pendulum moves forward away from the vertical is directly proportional to the deceleration. The instrument holds the moving scale at the maximum value attained until released by the observer.

Vehicle Sample

At the test location in each of the three states, approximately 300 commercial vehicles and 100 passenger cars were selected from traffic and tested. The commercial vehicles were selected so as to produce a sample composed of about 50% single-unit trucks and 50% combination vehicles. Selectivity was exercised also to provide a wide range of vehicle makes, gross weights, capacities, and brake types. The passenger cars were chosen to represent a wide range of makes and year models. Vehicles loaded in an unsafe manner, and those whose drivers objected to participating in the test, were excused. However, only a very small percentage of the drivers desired to be excused.

Test Operations

Each vehicle selected for test was stopped by a uniformed officer as it approached the test section. The driver was directed to a spot where a crew was stationed to weigh the vehicle, record pertinent descriptive data, and install the apparatus for measuring brake performance. Each axle of each vehicle to be tested, including passenger cars, was weighed. In Maryland the vehicles were weighed on loadometers

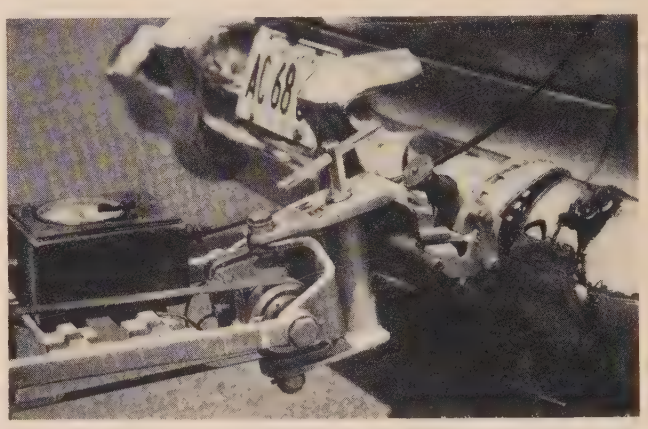


Fig. 5—Test wheel attached to passenger car with bumper clamp

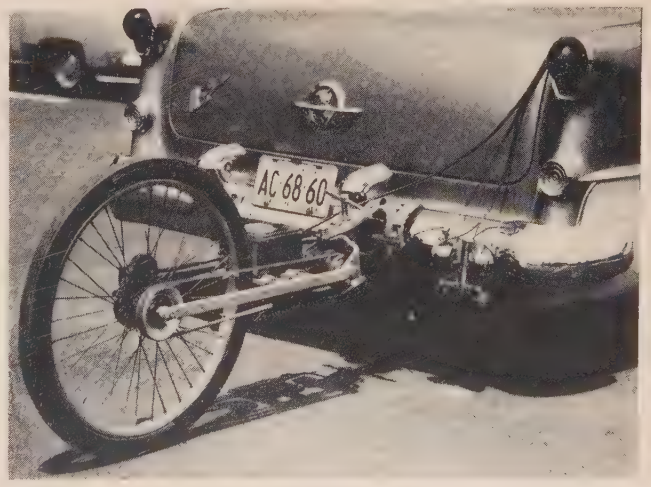


Fig. 6—Test wheel with detonator used for calibrating odometer

Fig. 7—Weighing trucks on loadometers prior to brake test

Fig. 8—Lineup of test vehicle and observer's vehicle in lane blocked off for brake test

set in pits in a parking area adjacent to the highway (Fig. 7), and in California and Michigan they were weighed on state-owned platform scales located adjacent to the test section.

Every vehicle tested was assigned a test number for use in coordinating the field data, which were recorded on four separate forms: one each for recording (1) axle weights, (2) vehicle characteristics, (3) skid data, and (4) brake performance data and other incidental data observed during the road tests.

Vehicle characteristics were recorded during the weighing operation. This description included the vehicle make, model, year model, type, manufacturer's gross weight rating, brake type, number of axles without brakes, total number of axles, and whether the vehicle was loaded or empty. When the weighing of the vehicle and the recording of vehicle characteristics had been completed, the apparatus for measuring brake performance was installed, and the road tests were conducted. The regular driver of the vehicle, under the direction of an observer who rode beside him, made the emergency stops. Test drivers were not used.

The road tests for each vehicle consisted of at least three emergency stops from a speed of 20 mph. The observer noted, in the case of a commercial vehicle, if there was a hand-control valve and, if so, whether it was used in making the stops. He also noted whether there was a valve for limiting the pressure applied to the front wheel brakes and, if so, the valve setting used in making the stops. After each stop the observer also recorded the brake-system application and braking distance as indicated by the test wheel, the decelerometer reading, whether the vehicle swerved, the brake pedal reserve, and whether the clutch was engaged or disengaged during the stop.

As previously mentioned, all traffic was excluded from the lane in which the tests were being conducted. However, as an added precaution in case some vehicle should slip between the rubber cones and into the test lane, each vehicle being tested was followed by a second observer in another vehicle. This second observer maintained such an interval between the two vehicles that other traffic could not squeeze between them. A large sign reading DANGER—BRAKE TESTS—SUDDEN STOPS was attached to the rear of the observer's vehicle, and a flashing red warning light was mounted on top of the vehicle as shown in Fig. 8. After each stop this second observer also measured and recorded the lengths of any skid marks left by the test vehicle.

Classification of Vehicles

One of the first steps in the analysis of the test results was the classification of data, first by vehicle type, second by vehicle capacity groups for each vehicle type, and finally by brake types for each vehicle capacity group. Table 1 shows the number of vehicles tested in each state, both in 1949 and 1955, by vehicle type and axle arrangement. Table 2 shows by capacity groups and by brake type, the number of vehicles of each type tested in 1955.

All commercial vehicles were segregated into vehicle capacity groups on the basis of the chassis manufacturer's gross vehicle weight or gross combination weight rating. Single-unit trucks were classified into four groups (very light, light, medium, and heavy), and the combination vehicles were classified into three groups (light, medium, and heavy). The ranges of gross weight ratings for each of the groups are shown in Table 3.

In many instances the vehicle chassis manufacturer's recommended maximum weight for a truck or truck-tractor, used in combination with truck trailers, was not available. When this occurred the combination was classified as light, medium, or heavy on the basis of the power unit when used as a single-unit truck.

The most common brake types for the single-unit trucks were hydraulic and vacuum-booster hydraulic.

The brake types most commonly found on the truck-tractor-semitrailer combinations were (1) vacuum-booster hydraulic on the power unit and vacuum-mechanical on the semitrailer and (2) air-mechanical on both units of the combination. The truck-full-trailer combinations and truck-tractor-semitrailer-full-trailer combinations were most commonly equipped with air-mechanical brakes on all units of the combinations. As in the 1949 study, air-mechanical brakes predominated on the heavy-capacity commercial vehicles of all types. The greatest percentage of the passenger cars were equipped with hydraulic brakes. However, the increasing popularity of power brakes was evident in that approximately 12% of the passenger cars tested were equipped with vacuum-booster hydraulic brakes. The percentage of passenger cars equipped with mechanical brakes observed in the study indicates that very few cars so equipped remain on the American highways. Out of the total sample of 310 passenger cars only two were equipped with mechanical brakes. One of the two was postwar vehicle of foreign manufacture and the other a prewar American car.

It is apparent from Table 2 that the limited size of the sample and the nature of its distribution by vehicle type, capacity group, and brake type precludes comprehensive comparisons of brake performance by brake type. However, the sample was adequate to compare the performance of the two predominating brake types observed on passenger cars, light single-unit trucks, and medium 2-axle truck-tractors with 1-axle semitrailers.

Table 2—Classification of Vehicles Tested by Vehicle Type, Capacity, and Brake Type

Vehicle Type and Capacity	Brake Type*	Number of Vehicles Tested			
		Md.	Mich.	Calif.	Total
Passengers Cars	H	92	82	98	272
	VBH	11	19	6	36
	M	1		1	2
2-Axle Trucks					
Very Light	H	40	23	40	103
	VBH		3		3
	M			1	1
Light	H	6	13	5	24
	VBH	50	68	59	177
	ABH			1	1
	M			1	1
Medium	H		1		1
	VBH	21	25	15	61
	ABH	3			3
	AM	9	1	2	12
	M		1		1
Heavy	VBH	1	1	1	3
	ABH	1			1
	AM	4	1	3	8
3-Axle Trucks					
Light	VBH	4		1	5
	ABH		5		5
	Other	2	1	1	4
Medium	VBH	2	7	9	18
	ABH	1	2		3
	AM	3	3	1	7
	Other	1			1
Heavy	VBH			3	3
	AM	7	10	10	27
2-Axle Truck-Tractors with 1-Axle Semitrailers					
Light	VBH-VM	5	13	2	20
	ABH-VM	4		1	5
	AM-AM			1	1
Medium	VBH-VM	2	13	7	22
	ABH-AM	4	1		5
	AM-AM	20	5	5	30
	Other	6	1	1	8
Heavy	AM-AM	25	5	4	34
	AM-VM	3	1		4

* Brake Types:
H = Hydraulic
M = Mechanical
E = Electric
VM = Vacuum-mechanical
AM = Air-mechanical
VBH = Vacuum-booster hydraulic
ABH = Air-booster hydraulic

Table 3—Classification of Vehicles by Capacity Group

Vehicle Capacity Group	Manufacturer's Gross Weight Rating, lb	
	Single-Unit Vehicles	Combination Vehicles
Very Light	10,000 and under	
Light	10,001—16,000	27,000 and under
Medium	16,001—24,000	27,001—44,000
Heavy	Over 24,000	Over 44,000

Consistency of Distance Data

As previously mentioned, the road tests of each vehicle consisted of three emergency stops for 20 mph. However, in many instances the first stop and sometimes the second stop did not represent the best possible braking performance of the vehicle, as frequently the driver was at first hesitant about making a true emergency-type stop, or did not thoroughly understand that the shortest possible stop was desired. In these instances, the length of the level test section was usually too short to make the additional test runs required to insure three consistent stops, and it was not practical to turn vehicles around on the highway.

Since each of the three runs did not always represent the best brake performance of the vehicle being tested, an average of the three runs would not correctly reflect the true level of performance. Therefore, the minimum brake-system application and braking distance recorded for each vehicle was used in the analysis of the test results. This procedure also was followed in analyzing the results of the two previous studies. On the average there was a variation of slightly over 4% between the minimum or the maximum and the average brake-system application and braking distance observed for the three stops.

General Levels of Performance

The levels of brake performance of the various types of vehicles selected from the general traffic in 1955 are indicated in Figs. 9 through 14. In Figs. 9, 10, and 11, where brake-system application and braking distance is the criterion of performance, the best performance is indicated by the shortest distance. But, in Figs. 12, 13, and 14, where deceleration is the criterion, the best performance is indicated by the highest deceleration. In studying these general results it must be kept in mind that the sample of vehicles of each type is made up of various gross weights, capacities, brake types, and conditions of maintenance.

Brake-system application and braking distance—Figs. 9 and 10 reveal quite clearly the differences in stopping ability of the various types of vehicles. The curves indicate the percentage of vehicles of each type that can stop in a given distance or less, from a

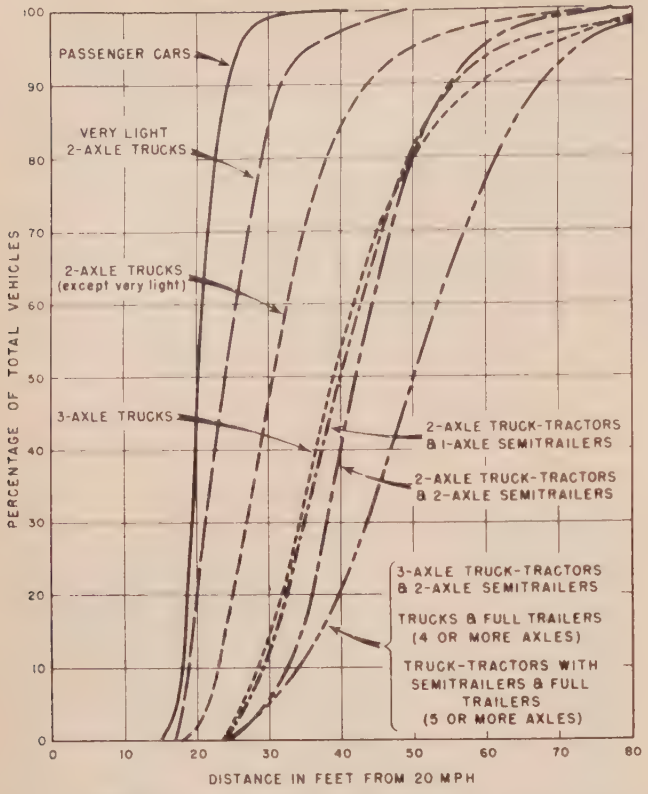


Fig. 9—Cumulative frequency distribution of minimum brake-system application and braking distances of motor vehicles selected from general traffic

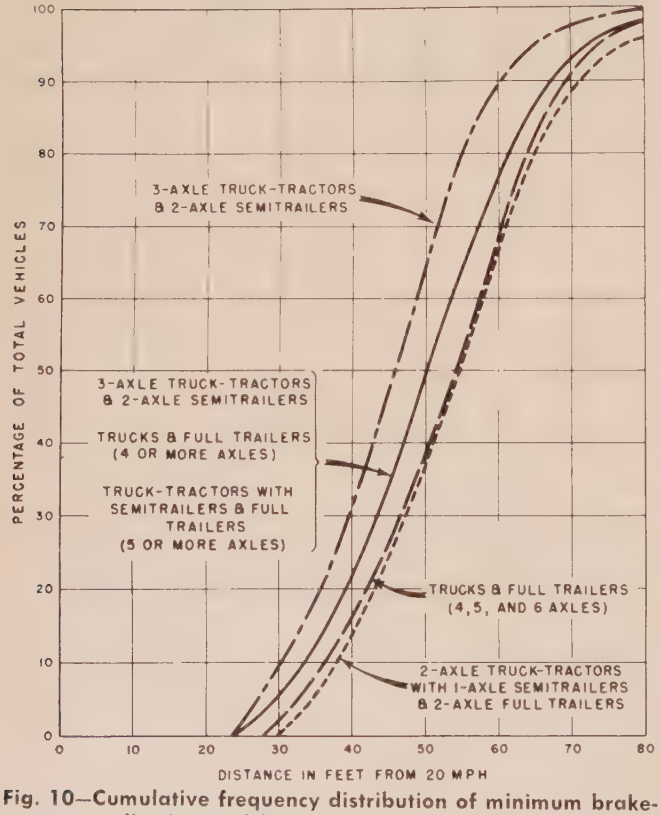


Fig. 10—Cumulative frequency distribution of minimum brake-system application and braking distances of larger combination vehicles selected from general traffic

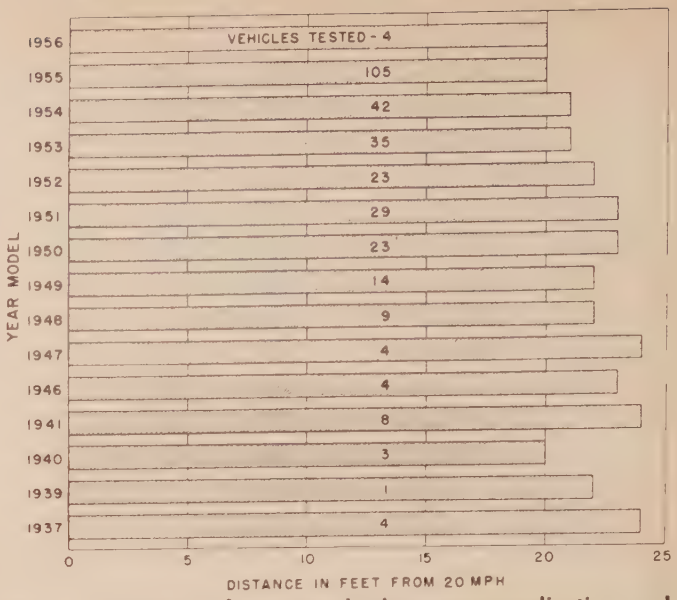


Fig. 11—Variation of average brake-system application and braking distance with year model for passenger cars with hydraulic brakes and vacuum booster hydraulic brakes

speed of 20 mph. For example, Fig. 9 shows that the 50-percentile values for the various types of vehicles were as follows: passenger cars 20 ft, very light 2-axle trucks 24 ft, other 2-axle trucks 31 ft, 3-axle trucks 39 ft, 2-axle truck-tractors with 1-axle semitrailers 40 ft, 2-axle truck-tractors with 2-axle semitrailers 42 ft, and all other combination vehicles 50 ft. It is of particular interest to note that the curves shown in the figure become closer together at the lower percentile values. Thus, improvement in the levels of braking performance of all vehicle types will bring the performance levels of the individual vehicle types closer together. It is obvious that the greatest possibilities for improvement exist with the larger commercial vehicles.

In Fig. 10, the performance of the group of larger combination vehicles, which had the poorest stopping ability of any of the groups represented in Fig. 9, is broken down into the three major types of these vehicles. For purposes of comparison, the composite curve for the entire group also appears in the figure. It is readily apparent that the 3-axle truck-tractors with 2-axle semitrailers, as a group, performed better than either the trucks with full trailers or the 2-axle truck-tractors with 1-axle semitrailers and 2-axle full trailers. The figure shows, also, that there was little difference in the braking performance of these latter two groups of combination vehicles.

The average performance of passenger cars equipped with hydraulic and vacuum-booster hydraulic brakes is shown by year model in Fig. 11. It is obvious that age had little influence on the stopping ability of this group of vehicles. On the average, the 1956 and 1955 model cars were the best performers, requiring 20 ft to stop from 20 mph. Also, the average stopping ability of the three 1940 model passenger cars tested equaled that of the 1955 and

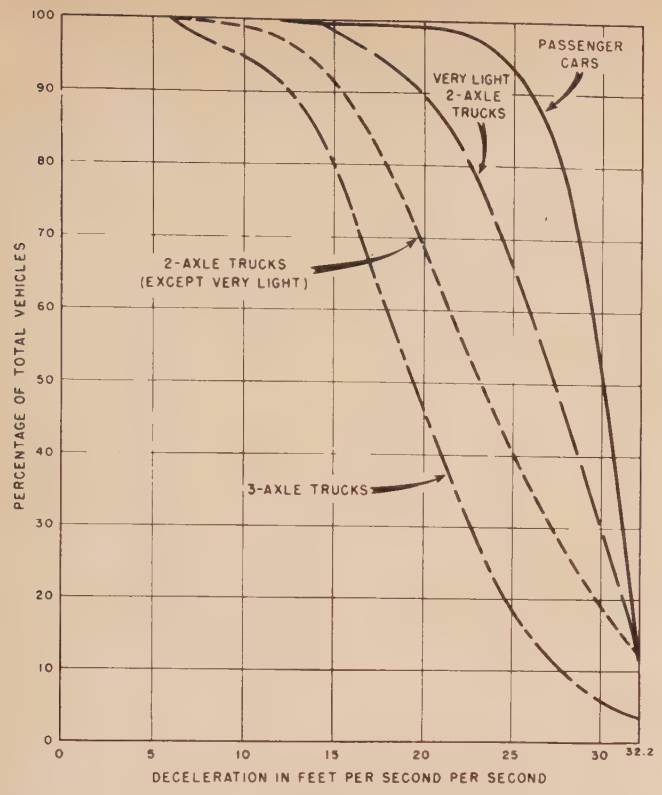


Fig. 12—Percentages of single-unit vehicles capable of given or greater deceleration

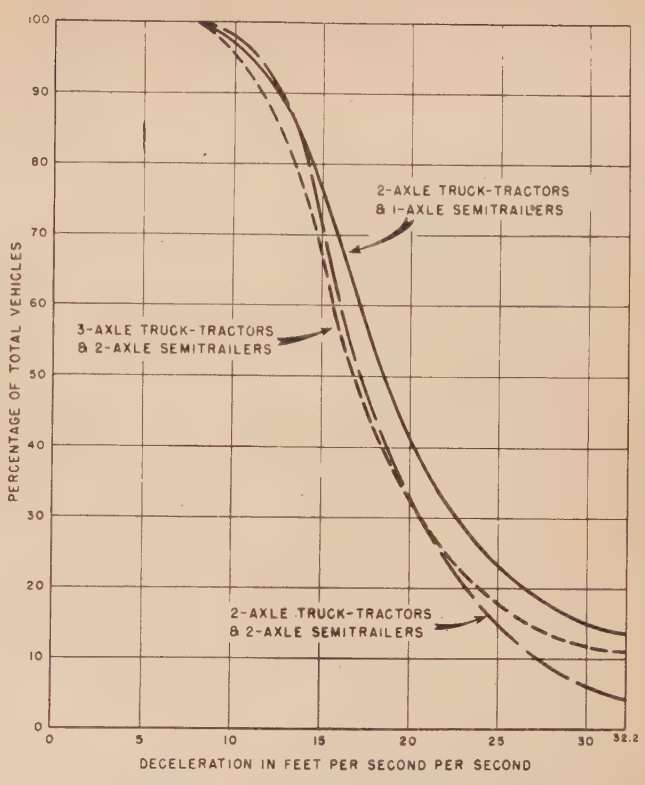


Fig. 13—Percentages of truck-tractors with semitrailers capable of given or greater deceleration

1956 cars. Although the 1947, 1941, and 1937 model passenger cars were the poorest performers, on the average they required only 24 ft to stop. Since the sample was quite small for any one year model of passenger cars of 1948 vintage and older, the results doubtless do not precisely reflect the stopping ability of each individual year model of the vehicles in this group. However, the total sample of these older vehicles was large enough to produce reliable results when considered as a group. The average brake-system application and braking distance for the 1948 model and older passenger cars was 23 ft from 20 mph.

Deceleration—Although measurement of brake-system application and braking distance from a known speed provides the most reliable indication of a vehicle's braking ability, it is also significant to consider maximum deceleration, since brake performance enforcement in many jurisdictions is based entirely on measurement of deceleration. The curves shown in Figs. 12, 13, and 14 indicate the decelerating abilities of the various types of vehicles tested. It should be kept in mind that the decelerations measured were not sustained throughout the stops but were the maximum decelerations indicated during the stops.

Each curve shown in the figures indicates the percentage of vehicles of a particular type which reached a maximum deceleration of a given or greater value. For example, Fig. 12 shows that 50% of each of the vehicle types represented by the four curves were capable of the following or greater decelerations,

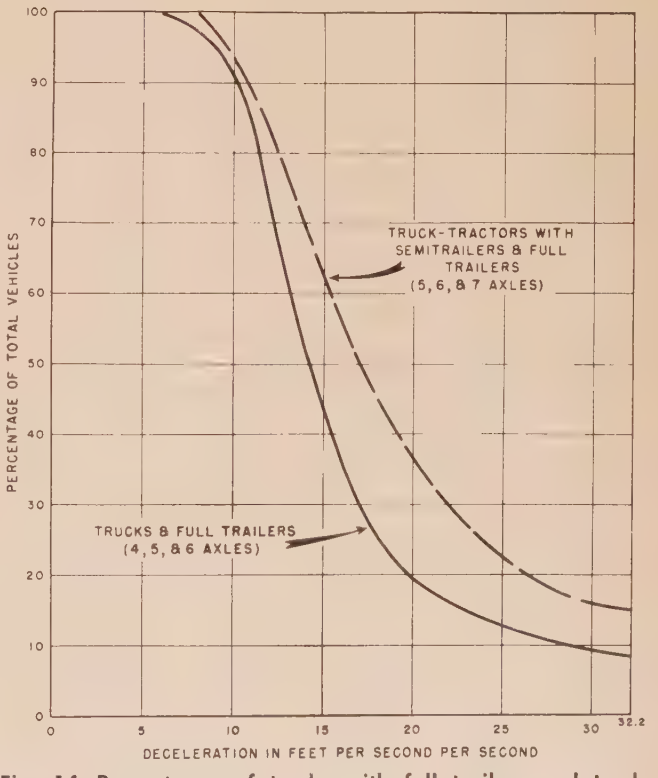


Fig. 14—Percentages of trucks with full trailers and truck-tractors with semitrailers and full trailers capable of given or greater deceleration

Table 4—Comparison of Brake-System Application and Braking Distances from 20 mph for Various Vehicle Types Tested in 1942, 1949, and 1955

Vehicle Type	Brake-System Application and Braking Distance, ft								
	15-Percentile			50-Percentile			85-Percentile		
	1942	1949	1955	1942	19 9	1955	1942	19	1955
Passenger Cars	21	17	18	25	21	20	42	26	23
2-Axle Trucks	28	23	22	40	28	29	64	41	39
3-Axle Trucks	40	26	30	54	37	39	81	56	54
2-Axle Truck-Tractors with 1-Axle Semitrailers	38	30	31	52	39	40	75	53	52
2-Axle Truck-Tractors with 2-Axle Semitrailers	43	34	35	61	43	42	83	60	52
3-Axle Truck-Tractors with 2-Axle Semitrailers	49	42	33	61	53	46	82	69	57
Trucks with Full Trailers (4, 5, and 6 Axles)	48	39	39	65	52	54	87	70	66
Truck-tractors with Semitrailers and Full Trailers (5, 6, and 7 Axles)	47	45	41	62	59	53	82	73	67

expressed in feet per second per second: passenger cars, 30.1; very light 2-axle trucks, 27.4; other 2-axle trucks, 23.2; and 3-axle trucks, 19.4. Similarly, Fig. 13 shows that 50% of each of three types of truck-tractor-semitrailer combinations were capable of the following or greater decelerations, expressed in feet per second per second: 2-axle truck-tractors with 1-axle semitrailers, 18.5; 2-axle truck-tractors with 2-axle semitrailers, 17.2; and 3-axle truck-tractors with 2-axle semitrailers, 16.8. Fig. 14 shows that 50% of the trucks with full trailers and truck-tractors with semitrailers and full trailers were capable of decelerations of at least 14.2 and 17.1 fps per sec, respectively. It is evident from Figs. 12, 13, and 14 that there is a wide range in the decelerating ability of the various types of vehicles, just as it is evident from Figs. 9 and 10 that there is a wide range in their brake-system application and braking distances. It is also apparent that the smaller vehicles are capable of the greater decelerations.

Performance Level Changes Since 1942 and 1949

Between 1942 and 1949 there was considerable improvement in the brake performance of all types of motor vehicles selected from the general traffic. However, between 1949 and 1955 the improvement in brake performance of the various types of vehicles, with two exceptions, was quite small. Table 4 summarizes the 15-, 50-, and 85-percentile values of brake-system application and braking distance for stops made from 20 mph by the various vehicle types tested in 1942, 1949, and 1955. In studying this general summary it must be kept in mind that the sample of vehicles of each type is made up of various gross weights, capacities, brake types, and conditions of maintenance. It will be noted from the table that there was appreciable improvement between 1942 and 1949 in the braking performance at all three levels for each of the vehicle types tested. The greatest improvement, as would be expected, was at the 85-percentile level. Although the improvement at the 50- and 15-percentile levels was progressively smaller, it was still appreciable for most vehicle types.

Between 1949 and 1955 there was again some improvement at the 85-percentile level in the brake performance of each vehicle type even though it was very small for some types. However, at the 50- and 15-percentile levels the performance of some vehicle types in 1955 was slightly poorer than it was in 1949. Only two vehicle types showed continuous improvement at all three performance levels. The truck-tractor-semitrailer-full-trailer combinations showed moderate improvement, and the 3-axle truck-tractors with 2-axle semitrailers showed appreciable improvement at all three performance levels in both 1949 and 1955. Of course, it is evident that there was more room for improvement in the case of these two types.

The improvement in the 3-axle truck-tractors with 2-axle semitrailers is illustrated throughout the range of performance levels by comparing the cumulative frequency distribution curves, shown in Fig. 15, for the 1955 tests with similar curves plotted from data obtained by the tests conducted in 1949 and 1942. One of the most revealing observations to be made from these curves concerns the percentages of vehicles tested which could stop within 50 ft—the present standard of the Uniform Vehicle Code for combinations of commercial vehicles. In 1942, only 17% of these vehicles could stop within 50 ft, but in 1949, 36% could stop in 50 ft, and in 1955, 64% could stop within 50 ft or less. Between 1949 and 1955 this vehicle type showed greater increase than any other type in percentage of vehicles capable of complying with the present requirement of the Uniform Vehicle Code. Such improvement in the performance of this vehicle type leads to the belief that there are excellent opportunities for considerable improvement in the braking performance of other commercial vehicle types.

If we consider the ability of the various vehicle types to comply with brake performance regulations, the improvements between 1949 and 1955 appear a little more encouraging than indicated by Table 4. Table 5 shows by vehicle type the percentage of

Table 5—Vehicles Tested in 1949 and 1955 Capable of Meeting Uniform Vehicle Code Requirement, as Revised in 1954, for Brake-System Application and Braking Distance from 20 mph

Vehicle Type	Requirement for Brake-System Application and Braking Distance, ft	Vehicles Tested Capable of Meeting Requirement, %	
		1949	1955
Passenger Cars	25	90	92
Very Light 2-Axle Trucks	30	76	84
2-Axle Trucks other than Very Light	40	84	84
3-Axle Trucks	50	78	80
2-Axle Truck-Tractors with 1-Axle Semitrailers	50	81	81
2-Axle Truck-Tractors with 2-Axle Semitrailers	50	73	80
3-Axle Truck-Tractors with 2-Axle Semitrailers	50	36	64
Trucks with Full Trailers (4, 5, and 6 Axles)	50	44	38
Truck-Tractors with Semitrailers and Full Trailers (5, 6, and 7 Axles)	50	22	41

vehicles tested, both in 1949 and 1955, which could meet the present Uniform Vehicle Code requirement for brake-system application and braking distance for stops made from 20 mph. In studying the results presented in this table, it should be kept in mind that the requirement varies for the different vehicle types. For example, the requirement for very light 2-axle trucks (those having a manufacturer's gross vehicle weight rating of 10,000 lb or less) is 30 ft; for 2-axle trucks other than very light, 40 ft; and for all other types of commercial vehicles, 50 ft.

As previously described the greatest increase in percentage of vehicles capable of meeting the requirement was made by the 3-axle truck-tractors with 2-axle semitrailers. The next greatest increase was made by the truck-tractors with semitrailers and full trailers. They improved from 22 to 41% able to meet the requirement—an increase of 19 percentage points. Moderate increases in the per cent of vehicles able to comply with their respective requirements were made by the very light 2-axle trucks and the 2-axle truck-tractors with 2-axle semitrailers, which gained eight and seven percentage points, respectively.

The passenger cars and the 3-axle trucks each showed a small improvement of two percentage points, while the 2-axle trucks having a manufacturer's gross vehicle weight rating of over 10,000 lb and the 2-axle truck-tractors with 1-axle semitrailers showed no change. The only vehicle type which lost ground with respect to its ability to meet the present Uniform Vehicle Code requirement for brake-system application and braking distance was the truck-full-trailer combination. In 1949, 44% of the sample of this vehicle type could comply, but in 1955, only 38% could comply.

It was not possible to compare the decelerating abilities of vehicles tested in 1949 and 1955, because deceleration was not measured in the 1949 studies of vehicles selected from the general traffic. However, it is of particular interest to consider the ability of each vehicle type tested in 1955 to comply with the Uniform Vehicle Code specification for deceleration. Table 6 shows the per cent of each type of vehicle tested in 1955 which could meet its respective requirement for deceleration. The deceleration specified for passenger cars is 17 fps per sec, while for all commercial vehicle types it is 14 fps per sec.

Well over 99% of the passenger cars, all of the very light 2-axle trucks, and almost 94% of the other 2-axle trucks were able to meet their respective requirements for deceleration when measured with a commercially available portable decelerometer. From 76 to 85% of the truck-tractor-semitrailer combinations and 3-axle single-unit trucks could comply, and 51 to 69% of the trucks with full trailers and truck-tractors with semitrailers and full trailers could comply with the 14-fps per sec specification for deceleration.

If the results shown in Table 6 are compared with those shown in Table 5, it will be noted that a greater percentage of vehicles of each type could comply with the Uniform Vehicle Code specification for deceleration than could comply with the corresponding specification for brake-system application and braking distance. However, in comparing the two tables, it should be kept in mind that the specifications for deceleration and for brake-system application and braking distance correspond to each other mathematically only for passenger cars and very light 2-axle trucks.

In making the foregoing comparisons of the performance of commercial vehicles tested in 1942, 1949, and 1955, the validity of the amount of improvement shown may be questionable without some consideration of the weight of the vehicles in each sample of data. In other words, if the weights of the 3-axle truck-tractors with 2-axle semitrailers were considerably lighter in 1955 than in 1949, the im-

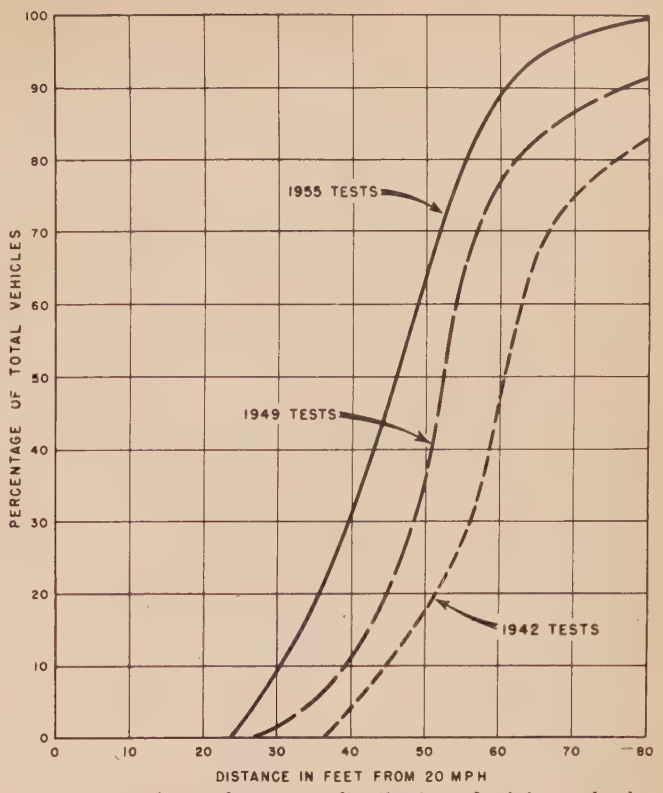


Fig. 15—Cumulative frequency distribution of minimum brake-system application and braking distances of 3-axle truck-tractors with 2-axle semitrailers selected from general traffic

Table 6—Vehicles Tested in 1955 Capable of Meeting Uniform Vehicle Code Requirement for Deceleration

Vehicle Type	Requirement for Deceleration, fps per sec	Vehicles Tested in 1955 Capable of Meeting Requirement, %
Passenger Cars	17	99.7
Very Light 2-Axle Trucks	14	100.0
2-Axle Trucks other than Very Light	14	93.9
3-Axle Trucks	14	85.0
2-Axle Truck-Tractors with 1-Axle Semitrailers	14	82.7
2-Axle Truck-Tractors with 2-Axle Semitrailers	14	81.7
3-Axle Truck-Tractors with 2-Axle Semitrailers	14	76.0
Trucks with Full Trailers (4, 5, and 6 Axles)	14	51.2
Truck-Tractors with Semitrailers and Full Trailers (5, 6, and 7 Axles)	14	69.4

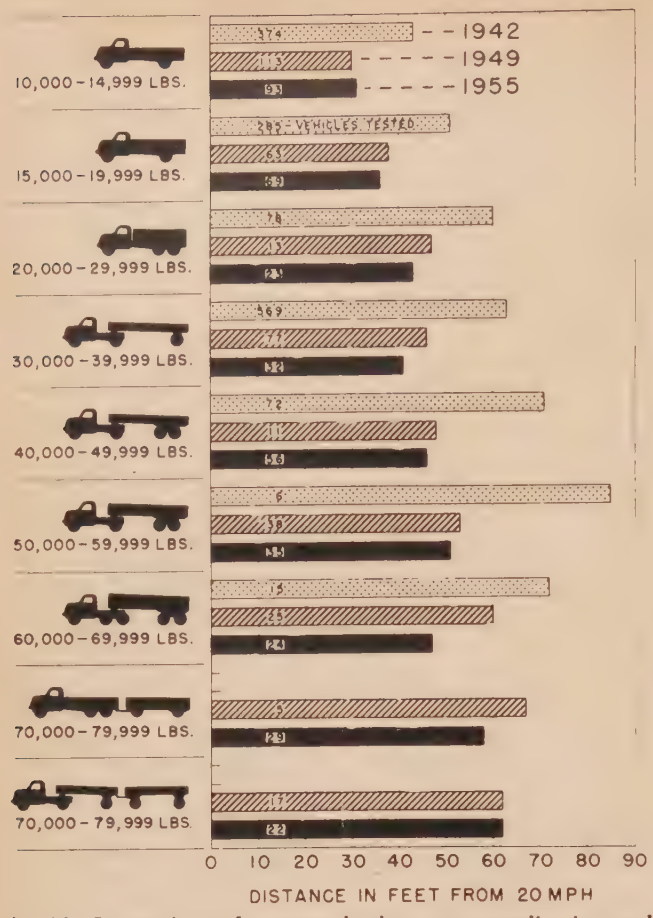


Fig. 16—Comparison of average brake-system application and braking distances observed in 1942, 1949, and 1955 for commercial vehicles loaded to commonly carried gross weights

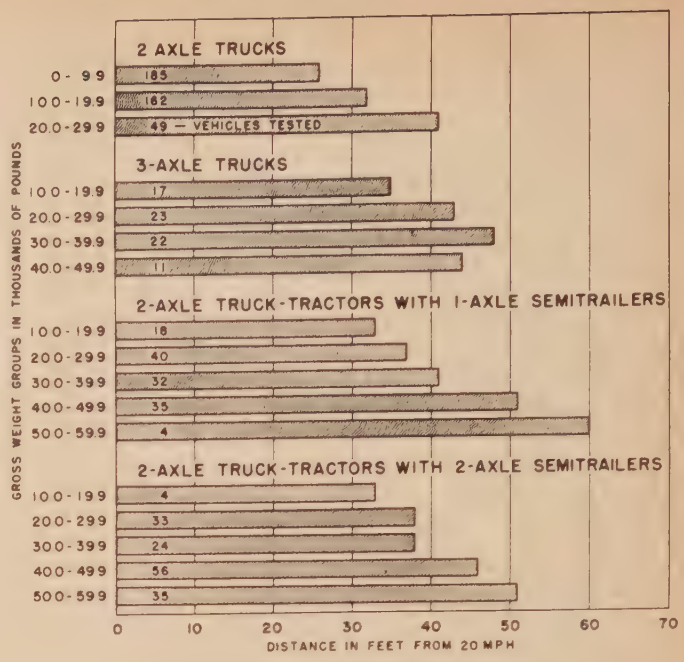


Fig. 17—Variation of average brake-system application and braking distance with gross weight for 2- and 3-axle trucks and for 3- and 4-axle truck-tractor-semitrailer combinations

provement might be attributed to weight rather than to the braking system. From the tabulation of average gross weights for the various types of vehicles, in Table 7, it is seen that the average weight was generally higher in 1949 than in 1942, and in 1955 it was generally higher than in 1949. However, the reverse is true of the 2-axle trucks. The average weight of the 2-axle trucks tested was progressively lighter in 1949 and 1955.

The decrease in average gross weight of the 3-axle truck-tractors with 2-axle semitrailers in 1955 may possibly account for a small part of their improvement in stopping ability. However, since the weight decrease was relatively small, only 3200 lb, which is less than 6%, the greater part of the improvement may be attributed to more efficient braking systems.

An appreciable increase in the average gross weight of the truck-full-trailer combinations, particularly those having five or six axles, can at least partly explain the poorer showing of these vehicles with respect to their ability to meet the present standard of the Uniform Vehicle Code. However, results of the controlled brake tests of commercial vehicles, which were reported in 1954,[1] indicate that considerable improvement can be made in the stopping ability of these vehicle combinations. The average weight increases for the four types of truck-full-trailer combinations were as follows: 2-axle truck with 2-axle trailer, 2800 lb; 2-axle truck with 3-axle trailer, 11,400 lb; 3-axle truck with 2-axle trailer, 17,000 lb; and 3-axle truck with 3-axle trailer, 14,000 lb. These increases amount to 6½, 28, 36, and 28%, respectively. It is of interest that the 3-axle truck with 2-axle full trailer showed the greatest increase in average gross weight of any of the vehicle types tested and, also, was by far the predominating truck-full-trailer type encountered in the 1955 tests.

The 3-axle single-unit trucks tested in 1955 were, on the average, 7300 lb heavier than those tested in 1949. This increase in average weight, which was approximately 35%, was undoubtedly a factor in their not showing greater improvement in ability to meet a 50-ft requirement for stopping ability. It will be recalled the 3-axle trucks showed improvement of only two percentage points in ability to meet such a standard.

Fig. 16 shows by vehicle type the improvement which has been made in vehicles of approximately the same weight. However, it should be pointed out that this figure does not consider vehicle capacities. The vehicles of each type were first classified into weight groups with a range of 5000 lb for the 2-axle trucks and 10,000 lb for the other types. Weight groups, which were representative of each of the seven most frequent commercial vehicle types tested, were selected. An average brake-system application and braking distance was then computed for each group and year of study. No results for 3-axle trucks with 2-axle full trailers or 2-axle truck-tractors with 1-axle semitrailers and 2-axle full trailers tested in

1942 appear in the figure because of the small size of the sample.

Fig. 16 confirms that the improvement in stopping ability of the various types of vehicles between 1942 and 1949 was quite large. It also confirms that, for most vehicle types, there was not much improvement between 1949 and 1955. In addition the figure indicates that, when compared on an equal-weight basis, the heavily loaded 2-axle truck-tractors with 1-axle semitrailers and 2-axle full trailers showed no change in stopping ability since 1949, but the heavily loaded 3-axle trucks with 2-axle trailers showed improvement. It will be recalled that the truck-full-trailer combinations as a group have previously been singled out as the only vehicle type to lose ground with respect to its ability to meet the standard for stopping ability set forth in the Uniform Vehicle Code.

Table 8 shows that the per cent improvement based on values indicated in Fig. 16, although not always identical, was similar to that determined for vehicles grouped according to capacity. The capacity group composed of the greatest number of vehicles was selected for each of six of the vehicle weight groups considered in Fig. 16. It is of particular importance to note that in every case where there was positive improvement indicated for vehicles grouped only according to weight, the same was true for the vehicles within that weight group when also grouped according to capacity. Likewise, when no improvement was indicated for vehicles grouped only according to weight, no improvement was indicated for the vehicles within that weight group when also grouped according to capacity. The foregoing discussion of Table 8 should not be interpreted as indicating that vehicle capacity has no effect on brake performance.

Gross Weight Relations

In order to demonstrate how brake performance varies with vehicle type, capacity, and brake type, it was necessary to establish a relation between weight and brake-system application and braking distance. This was accomplished by classifying vehicles of selected type and capacity into weight groups and computing the average distance for each weight group.

The relation between weight and distance is shown in Figs. 17 and 18 by types of vehicles, without respect to the manufacturers' ratings of capacity. So far as normal highway operation is concerned, brake-system application and braking distance increases with weight for a given type of vehicle. It is of significance to compare the average performance of the most common weight groups for each type of vehicle, as it is the performance of the vehicles with loads normally carried that must be given primary consideration in both the fields of regulation and highway design.

The average brake-system application and braking distances indicated in Figs. 17 and 18 are summarized in Table 9 by vehicle type for the gross weight groups having the greatest frequency of vehicles. These results emphasize the wide variation of stopping ability for the various types of commercial vehicles operating with their most frequently carried loads. It is significant that the range of average brake-system application and braking distances for these vehicle groups extends from 26 ft for the 2-axle trucks with a gross weight of less than 10,000 lb to 59 ft for the

Table 7—Average Gross Vehicle Weights of Commercial Vehicles Tested in 1942, 1949, and 1955

Vehicle Type[a]	Number of Vehicles and Average Gross Vehicle Weight					
	1942		1949		1955	
	Number	Weight, lb	Number	Weight, lb	Number	Weight, lb
2	1231	13,600	367	12,200	400	11,800
3	172	24,700	46	21,100	73	28,400
2-S1	1344	28,500	246	30,400	129	32,100
2-S2	186	35,100	98	41,500	153	40,400
3-S2	59	48,800	57	56,900	66	53,700
2-2	30	30,100	9	43,100	16	45,900
2-3	15	31,300	2	40,800	1	52,200
3-2	14	47,800	16	46,900	46	63,900
3-3	144	59,500	27	50,000	7	64,000
2-S1-2	14	39,400	50	58,100	44	59,700
2-S2-2	11	56,300	9	63,400	7	62,200
2-S2-3	6	53,100	1	94,900	2	52,000

[a] See section entitled, "Vehicle Type Code," for explanation of the code used.

Table 8—Improvement in Stopping Ability of Commercial Vehicles of Approximately the Same Gross Weight when Grouped without Respect to Vehicle Capacity and When Grouped According to Manufacturers' Vehicle Capacity Ratings

Vehicle Type[a]	Grouped Only According to Gross Weight: Weight Group, thousands of lb	Feet to stop from 20 mph: 1949	1955	Improvement: ft	%	Grouped According to Vehicle Capacity and Gross Weight: Capacity Group	Feet to stop from 20 mph: 1949	1955	Improvement: ft	%
2	10.0-14.9	30	31	−1	−3	Light	30	31	−1	−3
2	15.0-19.9	38	36	2	5	Light	42	37	5	12
2-S1	30.0-39.9	46	41	5	11	Medium	48	44	4	8
2-S2	50.0-59.9	53	51	2	4	Medium	54	51	3	6
3-S2	60.0-69.9	57	47	10	18	Heavy	56	46	10	18
2-S1-2	70.0-79.9	62	62	0	0	Heavy	62	62	0	0

[a] See section entitled, "Vehicle Type Code," for explanation of the code used.

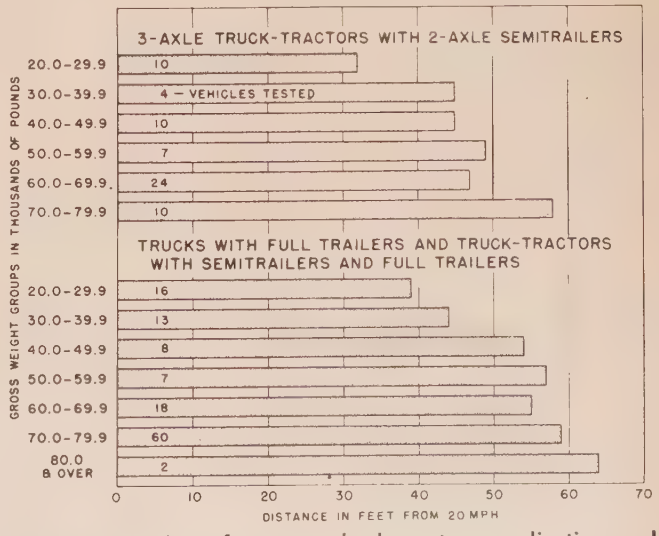


Fig. 18—Variation of average brake-system application and braking distance with gross weight for larger commercial vehicle combinations

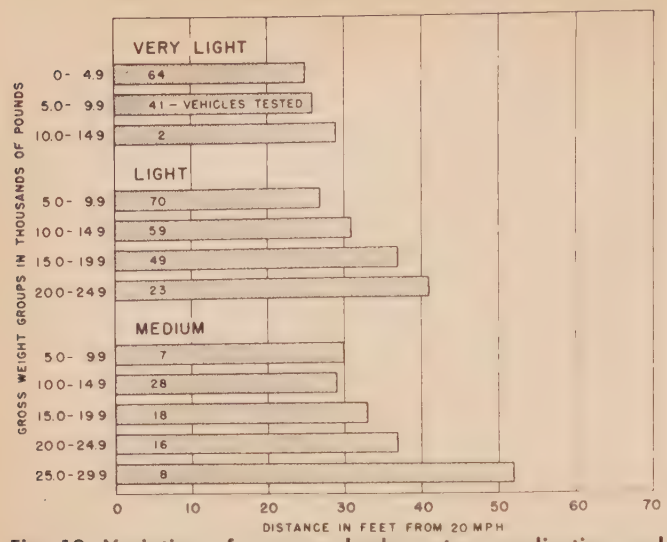


Fig. 19—Variation of average brake-system application and braking distance with gross weight by capacity groups for 2-axle trucks

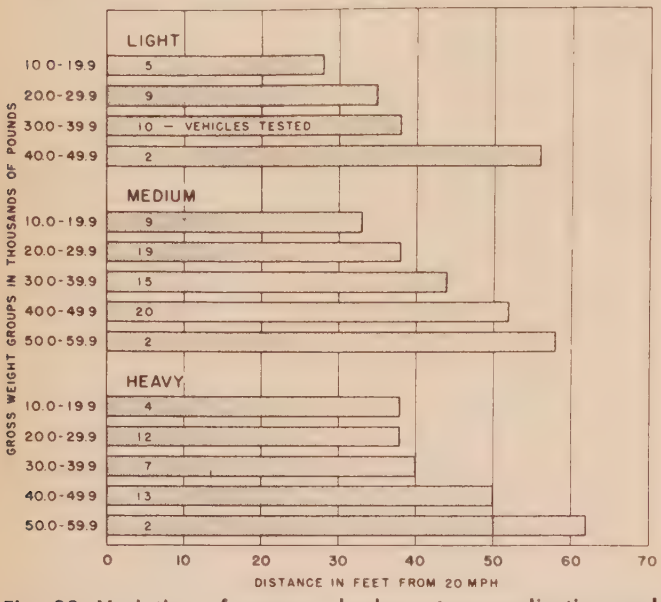


Fig. 20—Variation of average brake-system application and braking distance with gross weight by capacity groups for 2-axle truck-tractors with 1-axle semitrailers

trucks with full trailers and the truck-tractors with semitrailers and full trailers with a gross weight of 70,000 to 79,999 lb—a variation in stopping ability of 127%.

In Fig. 19 the performance of 2-axle trucks is compared by capacity groups. Results of tests on the heavy-capacity group 2-axle trucks were not included because the sample was too small. Again, it is seen that the distance required to stop increases with vehicle weight, as previously indicated for the vehicles not segregated according to their rated capacities. However, in examining Fig. 19 it is seen that the distance required to stop by the light-capacity 2-axle trucks in each of the three weight groups, 10,000–14,999 lb, 15,000–19,999 lb, and 20,000–24,999 lb

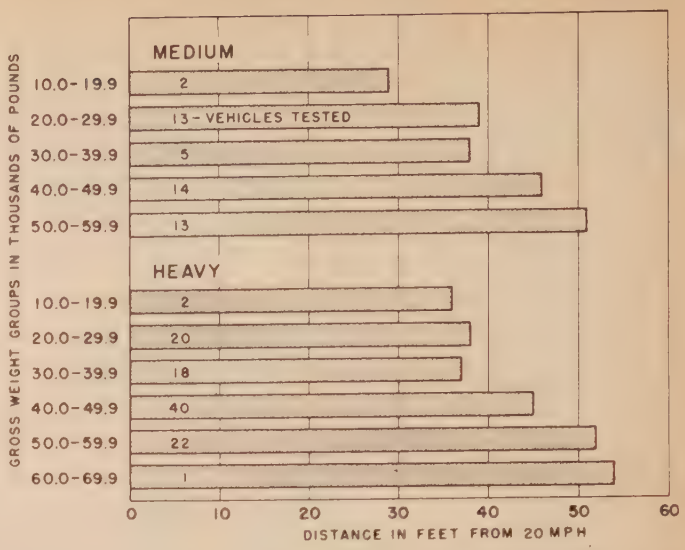


Fig. 21—Variation of average brake-system application and braking distance with gross weight by capacity groups for 2-axle truck-tractors with 2-axle semitrailers

was greater than that required by the medium-capacity vehicles in the corresponding weight group. Further examination of the figure reveals that the stopping ability for each weight group, except the heaviest of the medium-capacity 2-axle trucks, very closely approximates the stopping ability for the next lighter weight group of the light-capacity vehicles. Thus, in view of the difference in rating between the two capacity groups indicated in Table 3, it appears that the light- and the medium-capacity 2-axle trucks are rated about on a par with respect to their brake performance.

The average braking performances of light, medium, and heavy 2-axle truck-tractors with 1-axle semitrailers are shown in Fig. 20 in a similar manner. In contrast to the results shown for the 2-axle trucks, the average brake-system application and braking distance increased with an increase in capacity. This is evident since the stopping ability for a common weight group does not decrease appreciably in any instance with capacity. For example, for the 30,000–39,999-lb group the distances are 38, 44, and 40 ft for the light-, medium-, and heavy-capacity vehicles, respectively.

Also, the stopping ability for the light 20,000–29,999-, medium 30,000–39,999-, and heavy 40,000–49,999-lb groups, which may be consided to be representative of the average rating for the given capacity class, was 35, 44, and 50 ft, respectively. In general, for a common weight group the differences in brake performance of the medium- and the heavy-capacity units were small.

In Fig. 21 the braking performance of medium and heavy 2-axle truck-tractors with 2-axle semitrailers is compared. It is significant that for a given weight group—except for the 10,000–19,999-lb group composed of essentially empty vehicles—the average

distance required to stop varied with capacity group by not more than 1 ft. Thus, in considering the performance of both 2–S1 and 2–S2 truck-tractor-semitrailer combinations, it appears that little difference exists in the braking ability of medium and heavy 2-axle truck-tractors of approximately the same gross weight.

It has been demonstrated in Figs. 19, 20, and 21 how brake-system application and braking distances of the several vehicle types vary with gross weight and with rated capacity. In the case of the 2-axle trucks, it was indicated that weights beyond the rating of the light units could be hauled in a larger capacity vehicle with an improvement in level of brake performance for the the vehicle type as a whole. This was not true for combination units where vehicle capacity was not a significant factor. In fact, the loading of either a 2–S1 or 2–S2 of any given capacity group beyond its rated gross load would not materially affect the overall level of performance for that particular type.

Figs. 22, 23, 24, and 25 compare the braking performance of different commercial vehicle types in each of three capacity groups. However, because the sample of some vehicle types in certain capacity groups was not adequate for distribution into weight groups, not every vehicle type is considered in each capacity group.

Fig. 22 compares the stopping ability of the light-capacity 2-axle trucks and 2-axle truck-tractors with 1-axle semitrailers. If the stopping ability of these two vehicle types is compared for identical weight groups, the figure indicates the better performance for the combination vehicles. Because the light single-unit trucks and the power units of the light vehicle combinations are basically the same vehicles, the ability of the combination vehicles to stop in the shorter distance, when compared on an equal-weight basis, may be attributed to the added braking provided by the semitrailer.

However, since the same basic vehicle is rated for a heavier gross weight when used in combination, it is more logical to compare the performance of the 2-axle trucks weighing 10,000–19,999 lb with that of the 2-axle truck-tractors with 1-axle semitrailers weighing 20,000–29,999 lb. It will be recalled that Table

Table 9—Average Brake-System Application and Braking Distance from 20 mph by Vehicle Type for Weight Groups having greatest Frequency of Vehicles

Vehicle Type	Vehicle Weight Group, thousands of lb	Number of Vehicles in Weight Group	Average Brake-System Application and Braking Distance, ft
2-Axle Trucks	0—9.9	185	26
	10.0—19.9	162	32
3-Axle Trucks	20.0—29.9	23	43
	30.0—39.9	22	48
2-Axle Truck-Tractors with 1-Axle Semitrailers	20.0—29.9	40	37
	30.0—39.9	32	41
	40.0—49.9	35	51
2-Axle Truck-Tractors with 2-Axle Semitrailers	40.0—49.9	56	46
	50.0—59.9	35	51
3-Axle Truck-Tractors with 2-Axle Semitrailers	60.0 69.9	24	47
Trucks with Full Trailers and Truck-Tractors with Semitrailers and Full Trailers	70.0 79.9	60	59

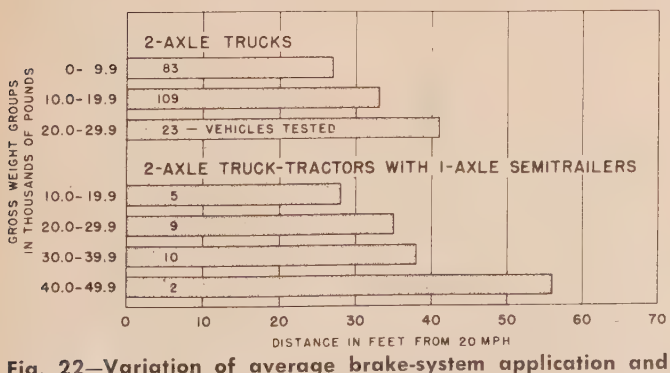


Fig. 22—Variation of average brake-system application and braking distance with gross weight for light-capacity trucks and combination vehicles

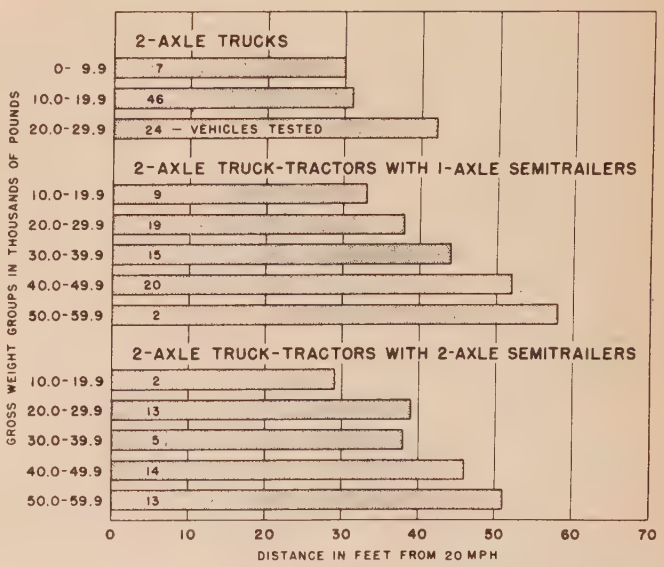


Fig. 23—Variation of average brake-system application and braking distance with gross weight for medium-capacity trucks and combination vehicles

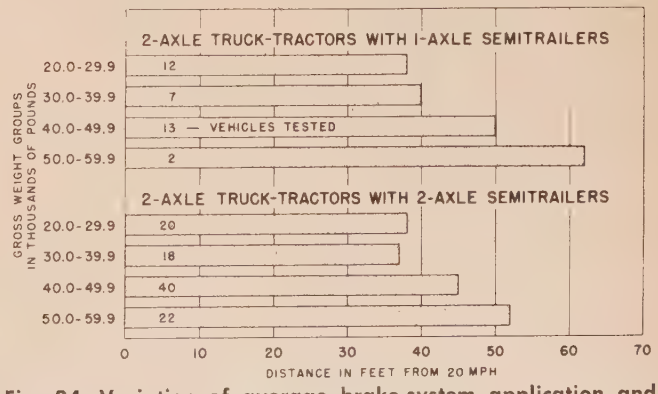


Fig. 24—Variation of average brake-system application and braking distance with gross weight for heavy-capacity 3- and 4-axle truck-tractor-semitrailer combinations

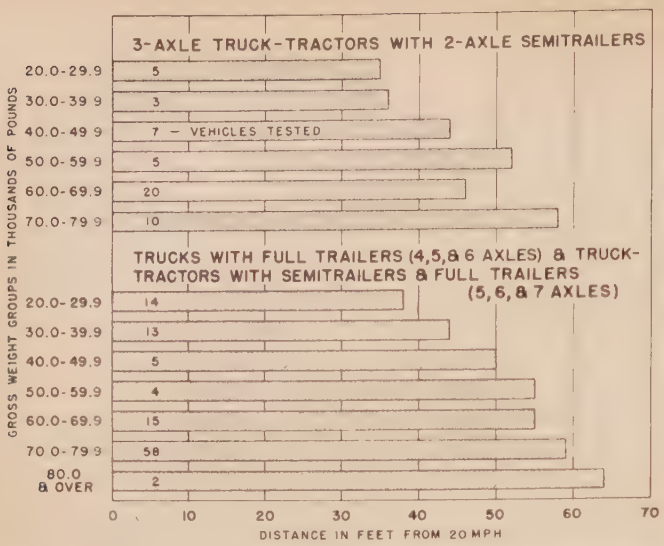


Fig. 25—Variation of average brake-system application and braking distance with gross weight for larger heavy-capacity vehicle combinations

3 defines the light-capacity vehicles as those having a manufacturer's gross vehicle weight rating between 10,000 and 16,000 lb or gross combination weight rating of 27,000 lb or less. In comparing the brake performance of the two vehicle types on this basis, Fig. 22 shows slightly better performance for the single-unit vehicles. On the average, these 2-axle trucks required 33 ft to stop from 20 mph, while the group of 2-axle truck-tractors with 1-axle semitrailers required 35 ft to stop.

Fig. 23 compares the stopping ability of medium-capacity 2-axle trucks, 2-axle truck-tractors with 1-axle semitrailers, and 2-axle truck-tractors with 2-axle semitrailers. In comparing the medium-capacity vehicles, it is appropriate to compare the performance of the single-unit vehicles weighing 20,000–29,999 lb with that of the combination vehicles weighing 40,000–49,999 lb since these weight groups bracket the upper limits of the respective medium-capacity ratings in Table 3. In comparing the vehicles on this basis, the figure shows that on the average the 2-axle trucks required 42 ft to stop from 20 mph, while the 2-axle tractors with 1-axle semitrailers required 52 ft and the 2-axle truck-tractors with 2-axle semitrailers required 46 ft. Again, as in the case of the light-capacity vehicles, the 2-axle trucks showed the better performance.

The somewhat poorer performance of combination vehicles may largely be attributed to the inherent characteristics of the braking systems. A certain amount of additional time is required to transmit power to and to build up force in the trailer brakes. During this time the vehicle combination is traveling faster than at any other time during the stop. Therefore, for a given increment of time it travels farther, and the brake-system application and braking distance is bound to be stretched out somewhat. In the best performing vehicles the application and brake force buildup time is held to a minimum.[1]

In comparing the performance of the two medium-capacity truck-tractor-semitrailer types considered in Fig. 23, it is seen that the average brake-system application and braking distance of each weight group of the combinations with single-axle semitrailers was approximately equal to that of the next heavier weight group of the combinations with tandem-axle semitrailers. For example, the 2–S1 combinations weighing 20,000–29,999 lb and the 2–S2 combinations weighing 30,000–39,999 lb on the average required the same distance to stop—38 ft. Fig. 24 indicates the same general relation for the heavy-capacity vehicles of these two types. Thus for vehicles within the same capacity group, it appears that the brake performance of a 2-axle truck-tractor with 2-axle semitrailer of a given gross weight is generally on a par with the performance of a 2-axle truck-tractor with 1-axle semitrailer which is approximtaely 10,000 lb lighter.

In Fig. 25 the stopping ability of heavy-capacity 3-axle truck-tractors with 2-axle semitrailers is compared with that of heavy-capacity truck-full-trailer and truck-tractor-semitrailer-full-trailer combinations. It is important to note that in each weight group the 3–S2 combinations required less distance to stop than the other vehicle combinations in the corresponding weight group.

Considering the results shown in Fig. 25, and those shown in Figs. 23 and 16, it appears that brake-system application and braking distance, for vehicles within a given capacity group, increases with the number of vehicles in combination. The reason for this is the same as that previously given to explain why truck-tractor-semitrailer combinations require more distance to stop than comparable single-unit vehicles. When an additional vehicle is included in a combination, additional time is required to transmit power to, and to build up full braking force in some of the brakes of the combination. This is due primarily to the longer lines and added connections involved in triggering the power to operate the brakes. Of course, longer application and buildup time means longer brake-system application and braking distance.

Brake Performance for Common Gross Weights

In order to define the practical levels of brake performance for vehicles currently using the highways, the average, 15-, and 85-percentile values of brake-system application and braking distance observed for vehicles as commonly loaded are summarized in Table 10 by vehicle type, capacity, and weight range. Most of the previous analyses have been concerned only with the average performance level for a given group of vehicles. In this table the 15- and 85-percentile values are included to indicate the spread of data about the average.

The 15-percentile values represent the performance

of the best 15% of the vehicles in a group; and the 85-percentile values, the performance attained by 85% of the vehicle (and not attained by the worst 15%). The 85-percentile value may be useful for highway design purposes, whereas the 15-percentile value will give some idea of what can be expected in the future.

A significant point to be made from this summary is the need for a concentrated effort to bring the brake performance of the truck-tractors with semitrailers and full trailers, the trucks with full trailers, and the heaviest groups of truck-tractors with semitrailers into line with the performance of the other groups of truck-tractors with semitrailers. These results should also prove beneficial in testing the practicability of minimum regulatory requirements, and in developing highway design factors. The 15-percentile values show definitely that considerable improvement is possible for each type of vehicle.

Axle-Weight Relations

The previous analyses of brake-system application and braking distances have been made either directly or indirectly with respect to gross vehicle weight. The following discussion considers the relation existing between axle load and brake-system application and braking distance for 2-axle trucks and 2-axle truck-tractors with 1-axle semitrailers. It was not possible to consider other vehicle types since the number of vehicles of a given type was either too small, or the typical axle load distribution for a given type of vehicle was such that all axle loads were not approximately the same.

In order to establish the relation, the vehicles were first classified by axle-weight groups with a range of 4000 lb. Only the principal load-carrying axles were considered; the weight of the front axle was disregarded. If both axles of a truck-tractor-semitrailer combination did not fall in the same weight group, that particular vehicle was eliminated from consideration. Following the classification, an average brake-system application and braking distance was computed for each axle-weight group.

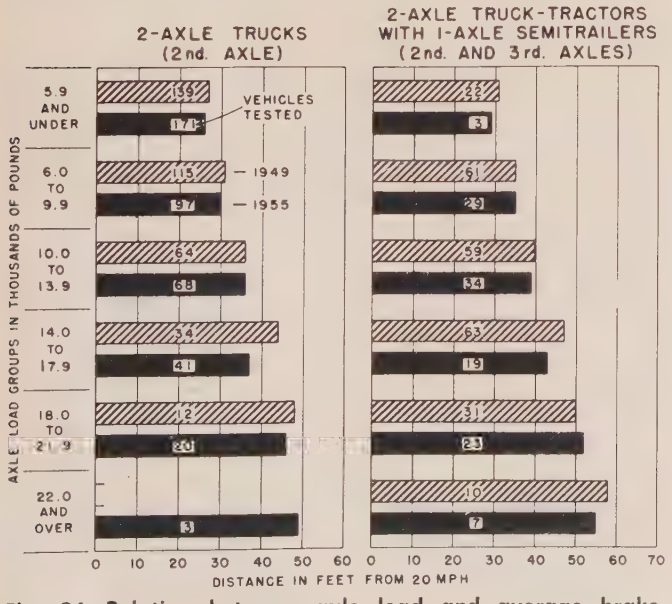


Fig. 26—Relation between axle load and average brake-system application and braking distance for trucks and combination vehicles

Table 10—Summary of Brake-System Application and Braking Distances from 20 mph for Commercial Vehicles with Gross Weights Commonly Carried on Highway

Vehicle Type[a]	Capacity Group	Weight Group, thousands of lb	Number of Vehicles	Distance, ft		
				Average	15-Percentile	85-Percentile
2	Very Light	0 — 4.9	61	25	20	30
		5.0— 9.9	33	26	22	30
	Light	10.0—14.9	60	31	24	37
		15.0—19.9	49	37	28	48
		20.0—24.9	23	41	29	48
	Medium	15.0 19.9	18	33	30	36
		20.0—24.9	16	37	31	43
		25.0—29.9	8	52	37	63
	Heavy	20.0—29.9	2	38		
		30.0 34.9	3	44		
3	Light	20.0—29.9	6	42		
	Medium	20.0—29.9	6	45		
		30.0—39.9	11	48	37	58
	Heavy	20.0—29.9	11	42	29	49
		30.0—39.9	10	47	34	53
		40.0—49.9	6	43		
2-S1	Light	20.0 -29.9	9	35	27	37
		30.0 39.9	10	38	29	45
	Medium	30.0—39.9	15	44	36	50
		40.0 49.9	20	52	38	62
	Heavy	30.0—39.9	7	40	33	45
		40.0—49.9	13	50	42	53
		50.0—59.9	2	62		
2-S2	Medium	40.0—49.9	14	46	36	54
		50.0 59.9	13	51	39	58
	Heavy	40.0—49.9	40	45	39	51
		50.0—59.9	22	52	41	59
3-S2	Heavy	60.0—69.9	20	46	37	54
		70.0—74.9	10	58	50	63
2-2	Heavy	60.0—64.9	5	56		
3-2	Heavy	70.0—79.9	29	58	45	67
3-3	Heavy	70.0—79.9	5	65		
2-S1-2	Heavy	70.0—79.9	20	62	48	74
2-S2-2	Heavy	75.0—84.9	4	59		

[a] See section entitled, "Vehicle Type Code," for explanation of code used.

The results are shown in Fig. 26 for vehicles tested in 1955 and 1949. So far as the general traffic is concerned, average brake-system application and braking distance for both years of study definitely increased with an increase in axle load. This would be expected, to some degree, since many vehicles were undoubtedly tested with either partial loads or overloads with respect to the stopping ability engineered into the vehicles. The figure also indicates that, except for the vehicles in one axle-weight group, there has been little change in the relation between brake performance and axle load from 1949 to 1955. Both types showed a moderate reduction in brake-system application and braking distance for the vehicles with axle loads of 14,000–17,999 lb. The improvement amounted to 7 ft for the 2-axle trucks and 4 ft for the 2-axle truck-tractors with 1-axle semitrailers.

The results presented in Fig. 26 are not to be construed as indicating that heavy axle loads cannot be

Table 11—Relation between Brake Type and Brake-System Application and Braking Distance for Light 2-Axle Trucks

Gross Vehicle Weight Group, lb	Number of Vehicles		Average Brake-System Application and Braking Distance from 20 mph, ft	
	Hydraulic Brakes	VBH Brakes[a]	Hydraulic Brakes	VBH Brakes[a]
5,000— 9,999	10	59	26	26
10,000—14,999	9	50	33	31
15,000—19,999	3	44	38	38
20,000—24,999		22		39

[a] VBH = Vacuum-booster hydraulic

adequately braked. The report published in 1954 [1] pointed out that a few of the vehicles tested in that study showed that axle loads commonly carried could be braked to the degree of wheel locking on dry concrete pavement with a relatively high coefficient of friction. One of several examples cited was a 2-axle truck-tractor with 1-axle semitrailer carrying 22,900 lb on the tractor drive axle and 22,800 lb on the trailer axle. This vehicle, which had had its brakes adjusted just prior to testing, stopped in 32 ft from 20 mph. Other heavily loaded vehicles tested in that study showed equally good brake performance.

A few of the vehicles tested in 1955 add further proof that axle loads now commonly carried can be adequately braked. It should be remembered that all of the vehicles tested in the present study were selected at random from the general traffic, and none of them had any special maintenance to prepare them for the tests. There were four examples of heavily loaded 2-axle truck-tractors with 1-axle semitrailers worth mentioning. The weights of the drive axle and trailer axle along with the brake-system application and braking distance for each of the four respective vehicles are as follows: (1) 21,000 and 22,800 lb—36 ft; (2) 19,500 and 23,800 lb—36 ft; (3) 22,500 and 22,100 lb—38 ft; and (4) 20,400 and 20,400 lb—38 ft. Results of the controlled tests reported in 1954 [1] indicate these vehicles could have done even better if they had had a brake adjustment immediately prior to the tests.

Regardless of what can be done, the fact remains that commercial vehicles from the general traffic—at least the two types considered in Fig. 26, and very likely all types—were not provided with braking ability in proportion to the loads carried on an axle. In view of the performance possible with current designs of braking systems, and in view of the performance of a few of the vehicles selected at random from the general traffic, the range of about 25 ft between the lightest and heaviest axle load group seems excessive. Some of the increase of brake-system application and braking distance with axle load might be attributed to longer application and buildup times, since the frequency of use of power brake systems increases with axle load. However, the magnitude of the difference between what can be done and what is generally being done suggests that insufficient attention is given to providing adequate braking force for the axle loads carried.

The existing condition, as revealed in Fig. 26, may be attributed to loading beyond the axle loads used in designing the brake components, inadequate maintenance of equipment, and obsolete design of the overall braking system. The operators have the responsibility of loading the axles in keeping with the braking force available, of properly maintaining the braking system, and of modernizing old equipment still being used on the highways. There is, also, a joint responsibility of both operators and manufacturers in the selection and provision of adequate braking equipment.

Brake-Type Relations

From the classification of vehicles in Table 2, it is apparent that comparisons of brake performance by brake type were possible for only the passenger cars, light 2-axle trucks, and medium 2-axle truck-tractors with 1-axle semitrailers. It was not possible to compare the stopping ability by brake type for other groups of vehicles because of the limited size of the sample and the nature of its distribution by vehicle type, capacity group, and brake type.

For passenger cars, the stopping ability of vehicles equipped with conventional hydraulic brakes was compared with the stopping ability of vehicles equipped with vacuum-booster hydraulic brakes, which are commonly called power brakes. Average brake-system application and braking distance was computed for each of the two groups of cars. Since the cars equipped with power brakes were predominantly 1953, 1954, 1955, and 1956 models, only those four year-models were included in the computations. The results showed that the average brake-system application and braking distance for the cars with hydraulic brakes was 20.5 ft, and for the cars with power (vacuum-booster hydraulic) brakes it was 21.5 ft.

The relation established between brake type and brake-system application and braking distance for commercial vehicles of the same type, capacity, and approximate weight is shown in Tables 11 and 12. The average performance of light 2-axle trucks with hydraulic and with vacuum-booster hydraulic brakes is compared in Table 11 for three weight groups. The results indicate slightly better performance for the vehicles with vacuum-booster hydraulic brakes in the 10,000–14,999-lb group, and identical performance for the two brake types in the other two weight groups.

Results of the 1949 tests indicated that a power booster on the brakes of light trucks in the 10,000–14,999-lb group was desirable, and that light vehicles grossing 15,000 lb and over should be equipped with a booster. The present study tends to indicate that, except for reducing driver fatigue, a booster is not beneficial on the brakes of present light 2-axle trucks if they are not loaded beyond the manufacturer's gross weight rating.

Table 12 shows the average brake performance for each of three gross weight groups of medium 2-axle truck-tractors with 1-axle semi-trailers equipped with two different types of brakes. Average brake-system application and braking distances from 20 mph are shown for vehicle combinations with vacuum-booster hydraulic brakes on the truck-tractor and vacuum-mechanical brakes on the semitrailer, and for combinations with air-mechanical brakes on both units. The table indicates that the vehicles with air-mechanical brakes were capable of stopping in 2- to 5-ft shorter distance than those equipped with vacuum power systems.

The 1949 tests showed a similar relation between the performance of the two brake types for vehicles selected from the general traffic. However, controlled tests conducted during the earlier study showed that the 2-axle truck-tractors with 1-axle semitrailers equipped with vacuum-booster hydraulic brakes on the tractor and vacuum-mechanical brakes on the trailer could be made to perform on a par with similar vehicles equipped with air-mechanical brakes. This was true for vehicles weighing up to 40,000 lb, provided proper components of the braking systems were selected and used, and provided the braking systems were properly maintained.

Table 12—Relation between Brake Type and Brake-System Application and Braking Distance for Medium 2-Axle Truck-Tractors with 1-Axle Semitrailers

Gross Combination Weight Group, lb	Number of Vehicles		Average Brake-System Application and Braking Distance from 20 mph, ft	
	VBH-VM Brakes[a]	AM-AM Brakes[a]	VBH-VM Brakes[a]	AM-AM Brakes[a]
20,000—29,999	7	8	40	35
30,000—39,999	8	6	46	42
40,000—49,999	4	11	51	49

[a] VBH = Vacuum-booster hydraulic; VM = Vacuum-mechanical; AM = Air-mechanical

Table 13—Average Minimum Brake-System Application and Braking Distance and Average Gross Weight by States for Various Vehicle Types

Vehicle Type[a]	Brake-System Application and Braking Distance, ft			Gross Weight, lb		
	Md.	Mich.	Calif.	Md.	Mich.	Calif.
Passenger Cars	23	19	21	4050	4000	3850
Single-Unit Trucks						
2-Axle (Very Light)	26	23	26	5250	5600	4950
2-Axle (Other than Very Light)	35	30	36	15,700	12,800	14,500
3-Axle	43	37	48	26,600	30,600	27,500
Truck-Tractors with Semitrailers						
2-S1	42	41	45	33,900	31,300	28,000
2-S2	44	43	46	40,800	40,200	38,900
3-S2	42	40	52	50,900	49,300	57,400
Trucks with Full Trailers						
2-2		48	55		48,100	43,200
3-2		48	57		47,800	70,200
Truck-Tractors with Semitrailers and Full Trailers						
2-S1-2		54	56		65,100	58,300
2-S2-2		57	46		70,600	50,000

[a] See section entitled, "Vehicle Type Code," for explanation of the code used.

Brake Performance Compared by Regions

The previous analyses have been concerned with average brake-system application and braking distances for the total sample of vehicles tested in three states. Each state actually represents a region since the normal sample of out-of-state vehicles was included. Table 13 shows by states the average brake performance and the average gross weight of each of the most common vehicle types tested in 1955. It is interesting to note that 10 of the 11 vehicle types considered showed better average stopping ability in Michigan than in either Maryland or California. For some vehicle types the better showing may be partly attributed to differences in average gross weights. However, the average weight was heavier in Michigan for six of the ten types which showed the best performance in that state. The average weight of the one type of vehicle which did not show its best performance in Michigan, the 2–S2–2 combination unit, was 20,600 lb heavier than in California.

In general, the average level of performance was better in Maryland than California. Of the seven vehicle types tested in Maryland, five had a better average performance in Maryland than in California. The average weight was heavier in Maryland for three of those five types. The passenger cars showed about 10% better performance in California, and the very light 2-axle trucks required the same distance to stop in both states. However, the average weight of each of these two vehicle types was slightly heavier in Maryland.

Since the improvement in stopping ability of the

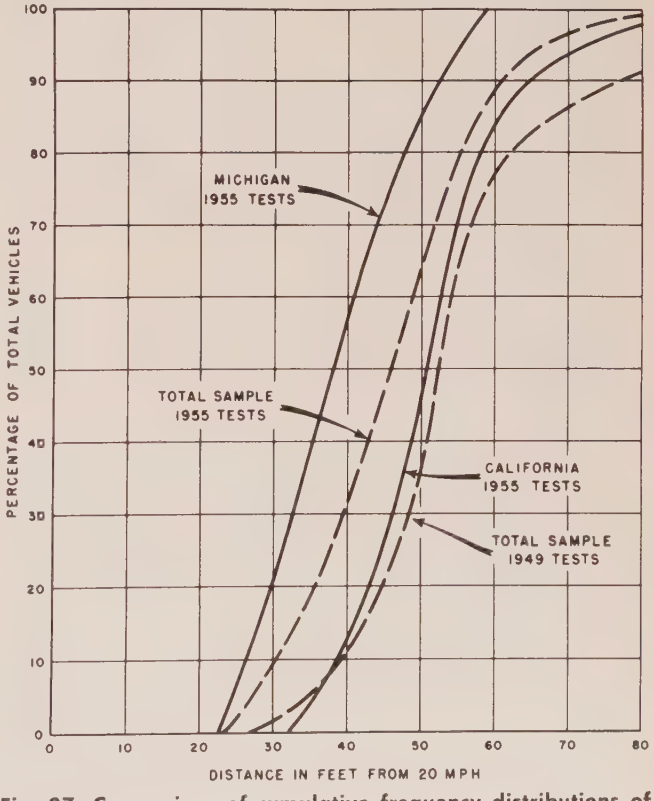


Fig. 27—Comparison of cumulative frequency distributions of minimum brake-system application and braking distances for 3-axle truck-tractors with 2-axle semitrailers tested in Michigan and California in 1955

Table 14—Comparison of Brake-System Application and Braking Distances by Weight Groups for 3-Axle Truck-Tractors with 2-Axle Semitrailers Tested in California and Michigan

Gross Combination Weight Group, lb	Number of Vehicles		Average Brake-System Application and Braking Distance from 20 mph, ft	
	Calif.	Mich.	Calif.	Mich.
20,000 29,999	1	9	43	31
30,000 39,999	4	.	45	
40,000 49,999	6	3	49	37
50,000 59,999	6	1	54	55
60,000 69,999	9	15	52	44
70,000 75,999	9	1	59	51
Summary, All Gross Combination Weight Groups	35	29	52	40

3-axle truck-tractors with 2-axle semitrailers has previously been emphasized, it is of particular interest to investigate the performance of this vehicle type in the individual states. Because only two vehicles of this type were tested in Maryland, comparisons will be confined to vehicles tested in Michigan and in California.

Fig. 27 shows four cumulative frequency distribution curves for brake-system application and braking distances of 3-axle truck-tractors with 2-axle semitrailers. Two curves represent the vehicles tested in Michigan and California in 1955. The other two represent the total samples of this vehicle type tested in 1955 and 1949. The 1949 curve can be considered as essentially representing the performance of vehicles tested in California, as only 2 of the 57 vehicles it represents were tested elsewhere. The wide variation in performance of similar vehicles tested in the two states is of particular interest. The figure indicates that only small improvements in the performance of these vehicles were made in California between 1949 and 1955. The greater part of the improvement indicated for the total sample was due to the performance of vehicles tested in Michigan.

It will be pointed out later that differences in terrain may contribute to differences in performance of vehicles tested in different states. However, for the 3-axle truck-tractors with 2-axle semitrailers some of the difference should be attributed to another factor. Reference to Table 2 will show that in 1949 only two vehicle combinations of this type were tested in Michigan. However, since 1949 this vehicle type has been gaining in popularity in the Midwest. The descriptive data recorded in 1955 shows that on the average the power units of these vehicles tested in California were five years old, and in Michigan they were 2½ years old. Thus, it is probable that the 3-axle truck-tractors with 2-axle semitrailers tested in Michigan were generally equipped with more modern braking equipment.

Since Table 13 shows that the average weight of these vehicle combinations averaged 8100 lb less in Michigan, it might be construed that the difference in weight was a major contributing factor to the difference in performance. Table 14 compares by weight groups the average brake-system application and braking distances for vehicles of approximately the same weight tested in each of the two states. It is significant that in general the vehicles of this type tested in Michigan were able to stop in 8- to 10-ft shorter distance than those of approximately equal weight tested in California. The results shown in Table 14 substantiate the relative performances indicated in Fig. 27.

The relative performance by states of the vehicles considered in Table 13 no doubt reflects the type of terrain they normally traverse. Most of the vehicles tested in Michigan usually travel over level to rolling terrain; many of those tested in Maryland frequently travel through hilly and mountainous country; and many of the vehicles tested in California often travel over mountains where long sustained grades are encountered. Where long or steep grades are encountered, wear of the brakes is generally more severe than in flat country, and more frequent brake adjustments are needed. The differences in brake performance by regions emphasize the need for more frequent brake maintenance for vehicles which travel over hilly or mountainous terrain.

Incidental Data of Interest

A considerable amount of data which may be termed incidental to the main purposes of the study were obtained in connection with the brake tests. Much of these data are interesting and useful for reference purposes and in some cases have a bearing on the results just reported.

Pedal Reserve—Pedal reserve is the distance in inches between the floor board and the back of the brake pedal at the end of an emergency stop. Except for vehicles equipped with certain power brake systems for which it is normal for the pedal to bottom on the floor board, zero pedal reserve is a good indication of need for brake maintenance. A comparison of the percentages of vehicles having zero pedal reserve found in 1942, 1949, and 1955 studies is shown in Table 15.

The decrease in percentages of vehicles with zero pedal reserve in 1949 and again in 1955 probably reflects improvements in both maintenance and design of the braking systems. The magnitude of the reductions is in keeping with improvements in stopping ability which were previously discussed.

Swerving—The term "swerving" describes an uncontrollable lateral movement of the vehicle when the brakes are applied which is involuntary on the part of the driver. In severe cases it may force the vehicle into the opposing traffic lane or completely off the road. If this condition exists in stops made from 20 mph, experience has shown that it is usually accentuated in stops from higher speeds. This is one very important reason why tests on the highway in traffic with the regular drivers should not be conducted at a speed greater than 20 mph. Some vehicles of each type swerved to some extent. The passenger

cars and the very light 2-axle trucks were the worst offenders. A comparison of percentages of vehicles that swerved in the 1942, 1949, and 1955 studies is shown in Table 16.

The condition of swerving appears to have improved somewhat since 1949, but there is need for much more improvement, particularly in the passenger cars and single-unit trucks.

Wheel Slides—The sliding of wheels on a vehicle making a stop indicates that more than adequate braking force is being applied, at least for some part of the stop, with respect to the wheel loads and the frictional characteristics of the particular pavement and tire. The ideal condition, which is difficult to attain because of varying conditions of pavement surface, tires, and brake lining, is to apply just enough braking force to bring the wheel to the point of impending skidding. The fact that a wheel does not slide does not necessarily mean that more braking force is needed. However, the number of vehicles of a sample that slid wheels, when considered in connection with the general levels of brake performance, does give a clue as to whether more braking force could be used.

The percentage of vehicles of various types that slid one or more wheels on any axle is shown in Table 17. A relatively small percentage of the commercial vehicles, especially the combination vehicles, slid wheels as compared to the passenger cars. A large percentage of the trucks and combinations that slid was empty: 40% of the very light 2-axle trucks, 37% of the other single-unit vehicles, 31% of the truck-tractors with semitrailers, and 31% of the other vehicle combinations. Many of the remaining vehicles with sliding wheels were only partly loaded.

There were 35% of the passenger cars, 8% of the very light 2-axle trucks, and 2% of the other 2-axle trucks that slid all wheels. None of the 3-axle trucks or vehicle combinations slid all wheels.

From the foregoing discussion of the vehicles with sliding wheels, and from the levels of performance described earlier in this report, a suspicion arises that loaded combination vehicles now operating in the general traffic could use additional braking force to advantage. The deficiency in braking effort, if such is the case, could be attributed either to loads greater than those for which the brakes were designed or to inadequate brake-system maintenance.

Performance of Miscellaneous Vehicle Combinations—In the tabulation of the various types of vehicles which were tested (Table 1), it will be noted that the 1955 study included three passenger-car driveaway-towaway combinations and five housetrailer factory-towaway combinations. Although the performance of such a small sample cannot be considered representative, it does provide some indication of the stopping ability of these combinations. The three passenger-car-towaway combinations required 39, 46, and 67 ft to stop. In the case of the combination which stopped in 39 ft the brakes on the towed car, as well as those on the towing car were applied in stopping. Both of these cars were new. The other two towaway combinations consisted of second-hand cars, and the brakes were operative on only the towing vehicle.

The power units of the five housetrailer factory-towaway combinations were very light trucks with manufacturers' gross weight ratings for use as single-unit vehicles varying from 5800 to 9500 lb. These combinations were equipped with hydraulic brakes on the power units and electric brakes on the trailers. The average gross weight of these five combinations was 12,800 lb, and their average brake-system application and braking distance was 36 ft. Additional housetrailer combinations were stopped for testing but were excused because the drivers were afraid a sudden stop would damage refrigerators or other equipment inside the trailers.

Conclusions

The following conclusions drawn from this study are based on the type and number of vehicles tested, and on the test procedures employed:

1. Except for two vehicle types, the improvement in the general levels of brake performance of motor vehicles selected from the general traffic was considerably smaller between 1949 and 1955 than it was between 1942 and 1949. The 3-axle truck-tractor with 2-axle semitrailer combinations and the truck-tractor-semitrailer-full-trailer combinations each showed an appreciable improvement of about the same magnitude in both periods. In contrast, for all other vehicle types, the improvement was appreciable in the earlier period while it was negligible in the later period.
2. In view of the small magnitude of improvement

Table 15—Percentage Comparison of Pedal Reserve

	1942	1949	1955
Passenger Cars	41	7	3.3
2- and 3-Axle Trucks	56	10	4.3
2-Axle Truck-Tractors with 1-Axle Semitrailer	75	7	1.9

Table 16—Percentage Comparison of Cars Swerving

	1942	1949	1955
Passenger Cars	20	34	28
Very Light 2-Axle Trucks			29
All 2- and 3-Axle Trucks	16	18	14
2-Axle Truck-Tractors with 1-Axle Semitrailers	12	12	5
Combination Vehicles with 4 or More Axles	6	8	4

Table 17—Vehicles that Slid One or More Wheels

	Per cent of Vehicles Sliding Wheels on							
Vehicle Type	Axle 1	Axle 2	Axle 3	Axle 4	Axle 5	Axle 6	Axle 7	One or More Axles
Passenger Cars	54	74						85
Very Light 2-Axle Trucks	18	74						76
2-Axle Trucks other than Very Light	6	48						51
3-Axle Trucks	0	45	48					51
2-Axle Truck-Tractors with 1-Axle Semitrailers	0	26	31					40
2-Axle Truck-Tractors with 2-Axle Semitrailers	1	17	37	45				53
3-Axle Truck-Tractors with 2-Axle Semitrailers	0	17	18	27	38			50
Trucks with Full Trailers (4, 5, and 6 Axles)	0	9	7	10	19	3		29
Truck-Tractors with Semitrailers and Full Trailers (5, 6, and 7 Axles)	0	11	33	22	48	4	4	61

in brake-system application and braking distance made by most vehicle types since 1949, it appears that the wide range in stopping ability of the various vehicle types will not be appreciably reduced in the near future. It follows that this wide range must be recognized in highway design, vehicle regulation, and driver-training activities.

3. From 38 to 64% of the largest combination vehicle types and about 80% of the 3- and 4-axle truck-tractor-semitrailer combinations and 3-axle trucks were capable of complying with the Uniform Vehicle Code specification of a 50-ft stop from 20 mph; 84% of each of the two classifications of 2-axle trucks and 92% of the passenger cars were able to comply with their respective distance specifications of 40, 30, and 25 ft.

4. Almost all passenger cars and 2-axle trucks, 76 to 85% of the 3-axle trucks and truck-tractor-semitrailer combinations, 69% of the truck-tractors with semitrailers and full trailers, and 51% of the trucks with full trailers were capable of meeting their respective Uniform Vehicle Code specifications for deceleration when tested with a commercially available pendulum-type decelerometer.

5. The observed average levels of brake performance in stops from 20 mph for commercial vehicles, operating in the general traffic with normal loads, was about 25 ft for 2-axle trucks with a manufacturer's gross vehicle weight rating of 10,000 lb or less, and ranged generally from 35 to 45 ft for other 2-axle trucks, from 40 to 50 ft for 3-axle trucks, from 45 to 55 ft for 3-, 4-, and 5-axle truck-tractor-semitrailer combinations, and from 55 to 65 ft for truck-full-trailer and truck-tractor-semitrailer-full-trailer combinations.

6. Many of the commercial vehicles, except very light 2-axle trucks, were found to be inadequately braked in proportion to the loads carried on individual axles. This condition may result from inadequate maintenance, axle loads heavier than those for which the brake components were designed, and/or poor selection of brake components. However, individual tests proved that vehicles with axle loads as high as 22,000 lb can be adequately braked.

7. A substantial percentage of passenger cars and single-unit trucks selected from the general traffic showed a tendency to swerve—a condition that may be very dangerous in making emergency stops from normal speeds.

DISCUSSION

Weight and Braking Distributions Between Axles Affect Stopping Ability

—P. J. Reese
Wagner Electric Corp.

IT is appropriate for the Bureau of Public Roads to be conducting this study, because the road is such an important factor in the stopping ability of motor vehicles. That such is the case is evidenced by the fact that it is so carefully specified for stopping-distance tests, like in this study, as dry, smooth pavement free of foreign material, approximately level, and located on a tangent. This is the optimum road condition for maximum braking performance of vehicles with pneumatic tires. If motor vehicles could operate on this kind of a road at all times the braking problem would be very much simplified, and better braking performance could be provided by the brake and vehicle manufacturer.

Unfortunately the weather, by reason of rain, snow, wet ice, or sleet can change the road condition from optimum to minimum on very short notice. In so doing emphasis is placed on proper braking distribution between the steering and driving axles and braking control to prevent wheel slides which result in loss of steering control if the front wheels slide, to prevent a sideways skid of the rear end of a single vehicle if the rear wheels slide, and to prevent jackknifing of combinations. In order to eliminate any chance of loss of steering control in bad weather, some operators of combinations disable the brakes on the steering axle, thereby reducing the maximum stopping ability on good roads.

The proper distribution of braking between axles is a subject about which there is considerable difference of opinion among the brake engineers of the truck manufacturers. Perfect distribution is possible for only one combination of braking versus loading versus rate of deceleration due to weight transfer from one axle to another during deceleration. Transfer varies with the rate of deceleration and with the location of the center of gravity of the load and vehicle. Due to the wide variations in weight on the rear axles of trucks from empty to loaded, it is possible for a truck to make a shorter skidless stop when carrying rated load than when empty.

To illustrate this, let us consider a 25,000 gross vehicle weight 2-axle truck and assume the following:

	Loaded	Empty
Weight on front axle, W_F	7,000	5,500
Weight on rear axle, W_R	18,000	6,000
Total weight, W	25,000	11,500
Center of gravity height/wheelbase, h/L	0.35	0.30

Braking distribution to be taken as ideal for loaded truck at 15-fps per sec deceleration.

Braking distribution at 15 fps per sec:

Weight transfer, loaded, $0.35 \times 25{,}000 \times 15/32 = 4100$ lb
Weight on front axle, $4100 + 7000 = 11{,}100$ lb
Weight on rear axle, $18{,}000 - 4100 = 13{,}900$ lb
Ideal front braking, $11{,}100/25{,}000 = 0.444$ (44.4%)
Ideal rear braking $13{,}900/25{,}000 = 0.556$ (55.6%)
Necessary friction coefficient, $15/32 = 0.469$

On pavement of 0.7 coefficient (best skidless stop)

1. Loaded or unloaded:

Skid impends at rear when rear retarding force, $F_R = 0.7W_R$

At which time, front retarding force, $F_F = 0.444F_R/0.556 = 0.559W_R$

Total retarding force is then $F = (0.700 + 0.559)\ W_R = 1.259W_R$

But retarding force equals inertia force, or $F = Wd/g$ where d = deceleration, g = gravitational acceleration.

2. Loaded truck:

Weight transfer, $\frac{h}{L}\frac{Wa}{g} = 0.351. \times 1.259\ W_R = 0.441\ W_R$

Rear axle load, $W_R = 18{,}000 - 0.441\ W_R$ or $(1 + 0.441)\ W_R = 18{,}000$. Hence $W_R = 18{,}000/1.441 = 12{,}500$ lb

Front axle load, $W_F = 25{,}000 - 12{,}500 = 12{,}500$ lb

Deceleration, $d = g \times F/W = 32 \times 1.259 \times 12{,}500/25{,}000 = 20.1$ fps per sec

Efficiency of utilization of existing friction, $d/fg = 20.1/(0.7 \times 32) = 0.899$

Retarding force at front wheels, $F_F = 0.559\ W_R = 7000$ lb

Utilized friction at front wheels, $F_F/W_F = 7000/12{,}500 = 0.56$

Efficiency of utilization of existing friction, front wheels only $0.56/0.70 = 0.80$

3. Empty truck:

Weight transfer, $0.30 \times 1.259\ W_R = 0.378\ W_R$

Rear axle load, $W_R = 6000 - 0.378\ W_R$ or $(1 + 0.378)\ W_R = 6000$. Hence $W_R = 6000/1.378 = 4355$ lb

Front axle load $W_F = 11{,}500 - 4355 = 7145$ lb

Deceleration, $d = 32 \times 1.259 \times 4355/11{,}500 = 15.25$ fps per sec

Efficiency of utilization of existing friction, $15.25/22.4 = 0.68$

Retarding force at front wheels, $F_F = 0.559 \times 4355 = 2435$ lb

Utilized friction at front wheels, $F_F/W_F = 2435/7145 = 0.341$

Efficiency of utilization of existing friction, front wheels only, $0.341/0.70 = 0.487$

This same problem of weight distribution versus braking distribution and control exists on combinations and affects their stopping ability but, as Mr. Petring concludes, individual tests proved that vehicles with axle loads as high as 22,000 lb can be adequately braked.

Predicts Greater Improvement For Future Brake Performance

—J. V. Bassett
Raybestos-Manhattan, Inc.

I WILL take issue with one conclusion, namely that there has been little improvement since 1949. I feel that there has been, and I am certain that the next survey will show a marked improvement, if some way can be found to stop, not reasonable overloading, but serious overloading. As an example, two years ago I was asked by an engineering department to investigate a continuing complaint at their Detroit branch. The operator admitted that his payloads exceeded the gross rating of the truck. In discussing this statement with the branch service manager, he stated that unless the owner carried these overloads he could not meet the payments on his truck.

One of the surprising things is that the 3-axle tractors and 2-axle trailers showed such good performance. More and more of the drivers of this class of vehicle disconnect their front wheel brakes. The practice started at the West Coat and is working East. My last experience with it was in Iowa and Minnesota.

The late Ed Dryer not only recognized the problem but did something about it. Ed stated that the trucks as built required full reservoir pressure to make a 30-ft stop with rated load, with no safety factor for overloads. Shortly before his death, he changed from the usual 0.35 friction material to 0.45 and made the 30-ft stop with the 60-psi design pressure. This change also went into tractor production with the avowed policy that on complaints of driving axle lining wear, the operator would be told, "We have improved the tractor brake performance; you bring your trailer up to our performance."

One trailer axle builder is now in production with blocks of 0.5 friction. Another company has been testing some brake blocks since June, 1955, with 0.55 friction. I might add here that the old saying that as coefficient of friction goes up, life of the lining goes down, is no longer true.

I believe that the next survey will show even greater improvement over 1955 than that of 1949 over 1942. While the valve manufacturers have made progress in speeding up application time, the fact that heavy vehicles can reach the required deceleration but not the required stop shows that even more improvement must be made. I am wondering if the ICC protection valves are not going to slow up brake action to some degree, as they are, of necessity, restrictions in the lines.

Discusses Requirements of Stopping Distance Measurements

—M. E. Nuttila
Cities Service Oil Co.

Importance of Subject

IT is apparent that the subject of brake adjustment is progressing from the decelerometer phase to the decelerometer and stopping distance phase, due to revisions adopted in the Uniform Motor Vehicle Code. It is important that the limits for stopping distances be made practical, and this can only be done through the type of testing herein recorded.

When measurement of distance is included, there is added a distinct hazard whenever a vehicle must be stopped for test purposes on the highway. When officially organized, proper precautions can be taken. However, a private individual, to accomplish similar tests, would have numerous problems unless an off-highway area is available. Because of this hazard when measuring actual stopping distance, there will be a preference for the use of equipment in inspection stations which would not require an actual measurement, but which would give an approximately equivalent reading.

Importance of Proper Data Based on Recent Legislation

Pennsylvania—It is my understanding that the present law calls for a stopping distance of 30 ft from 20 mph for all vehicles. The results of the brake tests as reported in Mr. Petring's presentation indicate such a requirement as being impractical, and some modification is indicated. Information of this type is important in preventing adoption of similar regulations elsewhere.

New York—The inspection code, which becomes effective in 1957, outlines the brake inspection requirements as follows: road test *or* brake testing machine must show less than the following stopping ability, from a speed of 20 mph.

1. Passenger vehicles, 25 ft
2. Single-unit vehicles with manufacturer's gross weight rating less than 10,000 lb, 30 ft
3. Same as (2) except over 10,000 lb, 40 ft
4. All others, 50 ft

As pointed out in the report under the section entitled "Ability of Vehicles to Comply with Specifications for Deceleration," it is easier to pass inspection based on instrument readings or brake-testing machines than by actual measurement. The alternate procedure allowing the use of a brake testing machine should prove the most popular because (1) it is easier to qualify, (2) it is not necessary to attain a speed of 20 mph in order to register an apparent stopping distance equivalent on the testing machine, and (3) the hazard of stopping on the highway is eliminated in those cases where a separate strip is not available.

General Levels of Performance

It is also stated that, "The curves shown in Fig. 9 become closer together at the lower percentile points, and it is obvious that the greatest possibilities for improvement exist with the larger commercial vehicles." This is probably true, and detailed data in the various charts broken down by weight groups seem to support this premise. However, it cannot be considered as a crystal-clear conclusion because of a possible difference in the relative number of lightly loaded axles. That there is a large variation in axle loadings is indicated by the subsequent charts. A more dependable conclusion could be drawn from a graph where all axles were known to be loaded to rated capacity or nearly so, because stopping distance is often better with partly loaded versus fully loaded axles. The partly loaded vehicles could very well have an abnormal effect on the lower portion of the curves. To repeat, an additional series of curves for vehicles with fully loaded axles only would result in the elimination of one variable and could more convincingly support the conclusion that the greatest possibilities for improvement exist with the larger commercial vehicles.

Future

Because of the effect of incorporating actual stopping distances in the Uniform Vehicle Code and the possible adoption of the code by an increasing number of states, the problem of proper brake performance and regulations is more vital than ever before. Also, as traffic moves faster, a greater percentage of stops from higher speeds is necessary, requiring brakes to work harder. Both factors place an increased emphasis on this subject of brake performance. There are, and will continue to be, many complications, and Mr. Petring's paper contributes in an important way to the proper solution of this problem.

DISCUSSION continues on page 426

Discusses Need for Better Brake Adjustment

—J. William Lord
Atlantic Refining Co.

ANYONE who has seen Carl Saal or Bill Petring go through the work of testing the brakes of just one truck has some slight conception of the huge job done by Bill Petring in traveling the whole country to make about 1300 such tests, gather the data, correlate them, and present the findings in a precise paper. It is a huge job, and we should be most appreciative of what both these men are contributing toward the welfare of the trucking industry.

The primary purpose of the study and report was aimed, "To show the levels of brake performance that now prevail for the various types of vehicles currently found on the highways."

In general the report says we are making progress in having better brakes but it also says, and quite plainly, that in the overall picture we could and should do a far better job than is being done with present equipment. Much the same thing was said in previous reports, when truck after truck gave a poor brake performance for lack of a simple brake adjustment.

Table 4 and Table 10 deserve study. Those 15-percentile figures should be thoroughly considered. Thus the first five of the 85-percentile figures exceed the corresponding 15-percentile best by:

1. 5 ft or 29%.
2. 17 ft or 78%.
3. 24 ft or 80%.
4. 21 ft or 67%.
5. 17 ft or 50%.

One may ask why be so concerned about 5-ft difference or a 39-ft stop from 20 mph. The answer is to be found in what happens not at 20 mph but at speeds closer to average road speeds, for instance, a conservative speed of 40 mph. Not too long ago a 2-axle heavy vehicle was subjected to brake test and, as found, it stopped in 43 ft. Then work was done on it and the stop was brought down to 32 ft, an improvement of 11 ft.

However, this vehicle was also tested before and after repairs at 40 mph. Now see what happens.

It took 190 ft to stop before any work was done and 127 ft to stop after repairs had been made—a difference of 63 ft, and here is what that 63-ft difference means. When making a stop from 40 mph before repairs and after traveling with brakes set a distance of 127 ft, the *speed had dropped to 24 mph*; but after repairs, the vehicle was standing still after braking 127 ft. This improvement could easily represent the difference in an emergency stop between no collision and a violent collision at 24 mph which is a highly dangerous speed as studies have proved.

Mr. Petring is saying over and over that we should be doing far better than indicated by the 85-percentile figures. The results have been presented so that all can see the wide spread between the 85-percentile values and the best 15 per cent: "The 15-percentile values show definitely that considerable improvement is possible for each type of vehicle." Similarly, "A few of the vehicles tested in 1955 add further proof that *axle loads now commonly carried can be adequately braked.*" Then he gives stopping distances of 36 ft, 36 ft, 38 ft, and 38 ft and he goes on to comment, "Results of the controlled tests reported in 1954 indicate these vehicles could have done better if they had had a brake adjustment immediately prior to the tests."

Mr. Petring's paper is very much a challenge to the trucking industry to get brakes into considerably better condition and to keep them in better adjustment. He points out what the leading 15% of the test sampling are doing and how far back the average and 85% are trailing. He goes further, and in Fig. 27 shows the relative conditions found in Michigan and California with California definitely trailing. Here is a bit of healthy competition which I hope takes fire. The question may properly be asked, "What are we going to do about it?"

As I see it, it calls for a far wider and better appreciation and understanding of the problem by operating and maintenance supervisors in general, and an appreciation that every "brake failure accident" reported to ICC is a black eye for the industry, not just for the outfit reporting the accident. A continued study, training, and instruction will help the situation. I'm sure that as the facts are more widely and better understood that we will see fewer brake failures and the need to report them to ICC. The problem is a vital matter we cannot ignore, and the sooner we tackle it the better.

ORAL DISCUSSION

Reported by C. P. Hoffmann, Jr.
American Trucking Associations, Inc.

William E. Williams, Public Utilities Commission of California: I agree with Mr. Bassett's statement that there has been a vast improvement in stopping distances. However, I feel that we have lost ground in transmission and buildup time due to improper piping and valving.

(Mr. Williams has developed a timing device which is capable of measuring transmission and buildup to 0.001 sec. With this device he has had an opportunity to investigate some of the vehicles which are on the road today.)

I have done some work with Navajo Freight Lines regarding brake lining. They installed thermocouples on the linings and took readings every 30 sec with the transmission in third gear and the engine operating at 1900 rpm. The brake lining caught fire at 1300 F. It was noted that when the brake lining caught fire (under the conditions described above) there was ⅜-in. push-rod travel remaining. The same test was repeated until the brake temperature read 2400 F, and under these conditions there was no push-rod travel left. These tests were run to determine when a runaway condition could be expected. The results indicated that a runaway could be expected on a 6 to 7% downgrade if the vehicle came over the top of the hill with the transmission in third over, engine speed of 1900 rpm, and road speed of 40 to 45 mph.

Do you think that the steering axle of a full trailer should be equipped with brakes?

Mr. Petring: Under present speed and load operating conditions, it is advisable to include brakes on a front steering axle of a full trailer. Diving or loss of steering control may result with a full trailer which has front wheel brakes or with one which does not have front wheel brakes. The need for additional braking power more or less dictates the need for brakes on the front or steering axle of a full trailer.

Lewis C. Kibbee, American Trucking Associations, Inc.: Truck operators and manufacturers must work together. Truck operators and manufacturers alike are learning that locking the wheels is not the only criterion which indicates a good braking system, if brake time is excessive. Increased loads, higher speeds, the need for better controlled stops as well as emergency stops, the use of front wheel brakes, and the relationship of steering ability to stopping ability must be continually investigated to insure ideal braking systems. In addition, manufacturers' sales representatives who oversell the vehicle should be aware that in all probability they are selling a vehicle with inadequate brakes. Brake-line "fade" is a problem on which the operator and the manufacturer must work together. With respect to full trailers breading away, I suggest that new developments in safety chains may be a help in this problem.

Carl Saal, United States Bureau of Public Roads: Under the nation's highway construction program, size and weight considerations are of the utmost importance. Indications for the future are that larger, heavier units may be operated. Naturally, brakes and stopping distances are an essential consideration. The operators and manufacturers have done a lot toward the solution of this problem; however, there is a need for continued team cooperation.

W. W. Vandercook, Boutell Driveaway Co.: Where can I get information pertaining to the selection of the proper components of an adequate braking system?

Mr. Petring: The manufacturers' recommendation relative to components such as brake size and brake chamber size are available from various manufacturers who relate their specific use to the particular axle loading which the vehicle is expected to carry.

THE APPLICATIONS OF RADIOACTIVITY FOR THE CONTROL AND TESTING OF AUTOMOTIVE MATERIALS

H. A. Tuttle and G. E. Noakes, Ford Motor Co.

This paper was presented at the SAE Summer Meeting, Atlantic City, June 6, 1956.

APPLICATIONS of nuclear energy have for some time provoked extensive conjecture and research in modern industry. Although industry is unable to make more than a token use of this new source of power, certain kindred developments have found considerable application. Most important of these have been in the field of radioactivity. Here, characteristic radiations from a wide variety of radioisotopes have become potent tools of the engineer in the research, development, and control of most materials and industrial processes. This paper has been prepared to illustrate typical applications of radioactivity for the control and testing of automotive materials in our company.[1] It can be said that all cars leaving our plants owe some of their high quality to such techniques.

Radioactivity applications are grouped under the following categories: radiography, nondestructive testing, gaging and control, tracer techniques, static neutralizers, and processing. Potential applications under consideration will be cited briefly. Within our company there are several groups employing radioactivity, each making specific applications to their field of endeavor. These include the scientific laboratory, engineering staff, where fundamental long-range research is conducted; engineering research, engineering staff, where automotive engineering measurements and tests are conducted; [2,3] and manufacturing research, manufacturing staff, where new materials and manufacturing methods are developed and applied. In addition, there are operating divisions of the company which utilize radioisotopes for specific applications.

Some of our applications are unique, but the majority are paralleled in other industries. It is estimated that over 1200 companies employ radioisotopes either in research or industrial processes. The field is still new enough to permit considerable additional exploitation.

Individual applications may originate anywhere in the company. Many problems develop which can be solved only by using radioactive materials. Each is studied to develop the feasibility of the solution by radioisotope techniques and then referred to the appropriate research or development group.

Historical Notes

Historically, the first application at Ford started with the use of natural radioisotopes, such as radium, for radiographic purposes about 1934. Because of the high cost and scarcity of material, extensive applications did not develop. However, at the close of

1 *Report of Panel on Impact of Peaceful Uses of Atomic Energy*, Vol. 2, January, 1956, pp. 270–280: "Peaceful Uses of Atomic Energy."

2 *Product Engineering*, Vol. 26, May, 1955, pp. 129–134: "Radioisotopes as Design Tools," by A. M. Smith.

3 "Radioactivity and Automotive Measurements," by A. M. Smith. Paper presented at SAE Golden Anniversary Summer Meeting, Atlantic City, June 14, 1955.

APPLICATIONS of nuclear energy in automotive manufacture have been made principally in the field of radioactivity. These are grouped under the following categories: radiography, nondestructive testing, gaging and control, tracer techniques, and static neutralizers.

Radioactivity techniques are being used in foundry operations to check stock and metal levels in cupolas and distribution of element additives. In steel operations, these techniques are being used to check assimilation of ore-concentrate fines and thickness of rolled sheet steel. Other applications include measurement of pipe and wall thickness in pressure lines and engines, and inspection of castings and welds for internal faults.

Radioactive techniques for improving processes, quality, and materials have potentially universal application. Greater industrial access to reactors will permit broader study and speed the development of new applications of radio-activity in industry.

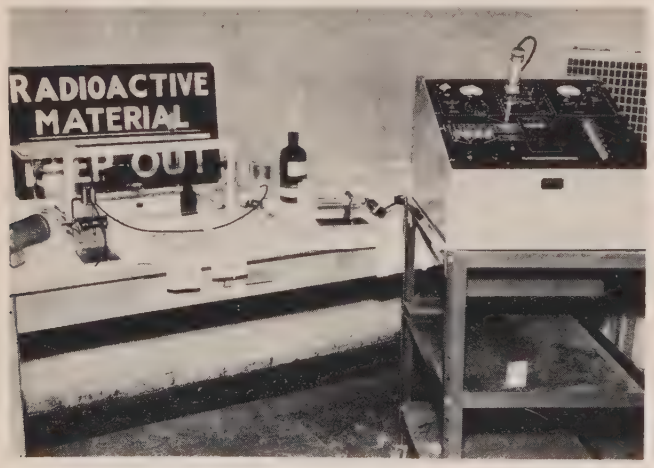


Fig. 1—Typical setup for low-level beta experiment

Fig. 2—Cobalt 60 vault, positioned for vertical radiograph

World War II when additional radioisotopes became available through the Atomic Energy Commission, interest was renewed. After some preliminary study in 1947, 40 millicuries of cobalt 60 were received in June, 1948, from Oak Ridge by the group which is now the manufacturing research department. This shipment is understood to have been the first made from Oak Ridge by commercial carrier transit to an industrial organization. Intensive studies of potential applications to all company operations followed. Many of the applications which are now in common use were evaluated and pioneered during this period. In 1951 a group at engineering research was organized to apply radioisotopes to automotive research. At about the same time the scientific laboratory was founded, with the inclusion of radioisotope research activities. The applications of radioactivity in one form or another have steadily increased, until today the various techniques have become well-recognized tools of research, and of engineering and manufacturing development.

Laboratory Facilities

Laboratory facilities for the use of radioactive materials vary considerably, depending largely on the

requirements of the types of applications under study. In some cases it has been possible to add minor necessary facilities to an existing laboratory, such as an X-ray radiographic laboratory or an X-ray diffraction physics laboratory. Fig. 1 is a photograph of a typical setup for a low-level beta experiment. For other applications, it has been necessary to establish completely new facilities. All laboratories are equipped with radiation detectors and counting equipment and associated instrumentation and safety equipment. The personnel working with radioactive materials are highly trained nuclear physicists, engineering physicists, and X-ray or physics technicians. Wherever possible, commercially available equipment is purchased for both experimental and applicational work. In the instances where this equipment is either not available or unsatisfactory, suitable equipment is designed and constructed in our own laboratories.

Protection of Personnel

No report can be made on the subject of radioactivity without consideration of radiological health precautions. Every possible effort is made to insure absolute safety in handling and working with radioisotopes. Precautions and radiation safety levels established by the Atomic Energy Commission form the basis for such a safety program. Survey meters, film badges, and radiation pencils are used to monitor dosages. Periodic inspection by the state health agencies and the industrial hygiene department of the company, together with periodic employee white-blood counts at the hospital are all part of the program. Sources of radiation are adequately shielded in special containers and in most areas are stored in locked underground vaults.

Initial applications are carefully evaluated. Authorization to purchase the radioactive material and to proceed with the development of the application is obtained from the Atomic Energy Commission. Radiation levels are kept to a minimum in all operations, and informal educational programs are conducted for untrained personnel associated with the operations. In field applications, all radioactive materials are kept in locked containers to prevent unauthorized individuals from handling the materials or otherwise exposing themselves to radiation.

Applications

Radiography—The most common use of radioactive materials, to date, has been radiographic inspection to complement conventional X-ray radiography.[4] Gamma-ray emitters such as cobalt 60, selenium 75, cesium 137, and iridium 192 are most generally used in this application. They provide energies comparable to X-ray generators and are low in cost, versatile in application, and highly portable. The gamma-ray source, generally in the form of a small metal pellet or capsule, is suspended near or inside the suspect part. A photographic film is placed on the opposite side of the part. Some of the gamma rays are absorbed by the part, but enough get through to darken or expose the film. Inclusions or cracks in the material cause darker areas on the film. A trained radiographer can interpret the radiograph and determine if the part is serviceable.

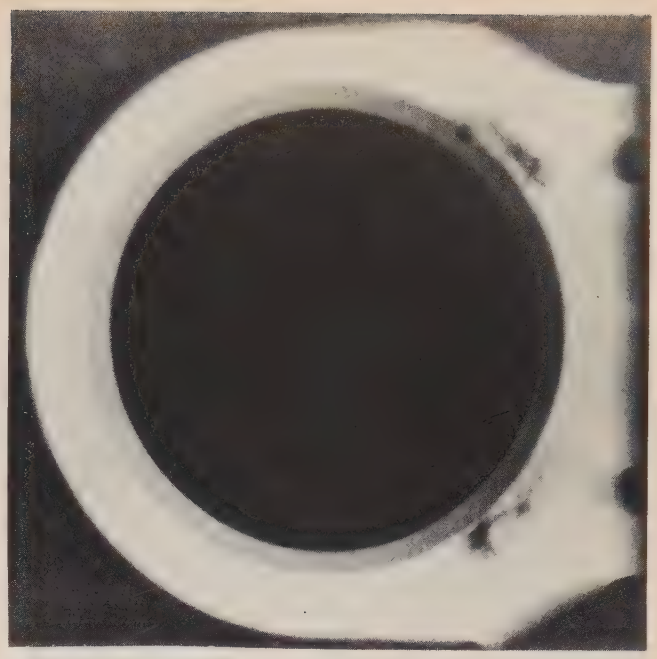


Fig. 3—Cobalt 60 radiograph of steel casting showing defect

Radioisotopes and associated equipment costing approximately $3,000 will perform many of the functions of a $50,000 X-ray generator. However, gamma radiography is generally used to supplement X-ray equipment. The modern X-ray radiographic laboratory makes provision for auxiliary gamma radiography. Fig. 2 shows an installation of this type where the radioactive source is housed below floor level. Remote positioning controls permit opening and raising the source. Many plants which cannot justify expense of an X-ray installation can, for specific requirements, obtain radiographs through the use of radioisotopes at relatively low cost. Fig. 3 is a photograph of a radiograph taken of a steel casting, using a one-curie cobalt 60 source. The shrinkage defects which would cause failure in service are clearly apparent. An assortment of various sizes and types of sources can be obtained for ferrous and nonferrous radiography. The following are typical examples of this type of application.

Example 1—Inspection of Welds in Rouge Office Building: A new three-story office building in the Rouge plant was built using a prepoured concrete floor type construction. The floors were raised and held in position by shear plates welded to the roof-support beams. Construction was done in the middle

4 "Experimental Work Employing Radioisotopes Cobalt and Selenium," by D. M. McCutcheon. Paper presented at Society of Nondestructive Testing, October, 1948.

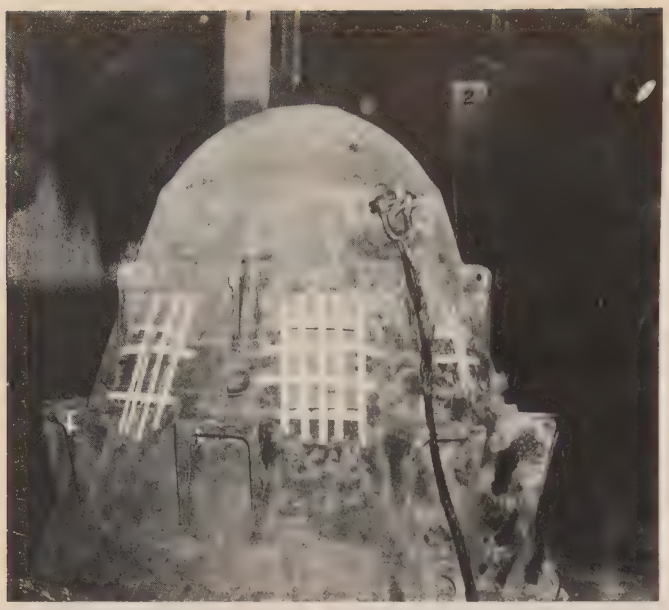


Fig. 4—Radiographing sections of 100-ton slag pot

of winter, and spot checks were required to insure satisfactory welds, since outward appearances could not be relied upon for absolute certification. An 0.7-curie cobalt 60 source was used to radiograph a series of welds on location. The results certified the work with a minimum of time and expense.

Example 2—Radiographic Inspection of Wing Section of a Company Plane: Stabilizer castings in the tail-wing section of a company-owned Convair plane require structural certification after every 200 hr of flying time. The stabilizers are encased in the tail section and cannot be visually inspected. Therefore, gamma radiographic inspection is made on location.

Example 3—Inspection of Slag Pots: When the steel division purchased nine 100-ton capacity slag pots, each weighing 30 tons, certain external defects appeared which were questioned by safety inspectors. Since the slag pots were too large to move to the X-ray laboratory, cobalt 60 radiography was used to inspect the pots in the field. Fig. 4 shows the setup which was used to perform this service. The source was located in the center of the overturned pot.

Other applications of this type include radiographic inspection of castings, of welds in high-pressure steam lines in the power house, and the detection of cracks in 100-ton crane hooks and trunions on metal ladles.

Nondestructive Testing—The term nondestructive testing, although it normally would include radiography, is used to classify those applications where a measurement is made on a process or part without interfering with the operation or destroying the part. The term implies batch-type testing or individual unit testing rather than continuous gaging. The applications of radioactivity to this type of testing in automotive production have been numerous. The following are two such examples.

Example 1—Penetron Pipe and Wall-Thickness Measurement: When it is necessary to measure the wall thickness of pipes, tanks, or tubes with only one side accessible, a commercial instrument of the Penetron [5] type is used. For flat or nearly flat surfaces, the thickness is related to the amount of back scatter of the gamma rays from a radium source. For pipes less than 6 in. in diameter, the absorbance along a chord is measured. Use of the gage has made it possible to determine the extent of corrosion in high-pressure steam lines, gas mains, and acid tanks without shutting down the operation. On one occasion a 4-in. high-pressure steam line in the power house ruptured. The question arose as to whether or not the entire boiler system should be shut down for repairs. By an inspection of the lines with the Penetron the hazardous portions were replaced, and the boiler placed back in operation without a costly and time-consuming major repair program. Fig. 5 shows a section of pipe located with the Penetron in which the bottom portion had eroded to a dangerous degree.

Example 2—Engine-Block Oil-Hole Wall Measurement: In the machining of the V-8 engine block using automation equipment it was found that one specific oil hole feeding a main bearing was drifting. In some blocks the hole had actually broken through the outer surface. It was immediately apparent that inspection of the blocks was required to ensure that sufficient wall thickness was present. Since no mechanical gage could be used in this application a radioisotope gage was devised. A probe was tipped with a small quantity of radioactive ruthenium and inserted into the oil hole. The intensity of beta radiation received by a Geiger counter held against the outer surface of the block determined the thickness of the metal remaining. The energy of the source was such that the beta radiation was detected only if the wall thickness was less than 0.080 in. In this manner several hundred engine blocks were quickly inspected. The information made it possible to segregate satisfactory blocks and to locate the cause of the defect. It was determined that the difficulty was caused by a core shift and not a drill drift in that particular group of engine-block castings.

Gaging and Control—Perhaps the most useful application of radioactivity in industry today from an economic standpoint is that of in-process gaging and control. A variety of gages for continuous noncontacting measurement of mass per unit area can be constructed. Gages of this type consist of a beta or gamma source and a detector. The amount of absorption that the beta or gamma rays undergo when passing through the material to be measured is a function of the thickness. The relation between the intensity and the thickness can be determined experimentally or theoretically and may be represented by a calibration curve or directly on a meter scale or recorder. The detector may be a scintillation counter,

[5] Manufactured by Engineering Laboratories, Inc., Tulsa 3, Okla.

a Geiger tube, or an ionization chamber, depending on the application requirements. The continuous signal may be used to actuate circuits or relays which control the process. Initial studies in 1948 revealed numerous potential applications, but refinements in measuring and control instrumentation were required before suitable applications could be developed. Typical successful gaging and control applications of this nature are described as follows.

Example 1—Liquid-Metal Height in Cupolas: The first installation of this type made in the Ford Motor Co. was a gage for measuring the liquid-metal height in a batch-type cupola where such measurements were vital to efficient cupola operation.[6] The gage consisted of a radioactive source located on one side of the cupola and a Geiger-counter receiver located on the opposite side. The source and the sensitivity of the receiver were adjusted so that liquid metal could be clearly distinguished from slag, scrap, and coke. Although the gage and its auxiliary controls proved practical, the application was not pursued since batch-type cupolas were discontinued in favor of continuous-pouring cupolas.

Example 2—Radioisotope Stock-Level-Height Gage: The more extensive use of closed-top cupolas and water-cooled cupolas resulted in a need for an improved method for the determination of the height of stock in the cupola stack. Raw materials including metal scrap, limestone, and coke are introduced into the cupola in the melting of iron. If this stock level becomes too high or too low, the operation of the cupola is impaired. Conventional gages are of the pneumatic type in which probes are inserted into the stack at a predetermined level. If they meet an obstruction such as stock they immediately retract, indicating a full stack. In operation they are easily bent, become jammed, or misgage by entering a void in the stock. A gage was designed which used the absorption of gamma rays as an indicator of the presence or absence of stock at a level selected for optimum operating conditions. The gage proved reliable and successful and was coupled to the charging mechanism to permit charging control. The gage has proved so satisfactory that gages of the same design have been ordered for 12 additional cupolas from a commercial engineering company.[7]

Example 3—Steel-Mill Radioisotope Thickness Gage: The problem of thickness measurement of sheet steel during rolling has long been a problem in the steel industry. Contacting gages are inaccurate and costly to maintain. Noncontacting X-ray gages are more successful but are costly to maintain and operate. The radioisotope gage consists of a high-intensity, high-energy beta source located on one side of the sheet of steel with a radiation detector located on the opposite side. The intensity of the radiation is measured and can be related to the thickness. The signal from the ionization-chamber detector is fed to a balancing or bridge circuit. The error signal, or degree of unbalance, is a measure of the error in thickness of the steel, and this signal is relayed to the roll screwdown motors. The rolls open or close automatically to compensate for an off-specification thickness until the bridge becomes balanced. By this monitoring or feed-back device a more constant thickness of steel is maintained than was possible with previous contacting gages and manual correction. A gage of this type has been purchased from an outside supplier [8] and has recently been installed on a three-stand

Fig. 5—Eroded pipe section detected by Penetron

Fig. 6—Continuous thickness monitoring by beta-ray gage in steel mill

[6] *American Foundryman*, Vol. 15, June, 1949, pp. 35–39: "Radioisotope Gage Indicates Liquid Metal Height in Cupolas," by D. M. McCutcheon.
[7] Made by Radioactive Products, Inc., Ferndale, Mich.
[8] Manufactured by Industrial Nucleonics Corp., Columbus 12, Ohio.

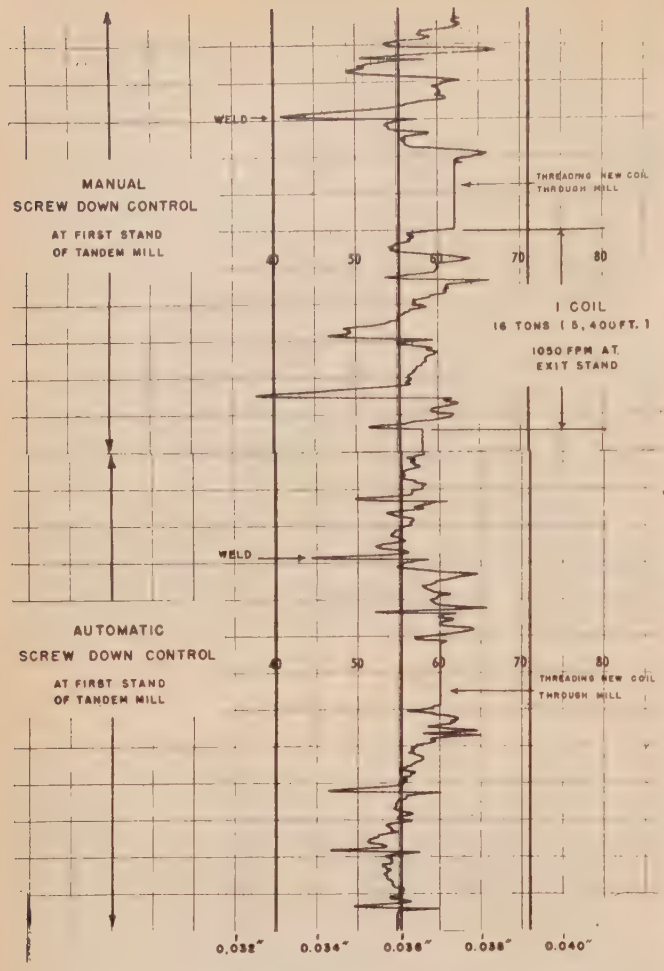


Fig. 7—Beta-ray gage charts showing improved quality with automatic control

tandem cold mill in the Ford Steel Division. Tolerance of ± 0.001 in. in 0.036-in. sheet steel can be maintained, for example, where as previous types of gages produced variations up to ± 0.004 in. The greater uniformity of thickness and the resulting increased square feet per ton of yield will reflect large dollar savings to the company. Fig. 6 is a photograph of the gage monitoring sheet steel. Fig. 7 is a typical strip chart showing the improvement in gage with automatic control. A similar gage is being installed in our chemical products plant to control the calendering operation in the manufacture of vinyl-coated fabrics. Improvements in quality through uniformity in thickness and the elimination of costly overcoating are anticipated.

Tracer Techniques—The use of tracers or "tagged" atoms is becoming increasingly important in obtaining knowledge of many materials and industrial processes. The materials in question can be made radioactive and their basic behavior studied as in wear measurements, electroplating, painting, and cleaning investigations. A radioisotope of a particular

Fig. 8—Adding radioactive sulphur to molten iron

element can be introduced at some stage of a process in a pilot operation and then followed and located with appropriate equipment. The tagged atoms giving off their identifying radiation can be distinguished from their neighbors. The number present can be determined from the intensity of the radiation, and the location of a decaying atom can be found by autoradiography. An autoradiograph is a photograph obtained when a film or plate is placed in contact with a surface containing a radioactive material and is taken by the radioactive atoms themselves as they decay, giving off particles and rays exposing the emulsion. Information is obtained about materials and processes which is often unobtainable by any other means.

Our first tracer study took place in 1949 when radioactive sulfur was added to molten iron in an investigation of nodular-iron formation. A photograph showing the addition of radioactive sulfur to a ladle of iron is shown in Fig. 8. The operation took place under the surveillance of industrial hygiene personnel. The results of this study were inconclusive, but they showed the potential of such techniques.

Since these initial experiments, an increasingly large number of tracer applications have been found. A few of these are listed below.

Example 1—Addition of Radioactive Cerium to Iron: A program to study the effects of additives to iron to form nodular iron is now under way. Since the amount of additive retained is very small there are few methods of studying its disposition adequately. It is known that, in general, deoxidizers aid the formation of nodular iron. Cerium, as an additive, is used to some extent for this purpose. A sample of cerium has been submitted for irradiation in a pile. The radioactive cerium will be added to the iron, and autoradiographs will be obtained. The study will provide valuable information on the cerium distribution and its influence upon the graphite formation.

Example 2—Location and Analysis of Boron in Steel By Neutron Activation: A novel tracer tech-

nique has been developed and used with partial success at the scientific laboratory to determine the location and distribution of boron atoms in steel. The method makes use of a nuclear reaction in which a slow neutron colliding with the nucleus of a boron atom is captured by the boron nucleus with subsequent emission of an alpha particle. The alpha emission is recorded on a special photographic emulsion. This entire process must be accomplished in a reactor because of the very short half-life of the radioactive boron.

In practice this is accomplished by placing a photographic film in contact with a flat piece of the metal to be tested. The metal and film are then placed inside a lead container and introduced into the thermal column of a research-type slow-neutron reactor. The container, while impervious to the intense gamma rays in the interior of the reactor, readily transmits neutrons which induce the radioactivity-producing reactions in the boron atoms.

In the preliminary experiments undertaken in the laboratory, the objective has been to explore the usefulness of this method as a means of obtaining analytical information on the amount of boron present in the steel. In addition, information as to whether the boron remains in solution homogeneously distributed throughout the crystal lattice of the steel or whether there is a tendency to segregate at the grain boundaries is obtained. Such information is potentially useful to the metallurgist and may be acquired using this technique. Further refinements in the experimental technique are necessary to obtain the more quantitative information that it is felt the method is capable of yielding. Fig. 9 summarizes graphically this particular application.

Example 3—Radioactive Tracers in Blast-Furnace

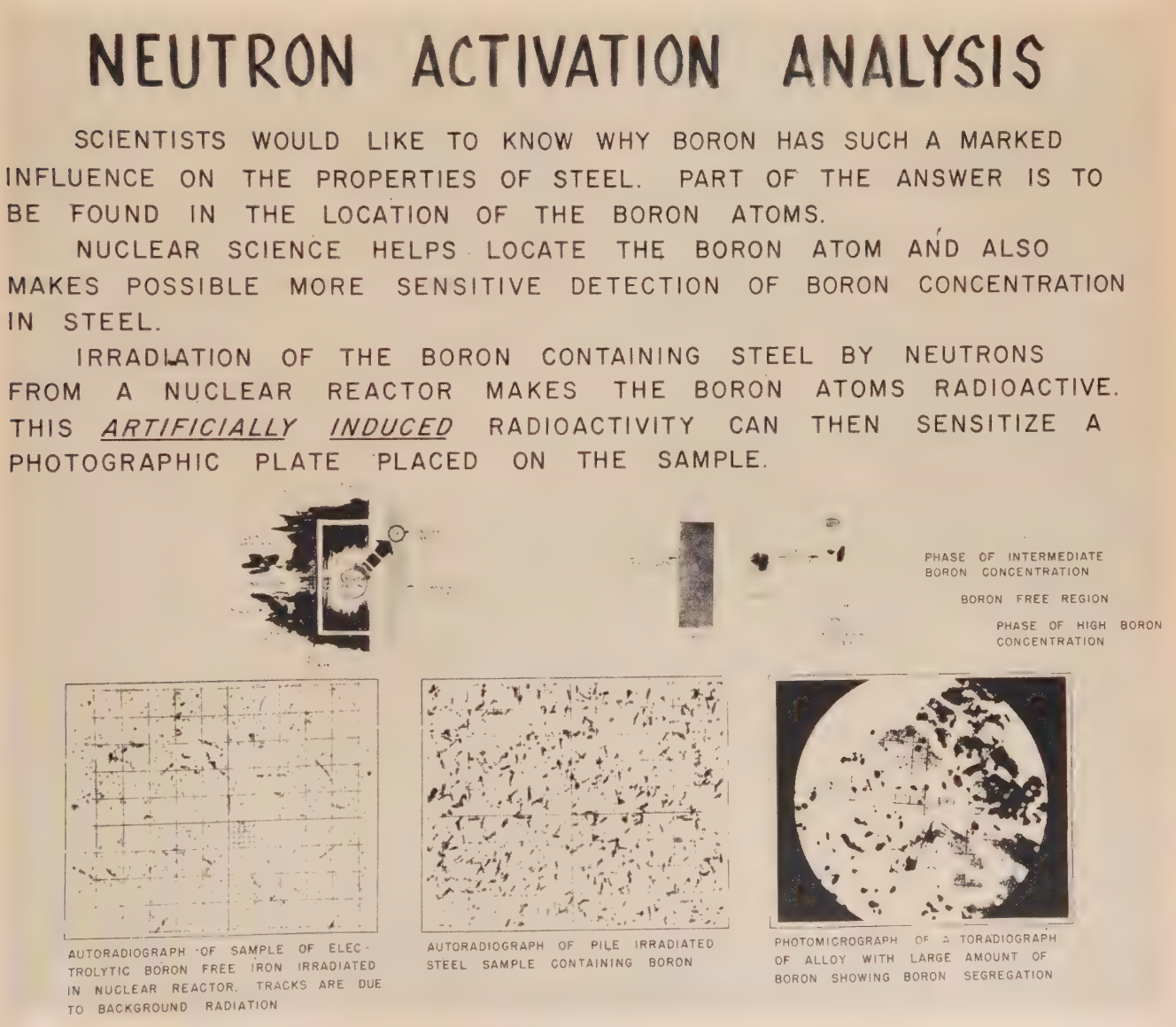


Fig. 9—Location of trace amounts of boron using a tracer technique

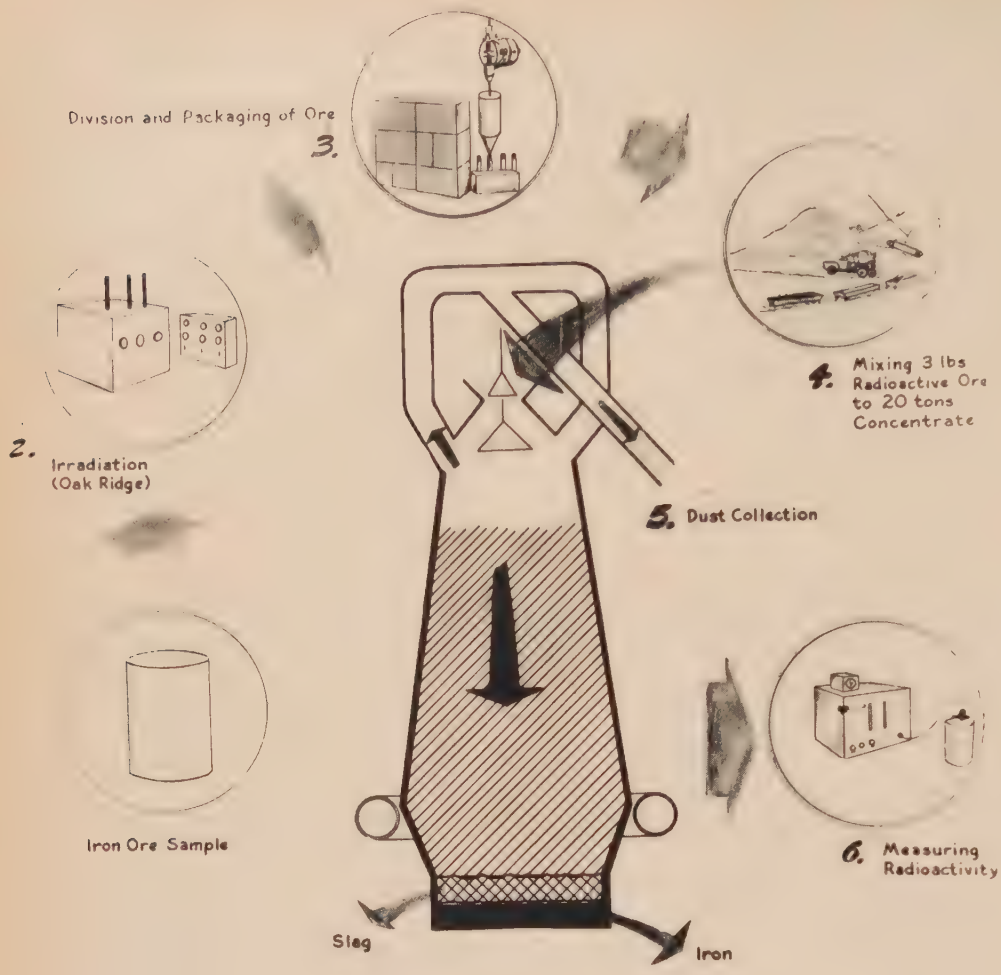


Fig. 10—Blast-furnace tracer study

Study: The largest tracer experiment ever conducted at the Ford Motor Co. was in the study of direct utilization of ore-concentrate fines in the blast furnace.[9] In this study, powdered iron ore which had been made radioactive by irradiation was introduced into several carloads of fines and fed to the blast furnace. The iron pigs, the dust in the dust catcher, and the slag were monitored while consumption of the tagged ore was taking place. It was found that about 60% of the fines were effectively used, the rest being lost up the stack. This confirmed the need for sintering operations prior to the utilization of this type of ore in the blast furnace. The total amount of radioactive material used in this study was approximately two millicuries. The radioactivity of the mixture introduced into the blast furnace was 10% above the normal background from cosmic rays and natural radioactivity. This level of radiation is only 1% of that approved by the Atomic Energy Commission for constant whole-body exposure. Fig. 10 summarizes the steps involved in this study.

Example 4—Sinter-Plant Production-Rate Study: The cycle time of the sinter plant was determined by putting a tracer in the ore at the beginning of the cycle and then detecting it at various stages. By this technique an accurate cycle time under various operating conditions was determined. Since only 80 microcuries of radioactive iron were distributed through half a ton of sinter material, radiation hazards were negligible. Nevertheless, air monitoring and personnel precautions were observed.

Example 5—Location of Obstructions in a Pneumatic-Tube System: One company foundry had a system of pneumatic tubes approximately three-quarters of a mile in length through which metal samples were rapidly dispatched to the laboratory for analysis, and the analyses returned. These tubes passed up and through the ceiling structure. At times obstructions developed in the tube, requiring long periods to locate and repair, owing to the relative inaccessibility of the tubes. To locate these obstructions more rapidly, a three millicurie capsule of cobalt 60 was placed in a

[9] "Radioactive Tracers in Blast Furnace Study," by I. W. Rozian and A. M. Smith, presented at Blast Furnace Operators Association, February 12, 1954.

carrier and shot through the tube. The tube was then traced by a man carrying a survey meter. The position of the obstructed carrier was quickly pin-pointed and repairs carried out. The location of the obstruction required less than 10 min.

Example 6—Wear Measurements: Wear measurement constitutes one of the most useful applications of tracer techniques.[10] Wear measurements have been made in the past on the basis of weight or dimensional changes. These tests were time-consuming and inaccurate. Using radioactive tracers, wear studies can be much more exact. The part in question, a tool or a gear, for example, is irradiated and placed under test. Subsequent wear in microgram units can be measured. Although no specific example can be reported related to materials and manufacturing processes at this time, a number of such applications are under consideration. The engineering research activity of the company has made extensive use of this technique.[1,2]

Additional Applications—Static neutralizers utilizing radioisotopes are finding increased use in industry to reduce fire hazards and dust problems. One application in the manufacture of plastic parts uses air guns incorporating an alpha source. Compressed air passing through the gum is ionized and dissipates the static charges accumulated on the plastic part. This efficiently removes dust and lint.

Processing by radioactivity has not found extensive industrial application, but a program has been undertaken to investigate such possibilities. High-intensity gamma radiation shows some advantageous as well as harmful effects on the physical properties of polymers. Evidence of grafting between different polymers shows promise. Rubber vulcanization, curing of resin cores, and paints are also under consideration.

Potential Applications

Many problems in the application of radioactivity which appear very difficult or impossible today may be solved by the introduction of new instrumentation tomorrow. Many uses of these methods which often give a unique solution wait only on the statement of the problem. Analogous operations in different processes often permit direct transfer of a solution from one problem to another.

Expanded utilization in the fields of radiography, nondestructive testing, and gaging and control are immediately apparent.

Tracer work in relation to materials and processing has only begun. For metallics, tracer studies relating to composition, heat-treatment, machining, forming, and wear will add considerably to basic understanding. For nonmetallics such as plastics, rubbers, and ceramics the same potential exists.

New techniques and new developments in instrumentation will give added breadth to information gained from the applications of radioactivity. Activation analysis with the gamma-ray spectrometer, for instance, provides the engineer with an analytical tool which has from 10 to 100 times the sensitivity of the direct-current spectograph.

Industrial access to reactors will permit studies in neutron diffraction and irradiation of materials as well as other fundamental studies not yet envisioned.

As these and a host of other yet unforeseen applications based on the unique radioactive radiations are applied to modern industry, the nuclear age clearly holds forth the promise of improved and cheaper processes, improved materials, a more careful control of existing processes, and a more complete understanding of the natural phenomena upon which the usage of our materials is based.

[10] SAE Transactions, Vol. 63, 1955, pp. 506–514: "Wear Rates of Gears by Radioactive Method," by F. L. Schwartz and R. H. Eaton.

ORAL DISCUSSION

Reported by C. R. Russell

General Motors Corp.

W. E. Jominy, Chrysler Corp.: Were the piston rings in the wear experiment modified in composition to improve the half-life of the induced radioactivity?

A. M. Smith, Ford Motor Co.: No.

Question: By what means were the piston rings made radioactive?

Mr. Tuttle: Samples were submitted to the Oak Ridge National Laboratory for irradiation to produce radioactive iron.

J. S. Thorpe, Allegheny Ludlum Steel Co.: Have you any suggestions for a technique whereby a wear problem involving no lubrication could be investigated?

Mr. Tuttle: One surface can be made radioactive, and the amount of radioactivity transferred to the mating nonradioactive rubbing surface will indicate the wear of the former.

P. E. Friend, Socony Mobil Oil Co.: How do you handle the rear-axle assembly problem where radioactive hypoid gears are involved?

Mr. Smith: Essentially standard procedures were used. All work with rear-axle assemblies has been done with low levels of activity from 10 to 30 millecuries. The intensity of the radioactivity of the component for an experiment is determined by the sensitivity of the detection equipment. Scintillation counters appear to be the most practical equipment for such work because of their high sensitivity.

R. S. Spindt, Mellon Institute: Do you feel that the low rates of wear measurable by radioactivity techniques show significant effects?

Mr. Tuttle: Because of the high sensitivity of the method, a large amount of data is obtained in a short time. A wide variety of experimental conditions can be explored under more closely controlled conditions than was heretofore possible. This is particularly true in engine studies during break-in and cold starting. Every wear experiment performed at Ford has provided very valuable information.

Automotive

Nuclear-Heat Engines and

APPLICATION of nuclear energy for civilian automotive uses has possibilities, these authors say. Nuclear power for automotive applications, they feel, is technically feasible now where size and weight are not prime considerations; where size and weight are major parameters, discoveries of new materials for construction of nuclear-power reactors must be made.

New materials are needed for reactor fuels, heat extractants, neutron reflectors, reactor construction materials, controls, and radiation shields which must have unique nuclear properties in addition to conventional engineering properties.

This paper presents nuclear automotive propulsion devices in terms of technologies now available. The necessary radiation-shielding mass and weight requirements are presented for an ideal point-source nuclear-heat-power engine.

FIVE to 10 years ago, few people would have predicted that in 1956 engineers would be building nuclear powerplants of 100,000–200,000 kw, or that by 1970, 25% of all new public utility powerplants will be nuclear powerplants. In England where the cost of fuel is high and the shortage of both domestic petroleum and coal is acute, the rate of growth of nuclear powerplants may exceed that in the United States. Therefore, it may not be out of place to look at the question of nuclear-powered vehicles at this time.

Nuclear-heat engines can be considered as powerplants for vehicles, locomotives, aircraft, and ships. Currently about 25% of the total energy consumed in the United States is for operation of these types of automotive powerplants.[1]

Some interesting questions come to mind. How small would a nuclear-power source be? How much shielding is necessary? Could one buy an automobile with a supply of fuel adequate for the life of the car included in the purchase price of the car? How much

[1] "Peaceful Uses of Atomic Energy," Report of Panel on Impact of Peaceful Uses of Atomic Energy to Joint Committee on Atomic Energy, Vols. 1 and 2. Pub. by United States Government Printing Office, Washington, D.C., January, 1956.

Associated High-Temperature Materials

F. L. Schwartz and H. A. Ohlgren, University of Michigan

This paper was presented at the SAE Summer Meeting, Atlantic City, June 6, 1956.

will nuclear fuel cost? Will enough of it be available? How will it be produced? What is to be gained in the way of car performance by having a nuclear powerplant?

It is conceivable that a nuclear-power source could be very small if only fissionable material needed to be considered. If the average life of a vehicle is 160,000 miles, the average speed 40 mph, the average horsepower 25, and the average hydrocarbon fuel cost 2.5 cents per mile, then the total cost of fuel for the life of the vehicle is $4000, and the energy used is 100,000 hp-hr or 75,000 kwhr. One gram of U-235 is capable of producing 1000 kwhr. In addition to requirements for critical mass for a given geometry and for fissionable atoms necessary to overcome neutron losses to absorbing materials, about 300 g or ⅔ lb of nuclear fuel is required if used in an engine of 25% efficiency for the life of the vehicle. To compete with the cost of hydrocarbon fuel at 2½ cents per mile for the conditions mentioned, the nuclear fuel cost must approximate $13 per g.

With our present knowledge of nuclear energy, fission fuel replaces fossil fuel in a nuclear powerplant. The nuclear fuel is merely a substitute source of heat energy. As long as heat energy is used to produce mechanical power, one is confronted with the second law of thermodynamics, which is at once discouraging. It tells us that unless we can use the heat energy at higher temperatures than present experience allows, we can convert only a limited amount of heat energy to mechanical energy. The maximum temperature that is useful is that determined by the metallurgy of engineering materials; concomitant factors such as heat removal, conservation of neutrons, sizes of reactor vessels, and attendant equipment coupled with neutron and radiation shields necessitate massive structures. Thus, achievement of a nuclear-heat source for automotive purposes is dependent largely upon successful developments of materials of construction.

However, before looking at materials, a few observations on possible engine powerplants are in order. One must accept the ambient temperatures of the atmosphere as the lowest practical temperature at which heat energy can be rejected in any kind of a heat-power cycle. If 40 F is this heat-rejection temperature, then the maximum efficiency for conversion of heat energy into work varies with the heat-source

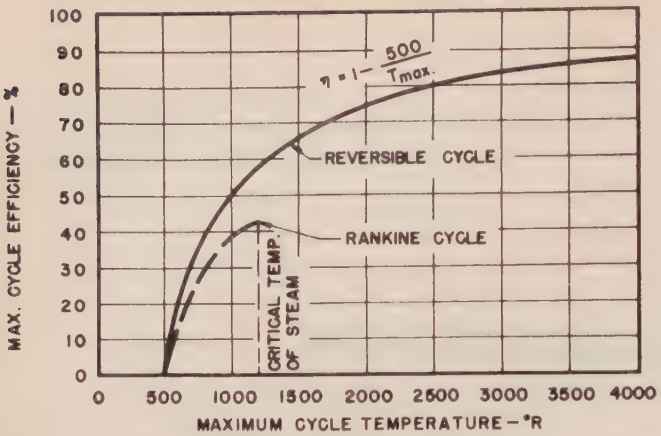


Fig. 1—Heat-power-cycle efficiency for ideal and actual cycles

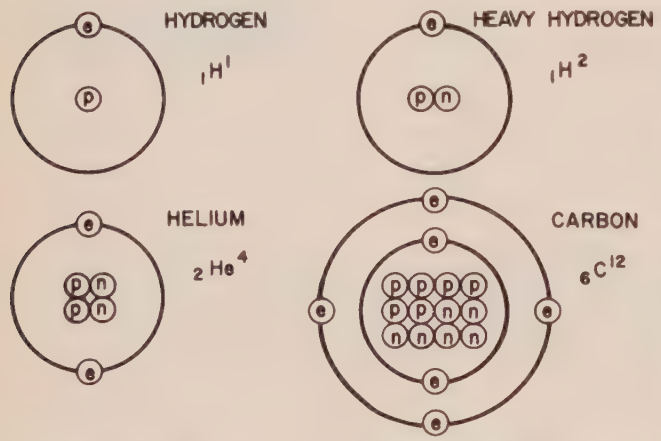


Fig. 2—Atomic structure of hydrogen, heavy hydrogen, helium, and carbon

temperature as shown in Fig. 1. This is the efficiency of any reversible cycle. However, all practical cycles are not reversible cycles, and therefore, they have a lower efficiency than the solid curve of Fig. 1. The dotted curve, for example, shows comparable ideal Rankine cycle efficiency using steam without superheat. Actual cycles with component efficiencies less than 100% will have cycle efficiencies as low as one-half of the ideal efficiencies shown in Fig. 1. The most efficient steam powerplant built in the United States has an efficiency of 37%; highly developed diesel engines have efficiencies of about 40%.

Thus, one is faced with the problem that 60 to 75% of the heat supplied to the powerplant, whether it be from combustion or fission, must be rejected to the atmosphere. In present reciprocating engines and open-cycle gas turbines this is easily accomplished by using air as the working medium and discharging the exhaust to the atmosphere, thereby replenishing the oxygen for combustion and rejecting the unavailable heat to the atmosphere. In any closed-cycle plant, whether using gas or a vapor, the unavailable heat must be transferred from the working medium to the atmosphere. This entails either large heat exchangers or large temperature differences. If large temperature differences are used, the heat-rejection temperature is made higher, and the cycle efficiency is lowered or a higher source temperature is required; the latter is affected by the ability of engineering materials to sustain higher temperatures. In any closed-cycle powerplant, the heat rejection equipment is likely to require more bulk and weight than the prime mover. This is especially true if air is the immediate receiver rather than water. The desirability of using a condenser on steam locomotives was recognized for many years, yet even here, where the carrying of large heavy auxiliaries was possible, it was never done. This practical difficulty alone could prevent the development of nuclear powerplants for many automotive applications. One solution is to use an open-cycle powerplant and transfer the heat from the reactors to a working fluid which is unaffected by radio activity. The second difficulty arises from the amount of shielding required for protection of people from radioactivity. Exposure to even minute amounts of gamma radiation over extended periods of time can have an eventual biological effect on personnel.

The Atomic Energy Commission reports that energy requirements for transportation can be expected to increase proportionally with the overall growth of energy needs. The commission has recognized, early in its programs, the possibilities for applying nuclear power to propulsion. This work, to date, has been limited to military applications.

In its recent report to the Joint Committee on Atomic Energy, the following conclusions were presented:

1. Nuclear energy, as applied to merchant ships, can become a significant source of power within the next 10 to 15 years, depending upon the relative competitive position of nuclear power as well as upon necessities of atomic propulsion for the American merchant fleet.

2. The use of nuclear-power systems for aircraft appears to be technically feasible in terms of foreseeable technology; the impact of nuclear power upon commercial aviation does not appear to be likely for many years.

3. The use of nuclear energy for locomotives has apparent technical feasibility now, but economically competitive locomotives are not foreseeable.

4. Atomic-powered motor vehicles such as cars, buses, or trucks require major technological breakthroughs not now in sight.

A series of mobile power reactors is currently being developed. Two submarine reactors have been built, and several more are in design stages. Reactors for aircraft are being developed by the Atomic Energy Commission and the Department of Defense.[2] The

[2] "Materials for Nuclear Power Reactors," by H. H. Hausner and S. B. Roboff. Pub. by Reinhold Publishing Corp., New York, 1955.

development of mobile nuclear-heat-power systems for aircraft, for merchant ships, for submarines, for locomotives, and for other vehicles faces a fundamental need for major breakthrough in new materials whose nuclear properties must be coordinated with the chemical and physical properties not now available to automotive engineering practices.

Those basic economic data which will determine the eventual competitive role of nuclear energy are some of the major parameters requiring definition and development. This paper is presented as an evaluation of current reactor technology and the associated high-temperature materials development in light of some requirements for automotive nuclear-heat-engine applications.

Although high-temperature materials have always been a problem in the design and construction of nuclear reactors, the problem has never been as critical as it now is in the development of automotive nuclear-heat engines. In this case, weight of the nuclear-heat-power source associated with a heat engine is of paramount importance. In addition to requirements for materials with unprecedented purity for successful operation of nuclear fuels, the same problems of containment of high-temperature operations which the automotive industry has met for many years in chemically fueled engines are present in nuclear-heat engines.

Progress in the development of automotive nuclear-heat engines depends to a major extent on achieving new types of reactor fuels, new types of nuclear reactor containers, new types of coolants and working fluids for heat engines, and new types of materials for biological shields and other components.

Nuclear Fission

An atom is the smallest particle of matter which retains its chemical and physical properties. However, it is made up of many smaller particles. An atom may be thought of as a minute solar system with a nucleus taking the place of the sun and surrounded by electrons which move about in orbits or shells much as planets move around the sun. The nucleus is made up of protons and neutrons. The neutrons differ from the protons in that they have no electrical charge, whereas the protons have a positive charge equal to the negative charge of an electron. The mass of a neutron or proton is roughly 1800 times the mass of an electron.

Several of the lightest and simplest atoms may thus be pictured as shown in Fig. 2. The mass of the atom is nearly all in the nucleus, and the total positive charge of the protons in the nucleus offsets the total negative charge of electrons surrounding the nucleus.

The chemical properties of materials depend on the electrons outside the nucleus. These contain relatively small amounts of energy, so that the energy of combustion when fuels burn is relatively small compared to the energy released when the nucleus is disrupted. In the early days of atomic studies, charged particles such as protons and alpha particles (helium nuclei) were accelerated to high velocities by passing them between enormous magnets. By such means, enough energy was given to such particles to penetrate and affect a physical change in the nucleus. Then, however, certain radioactive elements such as uranium were found to emit particles of their own volition with enough energy to disrupt the nucleus. One of these particles is the neutron.

Neutrons from a radioactive element may do one of several things. They may hit the nucleus of an atom and be absorbed, thereby creating a new isotope or element; they may cause the nucleus they hit to break into two fragments, thereby producing two new elements of lighter weight (this process is called fission); or the neutron may pass out into space and be lost.

Isotopes are atoms of the same element having same chemical properties but different masses in their nucleus. In Fig. 2, two isotopes of hydrogen are shown. Each has one orbital electron; but heavy hydrogen has one proton and one neutron in the nucleus whereas ordinary hydrogen has only one proton in the nucleus. The subscript denotes the atomic number and identifies the element; it is equal to the number of orbital electrons outside the nucleus and the number of protons inside the nucleus. The superscript is the mass number and denotes the total number of protons plus neutrons in the nucleus; it is proportional to the mass of the atom.

A graphical representation of the fission process is shown in Figs. 3 and 4. Fig. 3 illustrates an incident neutron colliding with a nucleus of uranium-235,[3] making the nucleus unstable as illustrated in Fig. 4. The products formed by the fission process are:

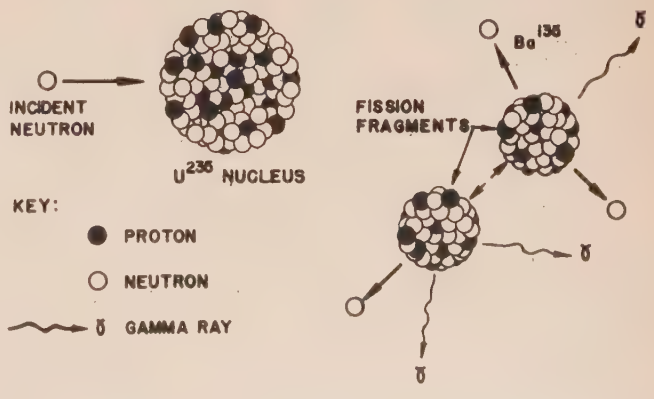


Fig. 3—Fission of U-235 nucleus caused by collision of incident neutron

3 "Metallurgy of Zirconium," National Nuclear Energy Series, VII-4, by Benjamin Lustman and Frank H. Kerze. Pub. by McGraw-Hill Book Co., New York, 1955.

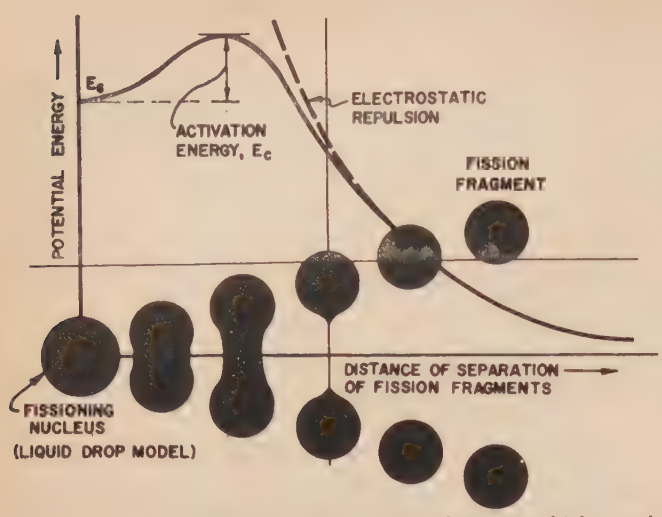


Fig. 4—Fission process showing energy changes which result in formation of nuclei of lighter elements

Table 1—Energy of Fission

	Mev per Fission	Kwhr per Fission
Energy available immediately after the fission process:		
Kinetic energy of fission fragments	162	
Kinetic energy of prompt neutrons produced in the fission process	6	
Energy of instantaneous gamma rays	6	
Energy from absorption of excess neutrons which are produced in the fission process and captured in nonfission processes by reactor materials	8	
Total	182	8.08×10^{-18}
Energy which appears as the fission products decay:		
Energy from fission-product gamma rays	5	
Energy from fission-product beta particles	5	
Total energy available	192	8.52×10^{-18}
Unavailable energy:		
Energy carried away by neutrinos accompanying the fission-product beta decays	11	0.488×10^{-19}

1. Energy equal to 192 Mev per atom, or about 3×10^{10} Btu per lb.

2. Neutrons equal to between two and three per fission (on the average of 2.5 for U-235).

3. Fission products—about 60 primary products which through decay produce about 300 different isotopes ranging in mass number from 80 to 60.

4. Beta particles which are high-speed electrons.

5. Gamma rays, which are extremely hard X-rays, have high penetrating power, and are hazardous to health.

As an example, consider the fission of the nucleus of uranium-235. This nucleus can be regarded as a cluster of 92 protons and 143 neutrons. If this cluster absorbs or captures a neutron, it becomes unstable. The energy which was available to hold the original cluster together is insufficient, and the unstable cluster breaks up into two or more new clusters. These new smaller clusters become the nuclei of lighter elements.

The production of two to three neutrons for each neutron captured by a nucleus means that, if additional atoms of uranium-235 are in proximity to the neutrons released by the original fissioning nucleus, they will capture such neutrons and also undergo fission. These atoms also release two to three neutrons per atom fissioned so that a chain reaction becomes possible.

A chain reaction of this type in which neutron production can be controlled can produce useful energy.

Control of a chain reaction is possible by controlling the number of neutrons which can be captured by fissionable atoms at any given instant. The number of neutrons can be controlled by insertion of suitable neutron absorbers into proper locations in a fissioning mass of fuel. The control of the amount or number of neutrons which are absorbed by materials that do not fission is a means of setting a predetermined power level of operation.

The probability that neutrons will be captured by nuclei depends on the speed of the neutrons. The velocity of neutrons emitted in fission is very high, too high to be useful in producing additional fissions in some elements. They may be slowed down by impact with lightweight atoms such as hydrogen, helium, or carbon. At lower speeds they readily penetrate the nucleus of atoms and cause fission. So called thermal neutrons are relatively slow-speed neutrons having velocities about the same as hydrogen molecules at room temperature. Thermal neutrons have the highest probability of producing fission in U-235.

The chance of collision between a neutron and a nucleus is defined by the nuclear cross-section. The nuclear cross-section is not the actual dimension of the nucleus but an area of influence around the nucleus such that if a neutron comes within that area capture or fission will take place. The effect of radioactivity on materials is therefore closely related to the nuclear cross-section. It becomes a nuclear property similar to such physical properties as density and stress.

Before a self-sustaining fission process is possible, a certain minimum quantity known as critical mass must be present. The amount required is determined by the probability that neutrons will be captured by fissioning atoms as compared to the probability that neutrons will escape and be absorbed elsewhere. Thus, a critical mass is interdependent upon a critical geometry. The energy released per atom fissioning is large. Table 1 indicates the amount and distribution of energy due to fission.

Materials for Nuclear-Heat-Power Reactors

Materials used for construction of nuclear-heat-power reactor systems require unique nuclear properties in addition to conventional engineering properties such as corrosion resistance, strength, ductility, stress, and thermal properties.

The engineer involved in development engineering and operation of power reactors might be classified under the following general headings: (1) nuclear fuels; (2) moderators and coolants; (3) reflectors;

(4) control rods; (5) vessels, mechanical equipment, piping, and the like; and (6) shielding.

Materials in these classifications are subject to interactions with radiation and nuclear particles. Alpha particles, beta particles, gamma rays, and neutrons produce chemical and physical changes in materals.[4,5,6] Such interactions are frequently referred to as radiation damage.

Nuclear Fuels—For technologies in the reasonable future, it can be expected that any mobile nuclear-heat-power source will depend upon one of the following materials as the fuel to sustain the fission process: (1) uranium-235 (found in nature); (2) uranium-233 (artificially made by neutron capture in thorium-232); (3) plutonium-239 (artificially made by neutron capture in uranium-238).

Reactor-fuel assemblages can be solids, liquids, and possibly gases. Most reactors undergoing present-day development have their fuels in solid form. Several promising new types of compact designs have their fuels in liquid form. If the fuel is in solid form, the elements may assume shapes as rods, tubes, and flat plates. The fuel elements are arranged so that coolants can extract the fission energy in the form of heat energy.

A diagrammatic arrangement of solid-fuel-element configuration which indicates passages for coolant, moderator, and space for control rods is shown in Fig. 5.

To prevent the fuel element in Fig. 5 from being corroded by the coolant and to contain highly radioactive fission products in the fuel element, the fissionable fuel is clad or jacketed. Examples of cladding materials which have suitable nuclear, thermal, and structural properties for nuclear-heat-power units are aluminum, zirconium, and stainless steel.

A liquid homogeneous fuel can be formulated in one of three ways, as follows: (1) an aqueous solution of a fissionable salt; (2) a fused salt in molten form; and (3) a molten metal of the fissionable material or some alloy thereof.

Certain reactor concepts employing high-temperature liquid fuels with suitable heat transfer means may offer considerable promise for several applications of automotive nuclear-heat engine systems.

Structural Materials for Heterogeneous Fuels—Reactor fuel elements are units which, when assembled in suitable mass and geometry in a nucleus, have contained fuel materials undergoing fission. Fast neutrons, beta particles, gamma rays, and fission fragments are generated in a fuel element during fission.[6] Materials used for fabrication of fuel elements must have certain nuclear, chemical, and physical properties.[2]

Requisite nuclear properties are low neutron absorption cross-sections and minimum changes due to radiation.

Desirable chemical properties are resistance to oxidation and corrosion.

Desirable physical properties are high mechanical strength, high heat capacity, and low thermal shock.

1. Zirconium: Zirconium can serve as a material to clad fissionable fuels. In water-cooled power-producing reactors, zirconium has desirable properties up to temperatures of 800 F.[7]

Zirconium has chemical properties similar to hafnium, titanium, and thorium. Table 2 gives a summary of properties for these materials.

As a fuel cladding material, it is not attacked severely by water or oxygen at high temperatures. The mechanical properties of zirconium metals and its alloys permit it to meet requirements for reactor fuels.

2. Stainless steels: Stainless steels have higher neutron absorption cross-sections than zirconium.

The chemical and physical properties of the stainless steels are suitable for power-producing reactors. Minor impurities influence these properties.

Irradiation effects in uranium and its alloys[8]

Table 2—Properties of IV-A Subgroup Elements

Element	Ti	Zr	Hf	Th
Molecular Weight	47.90	91.22	178.6	232.12
Principal Valence	4	4	4	4
Other Valences	3,2	3,2	3,2	3,2
Metal-Density, g/cc	4.50	6.4	12.1	11.4
Melting Point, C	1725	1857	2227	1730
Solubility of Oxygen in Metal	>1%	~1%	~1%	<0.1%
Melting Point of Oxides, C	1825	2677	2774	3050
Tetrafluoride				
Melting Point, C		~900	~900	~1000
Boiling Point, C	284	~900	~900	~1700
Tetrachloride				
Melting Point, C	-23	437	432	765
Boiling Point, C	136	331	317	922

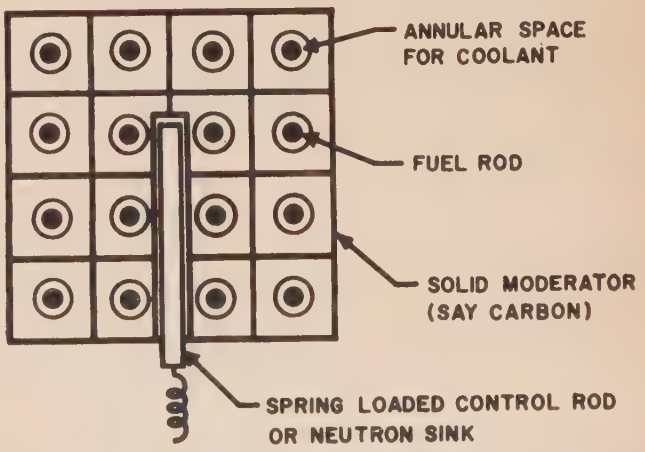


Fig. 5—Reactor core showing solid-fuel-element configuration with passages for coolant, moderator, and space for control rods

4 Report No. MDDC-962, Atomic Energy Commission, M.I.T., 1947, "Effects of Radiation on Materials," by A. O. Allen.

5 Report No. AECD-2810, Atomic Energy Commission, Oak Ridge, Tenn., 1947, "Effects of Nuclear Radiation on Materials," by S. Siegel and D. S. Billington.

6 "Introduction to Nuclear Engineering," by Richard Stephenson. Pub. by McGraw-Hill Book Co., New York, 1954.

7 "Elements of Nuclear Reactor Theory," by S. Glasstone and M. C. Edlund. Pub. by D. Van Nostrand Co., New York, 1952.

8 "Irradiation Effects in Uranium and Its Alloys," by S. H. Paine and J. H. Kittle. Presented at Nuclear Engineering and Science Congress, Cleveland, Dec. 12, 1955.

Table 3—Effect of Irradiation on Strength Properties of Standard 0.250-in. Diameter Uranium Tensile Bars[8]

Description of Specimen	Ultimate Strength 1000 psi	Ultimate Strength Change %	Yield Strength 1000 psi	Yield Strength Change %	Elongation, % (1-in. gage)	Elongation, % Change %	Young's Modulus 10^6 psi
Control Specimen	104		33		17		25
Irradiated at 120 C to 0.035% atom burnup	76	−27	71.5	117	0.36	−97.9	28
Irradiated, annealed 15 hr at 400 C *in vacuo*	65	−38	52	58	0.54	−96.8	
Irradiated, hot tensile at 285 C	71	−32	70	112	0.7	−95.9	12

Table 4—Moderators[9]

	Relative Moderating Ratio
H_2O	1.0
D_2O	87.0
Be	2.4
C (graphite)	2.5
Na	0.006 (poor)
NaK	0.002 (poor)

Table 5—Properties of Moderators

	Material H_2O	D_2O	Be	C	BeO
Density	1.00	1.10	1.84	1.60	2.80
Atomic or Molecular Weight	18	20	9	12	25
Atoms/cc or Molecules/cc	3.3×10^{22}	3.32×10^{22}	1.23×10^{23}	8.05×10^{22}	6.75×10^{22}
σ_a at 0.025 ev, barns	0.66	0.92 mb	9 mb	4.5 mb	9.2 mb
σ_s at 0.025 ev, barns	110	15	6.9	4.8	11.1
Epithermal σ_s, barns	46	10.5	6	4.8	9.8
Moderating Ratio	67	5820	150	169	180
Slowing Down Length, cm	5.7	11.0	9.9	18.7	12.0
Slowing Down Time, sec	10^{-5}	4.6×10^{-5}	6.7×10^{-5}	1.5×10^{-4}	7.8×10^{-5}
Albedo (Infinite)	0.82	0.97	0.89	0.93	0.93

produce dimensional changes due to thermal cycling and structural variables. Irradiation damage is minimized by proper fabrication techniques.

Table 3 presents some effects of irradiation on the strength properties of standard 0.250-diameter uranium tensile specimens.

Moderators—A large amount of the energy released during fission is in the form of velocity. In "thermal" power reactors, it is necessary to provide means for reducing the velocity of neutrons to such speeds that the probability increases that they will be captured by other fissioning nuclei.

Moderators are materials which slow down velocities of neutrons. Slowing down is accomplished by collisions between neutrons and the moderating material. The best types of moderators are those which have nuclei of masses which most closely approach the masses of neutrons. Thus, hydrogen is a good moderator.

In Table 4, relative moderating ratios are given for water, heavy water, beryllium, graphite, sodium and a sodium-potassium eutectic. For illustration, the moderating ratio for light water is 1.0. Thus, heavy water, beryllium, and carbon are all relatively better moderators from a nuclear viewpoint than light water, whereas sodium and sodium-potassium liquid metals are very poor moderators. Some general properties of materials found to be good moderators are given in Table 5 (footnote 10, p. 60).

Coolants—Heat removal from a nuclear-heat-power source is one of the controlling engineering parameters for automotive nuclear-heat engine systems. In reactor development programs underway, the media used for coolants would be water, air, and liquid metals.

The heat produced per unit volume in a nuclear-heat-power unit can be made as high as permitted by maximum working temperatures for construction materials and reactor fuels. Heat transfer surfaces can control the size of a nuclear-heat source. Thus, a coolant which has low probabilities for neutron capture must have the ability to transfer heat well and to withstand high temperatures.

Some properties of coolants are given in Table 6 (see footnote 10, p. 66). Table 7 [9] gives some heat transfer data for reactor coolants as a function of flow velocity.

Some of the coolant properties which require consideration for a specific reactor are given in Table 7. For the four possible coolants listed, the first column lists the relative absorption cross-section for neutrons, assuming heavy water = 1.0. The heat transfer coef-

[9] "Some Engineering and Economic Aspects of Atomic Power," by Leslie H. Landrum and Sinesio A. Zagnoli. Pub. by Spencer Chemical Co., Kansas City, Mo., 1956.

Table 6—Properties of Coolants

Coolant	Thermal Neutron Cross Section: Microscopic (barns) Absorption	Microscopic (barns) Scattering	Macroscopic Absorption Cross-Section at 650 F (cm^{-1})	Melting Point, F	Boiling Point, F	Operating Pressure, psia	Density, lb/cu ft 100 F	500 F	1000 F	Specific Heat, Btu/lb-F 100 F	500 F	1000 F
H_2O	0.802	164	0.0079	32	212	14.7	62.0			0.9976		
D_2O	0.92 mb	15.3	0.000037	38.87	214.7	1500		48.6			1.165	
Na	0.45	4	0.0074	208	1621	14.7		55.3	51.1		0.3150	0.3005
Na-K alloy (44% K)	1.1	3.2	0.0113	66.2	1518	14.7	5.61	52.9	48.7	0.2733	0.2583	0.2486
K	2.5	1.5	0.0166	147	1400	14.7		48.7	44.6		0.19	0.18
Bi	0.015	9.0	0.00059	520	2891	14.7			608			0.0369
Bi-Pb alloy (44.5% Pb)	0.17	9.9	0.0021	257	3038	14.7		646	624		0.035	0.035

ficient (film coefficients) which can be achieved for practical fluid velocities with suitable allowance for fueling are given in the second column. Considering differences in densities for comparable hydraulic heads, the relative pumping powers for each of the four coolants are given in the third column.

Since economics are important considerations in industrial-reactor design, engineers must consider the relative costs of the various materials.

Neutron Reflectors—Neutrons produced in the fission process are used in a nuclear-heat-power system for sustaining the nuclear chain reaction and for control. As the size of the nuclear-heat source is reduced, the ratio of surface to volume of the critical geometry increases. Thus, the probability of neutrons escaping from the system increases.

A reflector material is one which permits the neutrons, tending to escape from critical volume, to collide without absorption in such a manner that the neutrons return to the critical volume at the energies required to prompt fission.

Reactor Control—Control of a chain reaction at desired power level is accomplished by controlling neutron absorptions at rates which do minimize power surges, but do not shut down the reactor.

Control of neutron absorption can be achieved by controlling amounts of fission products at any given time in the reactor by the insertion of control rods, "shim" and scram rods, and by controlling the amount of neutron reflection.

Safety and assurance of reactor control are enhanced for thermal reactors since delayed neutrons are produced in sufficient quantities and over long enough periods of time to permit operation of control mechanisms.

Structural Materials—Choice of reactor structural materials depends upon size and type of reactor, the intended service, and operating temperature. Reactors which operate with thermal neutron energies can use only material with low-absorption cross-sections.

Table 7—Coolant Properties[9]

Coolant	Relative Absorption Cross-Section	Heat Transfer Coefficient	Pumping Power for Equivalent Heat Removal/F[a]	Cost per lb
H_2O	215	1000	1	$0.10/1000 gal
D_2O	1	1000	1	$28
Na	200	5000	3	$0.16
NaK	305	4000	5	$2

[a] Same temperature difference assumed.

Table 9—Neutron-Shielding Data

Neutron Energy, Mev	Shielding Material	Σ cm^{-1}	Thickness to Reduce Intensity by 1/10, cm
1	Hydrogen (in water)	0.281	8.2
	Oxygen (in water)	0.268	8.6
	Lead	0.178	13
10	Hydrogen (in water)	0.064	36
	Oxygen (in water)	0.050	46
	Lead	0.165	14

Reactors which operate with fast neutron energies permit selection of a wide variety. The economic design of power reactors to operate at high temperatures limits the selection of structural materials.

In regions of the reactor where neutron intensities are high, expensive materials such as zirconium are required. The vessel containing fuel, moderator, coolant, and reflector materials can be of more ordinary materials, such as stainless-clad carbon steels.

Materials as known to engineers today require compromises for reactor size, operating temperatures, fuel inventories, and economic parameters.

Table 8 presents some properties of structural materials. These properties include thermal neutron absorption cross-sections.

Neutron Shields—For safe operation of a nuclear-heat-power unit, provisions are needed to absorb all neutrons that tend to escape from the nuclear-heat source. Such absorption of escaping neutrons is accomplished by neutron shields.

Neutron-shielding materials must have nuclear properties which "slow down" fast neutrons and absorb slow neutrons. The neutron absorption process generally results in release of gamma rays. Thus, neutron-shield materials are located and contained within materials provided for gamma shielding. Table 9 (see

[10] IP-115, University of Michigan, Ann Arbor, Mich., 1955, "Engineering Applications of Nuclear Energy," by D. L. Katz, H. A. Ohlgren, K. Dedrick, J. G. Lewis, and M. Weech.

Table 8—Properties of Structural Materials[10]

Element	Thermal Neutron Absorption Cross-Section, Barns	Density, g/cc	Melting Point, C	Specific Heat, cal/g-C	Thermal Expansion Coefficient, Per Cx10^6	Modulus of Elasticity, 10^6 psi	Hardness, Bhn	Tensile Strength, 10^3 psi	
								Annealed	Cold-worked
Aluminum	0.22	2.70	660.2	0.22	24	10	20–25	13	24
Beryllium	0.010	1.85	1300	0.5	12	42	110	45	
Magnesium	0.059	1.74	651	0.25		6.5	50	32–46	32–50
Molybdenum	2.4	10.2	2620	0.065	5.5	40–50	147	100	250
Nickel	4.5	8.9	1455	0.11	13	30	75	47	125
Tantalum	21	16.6	2996	0.036	6.5	27	75–125	50	125
Titanium	5.8	4.5	1725	0.129	8.5	16.8	200	80	122
Vanadium	4.8	6.02	1735	0.12		20–22	260		
Tungsten	19	19.3	3410	0.034	4.0	50–60	260	50	300
Zirconium	0.18	6.5	1830		5.0–5.8	12		35	85
18-8 Stainless Steel (Fe, Cr, Ni, Mn, Nb)	2.9	7.92	1400–1420	0.12	16.7	29	160	90	
Inconel "X" (Ni, Cr, Fe, Ti, Nb)	4.1	8.3	1400–1420	0.11	13.9	31	200–400	160–180	

Table 10—Some Structural Materials for Gamma-Ray and Beta-Particle Biological Shields[10]

Category	Composition	Density $H_2O = 1.0$
Concrete	Cement, Sand, Gravel	2.3
Barytes Concrete	Cement, Barytes	3.5
Iron Ore Concrete	Cement, Iron and Iron Aggregate	6.0
Cast Iron		7.85
Lead		11.2
Uranium		19.0
Thorium		11.4

footnote 10, p. 77) indicates some satisfactory neutron-shielding materials and the thickness of shields needed to reduce neutron intensities to $\frac{1}{10}$ the value of incident neutrons.[11]

Gamma and Beta Shielding—The fission products and "capture" products resulting from fission are highly radioactive. The radioactivity is released in the form of gamma rays and beta particles. Reactors, therefore, must be shielded (biological shields) to protect life.

The absorption of beta particles (high-speed electrons) is accomplished by many materials with relative ease. The only known material capable of intercepting gamma rays, the high-frequency, short-wave electromagnetic waves similar to X-rays but much "harder," is dense matter.

Table 10 presents some structural materials used as gamma and beta shields and the reported densities in g per cm.

The greater the power output of a nuclear-heat engine, the greater is the mass of shielding matter that must be provided for tolerable health dosages at the surface of a shield.

Even though it might be possible to develop a nuclear-heat source and powerplant, which is light in weight and small in size, the size and weight of shielding materials surrounding the nuclear-heat engine are enormous by comparison.

Applications to Automotive Nuclear-Heat Engines

Problems of research, development, engineering, construction, and operation of automotive nuclear-heat engines are comparable in many respects to most automotive engineering programs. With present-day understandings of the fission mechanisms associated with thermal energy production, the nuclear-heat reactor can be considered to be a heat source which takes the place of the heat sources now using chemical fuels.

Nuclear-Heat Engine Parameters—Fig. 6 presents a well-known diagram of the relationship of the heat source to other components of a system, illustrating the ideal or theoretical cycle of the simple automotive powerplant in which the heat source is a nuclear device. The system is comprised of: (1) the nuclear-heat source which replaces the combustion unit; (2) the engine or turbine in which work is done by the expansion of the working fluid; (3) a heat exchanger or heat sink provided to reduce the temperature to permit recompression or pumping by means of (4) a pump or a compressor arranged so that the working fluid can be turned to the nuclear-heat source.

The efficiency of any heat powerplant is dependent on relationships of pressure ratios and enthalpy changes of the system. The enthalpy changes of the system in turn depend upon the temperature differences of the working fluid entering and leaving the components of the powerplant under consideration. The maximum temperature of the cycle establishes the maximum temperature required for the nuclear-heat source, so that an efficient heat engine cycle may be attained. Thus, the nuclear powerplant engineer is confronted with problems of correlating optimum nuclear engineering parameters with the conventional heat-cycle parameters including heat transfer, thermal stress, fluid flow, and construction materials. In addition to the materials requirements for the powerplant, the reactor powerplant engineer further needs knowledge and development of materials to contain the nuclear fission reaction, of radiation effects on coolants, moderators, reflectors, and safety shielding materials. The resolution of these many problems simultaneously requires extending present-day engineering knowledge to include the necessary data and parameters which are peculiar to nuclear engineering.

The science of nuclear reactor design, therefore, has reached a point where physical concepts no longer are limiting the progress of power reactors, where basic engineering problems themselves will determine the future progress of nuclear-power reactor systems.

Some of the areas which engineers must consider for a nuclear-heat-power system are discussed.

1. *Heat Transfer Correlations with Nuclear Data*—Relationships of heat transfer mechanisms, radiation, conduction, and convection with nuclear reactions and reactor design parameters such as critical mass and volume involve considerations of materials employed for heat conduction at minimum temperature differences with transfer of the fission energy to a thermal power cycle. Structural materials and nuclear properties of coolants influence the size of the reactor and the power level which can be attained. Since minimum volumes at maximum heat transfer involve the use of high-temperature structural materials, one of the major parameters controlling reactor engineering design for automotive heat sources lies in the development of materials possessing usable physical, chemical, and nuclear properties at high operating temperatures. The extraction of heat from a nuclear reactor to a coolant and the conversion of heat to useful power is affected by maximum temperatures, thermal stress, corrosion of materials, and neutron and radiation damage.

2. *Minimum Pumping Power*—Power require-

[11] Brookhaven National Laboratories Report 325, Upton, N.Y., July, 1955, "Neutron Cross-Sections," by D. J. Hughes and J. A. Harvey.

ments for pumping of coolants and recompression of working fluids can be significant as well as the control reactor design conditions. Pumps may require remote operation, remote maintenance, and/or special decontamination techniques. Mechanical developments for materials, seals, and shafts are also needed for the improvement of pumps. Drivers attendant with high efficiencies are important reactor design problems. The pumping power requirements for sustaining power level in a given nuclear-heat-power reactor must be correlated carefully with the power output of the automotive nuclear-heat engine.

3. *Temperature Levels*—A nuclear reaction can produce high temperatures. These temperatures can reach millions of degrees if the chain reaction is uncontrolled. In a controlled nuclear reaction, the maximum temperature which can be attained is fundamentally limited by construction materials as well as by the fundamental properties of the coolants used for transfer of energy. The construction materials can be those which contain the fuel in heterogeneous fuel elements, and the container materials for the reactor fuels, coolants, and controls.

4. *Nuclear and Thermal Stabilities of Materials*—The following characteristics of materials used in nuclear powerplants must be considered:

1. Ultimate strength
2. Yield strength
3. Stress-rupture
4. Creep
5. Fatigue
6. Elongation
7. Reduction in area
8. Hardness
9. Impact
10. Oxidation
11. Corrosion
12. Damping capacity
13. Relaxation
14. Modulus of elasticity
15. Thermal conductivity
16. Thermal expansion

Each of these properties in one way or another is influenced by nuclear interreactions and radiation damage. Theories and basic data for predicting what influence radiation has on such materials require much development. In some cases, the effects of radiation are seriously detrimental whereas in other cases certain improvements result. Some of these characteristics follow.

1. *Annealing*—A solid lattice subjected to radiation damage becomes unstable thermodynamically. If the atoms are free to migrate, they may gradually return to their former positions with accompanying evolutions of heat and reductions in hardness of materials. Such a process is known as annealing and becomes a function of temperature. Certain radiation-damage effects on materials are irreversible and cannot be removed by annealing. These irreversible effects might include accumulation of fission products in the fuel, induced radioactivity produced by absorption of neutrons and subsequent decay to new types of nuclear species, and accelerated chemical decomposition of the material as a result of radiation intensities. The combination of these may cause dimensional changes resulting in deformation of the material.

2. *Changes in Hardness and Strength*—In general, the effect of radiation on a metal is to increase its hardness and shear strength.[6] This effect is similar to cold-working. The result may be due to distortions of crystalline lattices which retard slippage of one crystal plane over another. Such effects are not appreciable in metals used for fabrication of reactor vessels and associated mechanical equipment, piping, and controls. The engineering test reactor will do much to further understanding of radiation effects on metals when neutron intensities exceed 10^{15} neutrons per sq cm per sec and high-energy radiation of about 10^{9} curies.[12]

3. *Effects in Electrical Conductivity*—Certain experimental investigations [4,5] have been conducted in this area. The electrical conductivity of a material is essentially the product of the number of electrons which can carry current and the mobility which the electrons can acquire per unit electric field strength. These effects are influenced by temperature, crystal lattice structure, and the dimensions of the crystal itself. Radiation tends to convert an ordered structure to one that is disordered. Consequently, electrical resistivity of materials tends to increase under radiation. This effect has proven to be one of the measuring sticks of determining radiation effects on materials.

4. *Effects on Thermal Conductivity*—Experimenters have reported that thermal conductivities tend to decrease due to effects of radiation.[13]

5. *Effects on Nonmetallic Materials*—Radiation

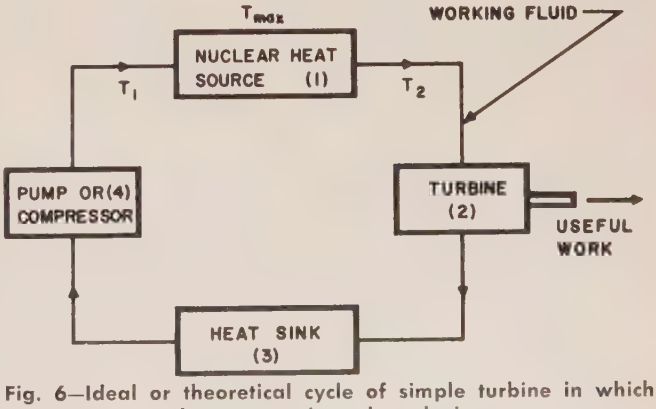
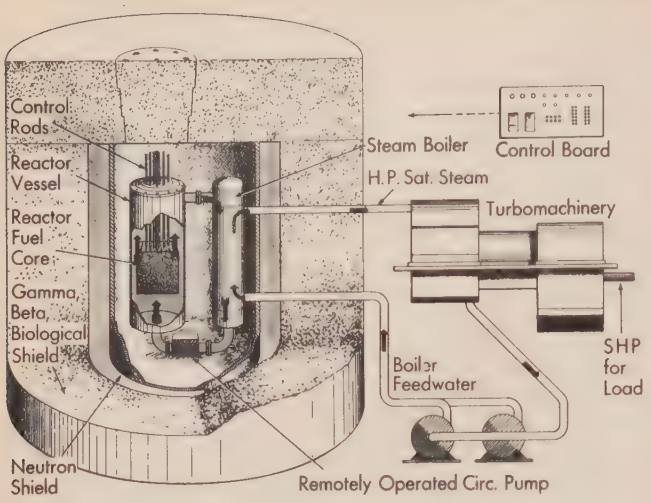


Fig. 6—Ideal or theoretical cycle of simple turbine in which heat source is nuclear device

12 "Effects of Nuclear Reactor Radiations on Structural Materials," by J. F. Zartman. Presented at Nuclear Engineering and Science Congress, Dec. 12, 1955.
13 *Proceedings of the Royal Society*, Vol. 208, 1951, pp. 90–108: "Thermal Conductivities of Some Dielectric Solids at Low Temperature," by R. Berman.

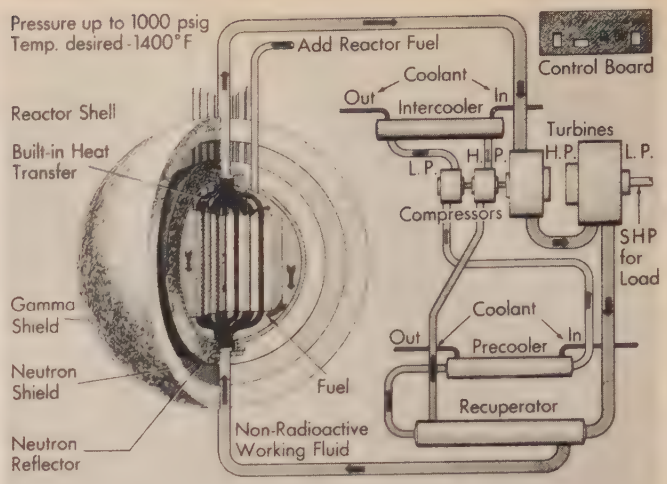


Pressurized Circulating Water Type

Fuel	—Highly Enriched Uranium
Fuel Assembly	—Zr - U
Moderator & Coolant	—Light Water
Neutron Shield	—Water
Gamma Shield	—Lead

Approximate weight of Nuclear Heat Engine with shield might be reduced to 100,000 lbs. (calculated). At 20 pounds per SHP, minimum size engine would be 10,000 SHP. At 25% efficiency, heat generation would be 29.8 megawatts.

Fig. 7—Nuclear-heat powerplant of pressurized circulating water type



Fuel Solution	—Possibly U 235 in Bismuth
Moderators	—Carbon
Coolant	—Helium (Maybe)
Reflector	—Carbon
Neutron Shield	—To be discovered
Gamma Shield	—To be discovered

Approximate weight of Nuclear Power Plant might be 50,000 lbs. At 20 pounds per SHP, minimum size engine would be 2500 SHP.

Fig. 8—Nuclear-heat powerplant of homogeneous liquid-metal-fueled type

Table 11—Spherical Shield Required for Point Nuclear Reactor

Assumed Radiation Dose Rate at Surface—6.25MR/hr
With Maximum Dosage of 36R/year
Neutron-Shield Mass And Size are Assumed Negligible

	Thickness, ft		Weight, lb	
Heat Power, megawatts	1	100	1	100
Shield Material				
Water	17.7	22.2	1,480,000	2,880,000
Concrete	9.18	11.3	463,000	870,000
Aluminum	7.91	9.65	352,000	642,000
Iron	2.83	3.46	45,400	83,600
Lead	1.59	1.89	12,200	20,600
Tungsten	1.012	1.21	5350	9100
Uranium	0.972	1.16	4550	7760
Material "x" (unknown) — Density 30 g/cc	Maybe 0.3	?	Maybe 250	?

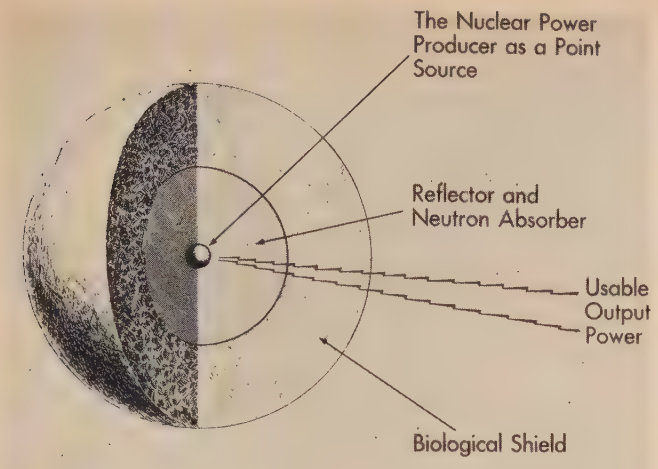
effects upon organic materials and aqueous materials are severe. Engineers must consider carefully such materials as gaskets, electrical motors, power wiring, lubricants, and any other components using nonmetallics. Intense radiations are being considered as catalysts and as energy of activation promoting chemical reactions, food preservation, and medical applications.

6. *Other Changes*—As maximum temperatures are achieved for various materials used in nuclear-heat-power reactors, much work will be needed to determine influences of radiation on all of the properties and physical characteristics that are listed.

Nuclear Steam Generators—Most water-moderated and -cooled reactors can be classed in this category of nuclear steam generators and associated steam turbomachinery. The types of reactors which are currently under development for the generation of saturated steam with subsequent conversion to power are the following ones:

1. *Pressurized Water System*—Pressurized water reactor systems (Fig. 7) are of the heterogeneous type, operating in the thermal range of neutron energies. For propulsion purposes, high enrichments of fuels are required for compact design. Water at high pressures, with the temperature of the water approaching the saturation pressure, is circulated by means of a pump through a high-pressure vessel in which a reactor fuel is located. The thermal energy removed by the circulating water is used to generate steam.

2. *Boiling Water Reactor*—A modification of the pressurized circulating water type of reactor is termed a boiling water reactor. In general, this reactor differs from the pressurized water type by producing steam directly from the reactor vessel, thus reducing the design pressures from about 2000 psig.



Probable Required Temperature above 5000° F for Nuclear Reaction

If known biological shields cannot be reduced in weight, and present tolerances of 36 roentgens per year is maximum human dose; minimum weight of automotive nuclear heat engine will be about 12,000 lbs. At 20 pounds weight per shaft horsepower, smallest usable power output will be 600 horsepower.

Fig. 9—Ideal point-source nuclear-heat-power engine

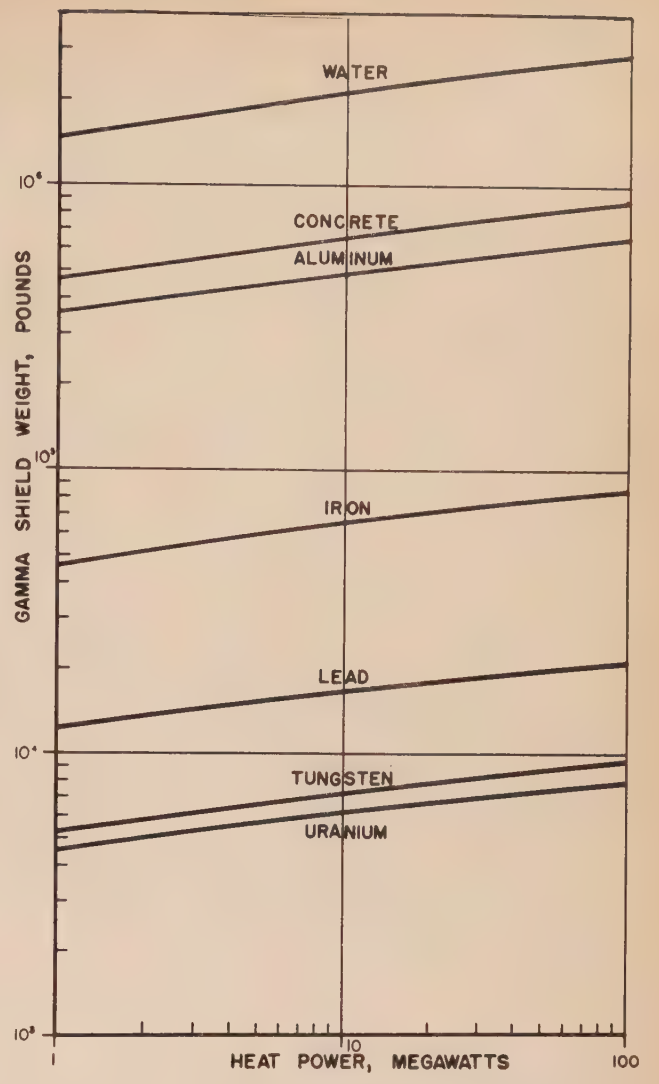


Fig. 10—Minimum weight of spherical gamma shield for point-source reactor

3. *Aqueous Homogeneous Type*—A method by which fuel elements fabrication and a part of the high-temperature problems of heterogeneous assemblages can be avoided is the dissolving of the fuel in an acid solution of water, forming a fissionable salt, an example of which is uranyl sulfate. When an adequate concentration of fuel is present in solution in a proper geometry, the fission reaction takes place at suitable temperatures and pressures. Heat is transferred from the liquid fuel so that high-pressure, saturated steam can be produced for use in turbomachinery.

Liquid-Metal Reactors—Temperatures of nuclear-heat sources can be increased while reducing design pressures by employing molten metals as coolants. Several types of liquid-metal reactors are currently under development.

1. *Heterogeneous Liquid-Metal-Cooled Reactors*—This nuclear-heat-power source employs a molten metal, circulating through a heterogeneous assemblage of fuel for removal of thermal energy. Molten metals such as sodium and sodium-potassium eutectics are being used. The molten metal exchanges its heat energy with a suitable working fluid for heat-engine operation. Such working fluids may be gases for gas-turbine operation, steam for steam turbomachinery, and binary materials. The maximum temperature and heat-engine efficiency for heterogeneous liquid-metal-cooled reactors are basically limited by the maximum temperatures which can be achieved for reactor-fuel element design.

2. *Homogeneous Liquid-Metal Reactor*—A recent achievement in development by the Brookhaven National Laboratories has indicated that a homogeneous liquid-metal-fueled reactor has promise for application. (See Fig. 8.) Such a reactor has the fissionable fuel dissolved in a liquid metal. An example is to dissolve uranium-235 in bismuth metal. The present level of temperature which might be achieved in this type of a reactor dependent upon structural materials, is reported to be 550 C.[14] By improvements of corrosion resistance of materials which can contain uranium-bismuth solutions, possi-

14 Brookhaven National Laboratories Report 152, Upton, N.Y., January, 1952, "Brookhaven Nuclear Reactor: Theory and Nuclear Design Calculations," by I. Kaplan and J. Chernick.

bilities for higher temperature operation are good. Depending upon the maximum temperatures which can be developed for container materials, this homogeneous liquid-metal reactor offers distinct possibilities in applying gas-turbine powerplants in conjunction with this nuclear-heat source.

Ideal Point-Source Nuclear-Heat-Power Engine

When considering heat engine cycles wherein the nuclear-heat source and heat conversion devices are theoretically ideal, it is possible to evaluate the minimum weight for various gamma-radiation shielding materials. Fig. 9 illustrates this ultimate nuclear-heat engine. Fig. 10 shows such weights plotted against heat power in megawatts. This figure assumes the nuclear-heat engine, including the heat source, power extraction device, and neutron shield, to be a point source. Under such conditions, the weight of an automotive nuclear engine will be controlled solely by available gamma-shielding materials. Table 11 presents the total weight of gamma shields for presently available materials of construction.

If it is assumed that chemically fuel heat engines have weight-horsepower ratios no more than 5/1, it can be seen that the shaft-horsepower outputs reach a minimum for each material under consideration. Then, to apply an ideal nuclear-heat engine to shaft-horsepower requirements less than the figures indicated, major breakthroughs in shielding design and materials are required.

Acknowledgments

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APPENDIX

Terms and Definitions of Nuclear Engineering

Absorber: A material which has a high affinity for neutrons but does not fission as a result.

Albedo: A unit used to measure ratios of negative to positive neutron currents. It is the ratio of the number of neutrons reflected back to the number of neutrons entering a reflector material.

Alpha Particle: A radioactive particle consisting of helium nucleus with positive charge and consisting of two protons and two neutrons.

Atom: The smallest part of an element which retains its chemical and physical properties.

Atomic Mass Unit (AMU): A method of relating atomic mass to oxygen.
1 AMU = 931.8 Mev = 1.657×10^{-24} g

Atomic Mass: Relative weight of atoms when oxygen is 16.000. For practical purposes the mass equals the total neutrons and protons in the nucleus.

Atomic Number: The number of protons or positive charges in a given nucleus.

Barn: A method of expressing probability of nuclear interaction. Practically can be considered as "target" area where one barn = 10^{-24} sq cm.

Beta Particle: A positive or negative electron emitted from radioactive species.

Breeder Reactor: A device in which a controlled chain reaction takes place so that the production of fissionable atoms is greater than the number consumed.

Burnup: The percentage of fissioning fuel used in a controlled chain reaction. Includes the amounts that are destroyed or converted to other materials by neutron capture.

Chain Reaction: A nuclear reaction occurring so that sufficient numbers of neutrons are conserved to prompt fission in other atoms for sustained periods of time.

Coolant: A liquid or gas used for extracting thermal energy (heat) from a nuclear reactor.

Critical Condition: The zero power level of a reactor which sustains a controlled chain reaction.

Electron: A negative charged particle which weighs 9.107×10^{-28} g.

EV: An amount of energy required to transfer an electron through one volt of potential difference
1 EV = 15.2×10^{-23} Btu

Enrichment: Normally refers to increasing the properties of fissionable atoms to nonfissionable atoms.

Fertile Materials: Normally considered to be materials which on neutron capture result in eventual production of fissionable atoms.

Fissile Materials: Materials capable of fission with the energy ranges of neutrons present.

Fission: The nuclear process by which a heavy element on reaction capture splits up into two or more fragments.

Fission Products: The light elements, radioactive and nonradioactive, resulting from fission.

Flux: In nuclear interaction this is considered the product of the number of particles per unit volume and their mean velocity.

Gamma Rays: A short-wave electromagnetic radi-

ation similar to X-rays but much more intense. Energies range from 10 Kev to 10 Mev.

Half-life: The length of time for radioactivity to reduce to one-half value.

Ionization: A method by which an atom or molecule acquires an electric charge.

Isotope: Varieties of elements which have common chemical properties but whose atomic weights are different.

Kev: One thousand electron volts.

Mev: One million electron volts.

Moderator: A material of low atomic mass which is used to slow down neutrons without capturing them.

Multiplication Constant: The ratio of neutrons of one generation to the neutrons of a preceding generation.

Neutron: A particle with no charge and whose atomic mass number is 1. Its symbol is ${}_0n^1$.

Neutron Producer: A reactor which produces neutrons for isotope production.

Photon: A quantum of energy which is the smallest amount of energy traveling at the speed of light.

Poison: A material in nuclear reactors which absorbs neutrons for no useful purpose.

Positron: An electron with a positive charge.

Power Density in Reactors: The power produced per unit volume of nuclear fuel.

Proton: A particle whose atomic mass is 1.0 with positive electric charge.

Radiation Damages: Undesirable changes in structural, chemical, and physical properties resulting from nuclear radiation.

Radioactivity: The process by which an unstable atom releases energy in the form of alpha particles, beta particles, and/or gamma rays.

Reactivity "k": Reactivity "k" is equal to the ratio of

$$\frac{k_{ex}}{k_{eff}} = \frac{k_{eff} - 1}{k_{eff}}$$

The ratio establishes the control of the power level of a nuclear reactor
where:

k_{ex} = excess neutron multiplication factor

k_{eff} = effective neutron multiplication factor

When reactivity is negative, a power reactor becomes subcritical; when it is zero, the power reactor is under control; and when it is positive the reactor becomes supercritical.

Reflector: A material incorporated in and surrounding the reactor fuel which reflects neutrons at energies so that they are useful in fission.

Recovery of Reactor Fuels: Processes for recovery of reusable reactor fissile and fertile materials with adequate separation of such things as fission products and spent structural materials.

Roentgen: A standard unit of radiation dose. The quantity of X-rays or gamma rays which produce one electrostatic unit of electricity per cubic centimeter of air at standard pressure and temperature.

Shield: A radiation-absorbing structural material which reduces radioactivity and nuclear particles to levels which permit human operation within reasonable distance from radiation source.

Temperature Coefficient: The change of reactivity divided by a change in temperature.

$$\alpha_1 = \frac{\partial \delta}{\partial T} = \left(\frac{\delta - \delta_0}{T - T_0}\right)$$

When the coefficient is negative, an increase in temperature decreases reactivity; when the coefficient is positive, the increase in temperature increases reactivity.

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ORAL DISCUSSION

Reported by C. R. Russell

General Motors Corp.

J. B. Duckworth, Standard Oil Co. (Ind.): What is the total weight of a reactor plus shielding for a 200-hp engine?

H. A. Ohlgren: By reference to Fig. 7, it is estimated that the minimum weight of a small reactor powerplant of a conventional type would be about 100,000 lb. It is, therefore, quite clear that such reactors are not going to be used to power automobiles.

H. V. Nutt, Naval Engineering Experiment Station: What was the composition of the fuel in the original University of Chicago reactor?

L. R. Hafstad: This reactor was constructed from graphite blocks and normal uranium fuel in the form of metal slugs and lumps of the oxide.

L. D. Gilmore, International Harvester Co.: What prospects exist for the direct conversion of nuclear energy to electrical energy without the intermediate step of the heat cycle?

Mr. Ohlgren: If a reactor could be operated at 10,000 C, thermocouples could be used to convert 6 to 8% of the energy to electricity with the remainder of the energy being converted through a conventional heat cycle. There is no available method for orienting the electrons for the direct conversion of energy to electricity.

C. R. Lewis, Chrysler Corp.: How serious would a crash accident be with a nuclear powerplant?

Mr. Ohlgren: Crash accidents are a major problem with all mobile reactors. However, certain types have inherent control mechanisms—a negative temperature coefficient of reactivity—which reduce this problem. The principal hazards would result from the spreading of radioactive material from the core.

Question: What limits the power output for a given size reactor?

Mr. Ohlgren: Heat transfer usually limits the power that can be extracted from a given size reactor. The power level is regulated within this limit by control rods.

LABORATORY FATIGUE

THIS paper deals with endurance testing of spur gears in modified 4-square gear testing machines, also referred to as special gear testing dynamometers,[1] power-circulating gear testing machines,[2] and single-pair gear test machines.[3]

In machines of this type the test torque is locked within a closed system (Hopkinson coupling), so that the only external power required to run the machines is that necessary to overcome static and dynamic friction. The ability to impart high loads to the test gears with only a small driving force and without external power-absorbing equipment constitutes the principal advantage of modified 4-square gear testing machines over dynamometer equipment for testing gears. Initial costs, operating costs, and maintenance costs favor the modified 4-square gear testing machines.

Since the gears are mounted in special gearboxes, modified 4-square gear testing is sometimes referred to as testing under an "ideal" or "simulated" environment. With the same line of reasoning, gear testing in opposed-transmission rigs, in chassis dynamometers, and in laboratory dynamometers is considered testing under a "normal" environment. This differentiation is neither realistic nor fair to any of these test methods, since it pertains only to the gear mounting design and neglects equally important factors such as mass-elastic system of the complete testing unit.

Much of the confusion concerning the relative merits of gear testing methods undoubtedly results from the dual definition of the word "gear." In one usage a gear signifies a toothed wheel, or cogwheel. In another, a gear designates a train of mechanisms, including such things as bearings, shafts, cogwheels, and gearbox.

If we consider a gear to be a train of mechanisms, a simple mechanical analysis shows that the performance of any part within this train is dependent upon the influence exerted by the other parts. In fact, the performance of any part within the train is tied in with the mass-elastic system of the entire product in which the train is mounted and with the specific type of service to which the product is subjected. A cogwheel, outside of its gearbox, is only a cogwheel, no matter how accurately it is machined, what it is made of, or how it is heat-treated.

If we accept this concept of what a gear is, then we must accept service usage as the absolute method for evaluating the performance of a gear and of the cogwheels contained in a gear. We must admit that only the customer can conceive the type of service which the gear will receive.

Competition and consumer relations do not allow us to develop our products by service performance alone. It has been necessary to develop field and laboratory tests which accelerate our gear development programs, prove our products before they reach the customer and provide us with quantitative data.

The test which most nearly resembles service usage is field testing, since it involves the complete product and is done in "typical" surroundings. Field testing has long represented the final acceptance criterion for gears, despite the fact that field tests can be so time-consuming, expensive, and laden with variables that inadequate sampling and testing can easily result in later difficulties in service performance and with competitive product costs. Field test conditions are difficult to reproduce from one test to another. If the testing is done on a test track, correlation with service performance is obtained through experience

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[1] ASTM Transactions, Vol. 28, 1940, pp. 687–713: "Selection of Steel and Heat-Treatment for Spur Gears," by H. B. Knowlton and E. H. Snyder.

[2] "The 5-in. Center Gear Testing Machine," by H. D. Mansion. Pub. by Institute of Automobile Engineers, London, 1944.

[3] SAE *Journal*, Vol. 57, May, 1949, pp. 38–40: "Gear Testing Methods for Development of Heavy-Duty Gearing," by R. P. Van Zandt and B. W. Kelley.

TESTING OF GEARS

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and is no more absolute than correlation obtained through experience between other accelerated gear testing methods and service performance.

The many disadvantages of field testing have encouraged the development of laboratory gear testing methods which can be more closely controlled and which are not subject to outside weather conditions. These laboratory methods for testing spur gears include chassis dynamometer, standard dynamometer, opposed tractor, opposed transmission, modified 4-square and single-tooth bending-fatigue tests. None of these methods exactly duplicates service conditions or field testing. Each represents a compromise which must be carefully considered in analyzing test results. Each has its own set of variables, some of which are common to service variables and to other testing methods, and some of which are peculiar to itself.

One of the most important advantages of these laboratory testing methods is the greater control possible over test conditions. Test speeds, test loads, lubrication, and many other variables can be held within close limits. Another advantage is that automatic controls and safety devices can often be incorporated into the testing machines to allow the tests to be made with a minimum of labor and a maximum utilization of the 24 hr in a day.

Physically, chassis dynamometer testing most nearly resembles field testing because the complete product is being used in the test, including the source of power. This type of testing requires large power-absorbing equipment, which introduces mass-elastic problems into the test. In certain instances, such as shock-loading, simulation of important service conditions has been evidently proved by experience, but

IN the first part of this paper, Mr. Halgren discusses modified 4-square testing of gears in relation to service performance, field tests, and other accelerated laboratory test methods. It contains a short description and photographs of nine different designs of modified 4-square gear testing machines now in use.

Description of machines includes comments on mounting rigidity, mass-elastic system, load application, measurement and control, shut-off devices, and lubrication.

Mr. Wulpi discusses single-tooth fatigue testing which is used to measure the bending strength of gear teeth. One tooth at a time is tested in the manner of a cantilever beam. By successive testing, many teeth of the same gear may be tested, and endurance limits obtained if desired.

Results of these tests show that it is possible to determine clearly the effect of various metallurgical factors, such as material, heat-treatment, or surface-finishing treatment upon the bending-fatigue strength of gear teeth.

test reproducibility is not too good. The testing method suffers from limitations in maximum load which can be applied to a gear without shock loading. It also measures the strength of only the weakest link in the complete product. In general, it is slow and expensive.

Standard dynamometer testing rigs also require large power sources and power-absorbing devices, both of which are expensive and introduce their own mass-elastic problems into gear tests. Excellent accuracy in test loads and test speeds is possible. Overloads can be applied within the dynamometer's capacity. Only the weakest member in the gearbox is tested.

Opposed-transmission testing rigs and opposed-tractor testing rigs use the same locked-in torque principle that characterizes modified 4-square gear testing machines. As in dynamometer tests, the gear is tested as an assembly, and failure occurs at the weakest member, which may or may not be the one to be evaluated in a gear development program.

Returning to the concept of a gear as a train of mechanisms, we are forced to admit that modified 4-square gear testing and single-tooth bending fatigue testing of gears should more appropriately be called cogwheel testing methods. These methods bear the least resemblance to service usage of a gear. Even the mountings are different.

Fig. 1—High-capacity (24,000-lb tangential load) single-pair gear test machine showing (a) motor drive at right, (b) hydraulic torque applier unit on front shaft between gearboxes, and (c) individual, temperature controlled lubrication systems for each gearbox below base of machine

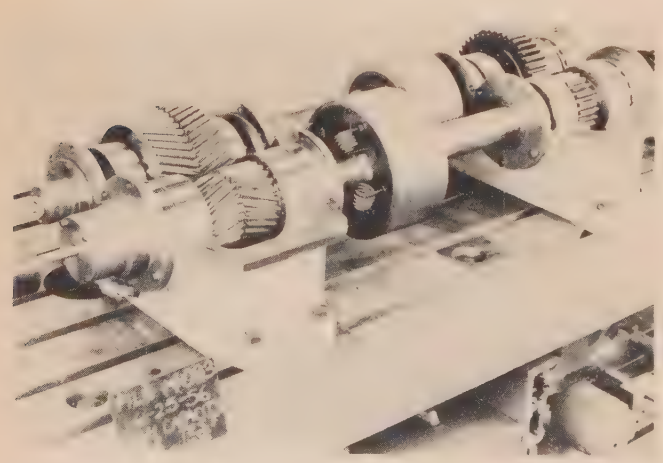


Fig. 2—Close-up view of testing machine seen in Fig. 1, illustrating (a) closely coupled bearing supports, (b) wide precision herringbone power return gears, (c) split, rigid gearbox design, and (d) hydraulic torque applier unit

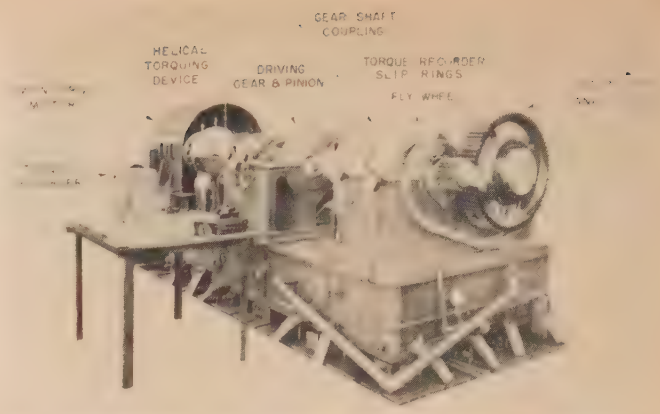


Fig. 3—Railway gear endurance test machine showing (a) overhung test gears, simulating actual mountings used in railway power units, (b) herringbone driving gears, (c) hydraulic torquing device, and (d) main drive unit

Lack of physical resemblance, however, does not preclude that these testing methods are no good and that erroneous data will be obtained from them. Many years of testing have proved otherwise.

Modified 4-square gear testing is a valuable tool for engineers and metallurgists who are concerned with gear durability and strength in relation to materials, heat-treatment, manufacture, design, and surface treatment. When used together with dimensional measurements, stress calculations, mounting rigidity data, metallurgical data, and experienced judgment, the results of modified 4-square gear tests enable them to select the most economical material and manufacturing processes for production of cogwheels which will function satisfactorily in almost all service requirements.

The value of modified 4-square gear testing is adequately attested by the continued expansion of testing facilities being provided within companies which have had experience with it.

Actually it is impossible to assess a comparative value between field testing and the various laboratory methods of gear testing which have been discussed. Field tests are primarily final acceptance tests. Laboratory tests whether conducted on individual cogwheels, partial assemblies, complete assemblies, or on the final product itself are development tests for evaluating gear materials, heat-treatments, design, manufacture, and surface treatments.

The International Harvester Co. uses all of these laboratory gear testing methods in its gear development programs. For example, the development of induction-hardened cogwheels for the Farmall Cub tractor was initially evaluated by modified 4-square tests and later corroborated by field tests.

Single-tooth bending-fatigue tests on shotpeened cogwheels have correlated field tests and service re-

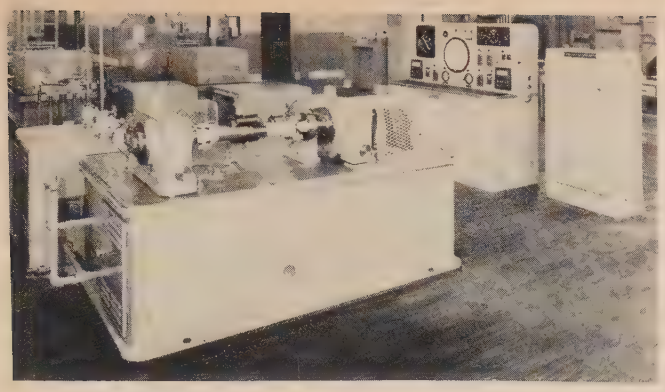


Fig. 4—Modified 4-square gear testing machine of 30,000-lb-in. capacity, showing (a) instrument cabinet at right, (b) testing machine in center foreground, and (c) temperature-controlled lubrication unit with individual oil systems for each gearbox at extreme left. Oil reservoirs and variable speed motor drive unit (800 to 2500 rpm) are located below machine base

sults. Proper heat-treatments for cogwheels made from emergency steels such as NE steels and boron steels were developed with the aid of modified 4-square gear tests and transmission dynamometer tests before these steels were released for production. Changes in fillet radii, evaluated quantitatively on various gear-testing machines, have shown corresponding results in field tests and in service. Changes in involute geometry have even been developed on the basis of modified 4-square gear test data and, with suitable alterations based on mounting rigidity, have proved to be successful in service.

In general, it can be said that the correlation between the results obtained in any of these laboratory gear testing machines and service performance has been very good. Where correlation has not been obtained, the laboratory test results have been helpful in analyzing the field conditions which caused the lack of correlation and in solving the problem involved.

In laboratory testing, as well as in field testing, the results which are obtained can only be as good as the ingenuity, perspective, knowledge, and care put into the planning, execution, and analysis of the tests and test data. Extrapolation of results beyond the particular set of variables present in that test can be dangerous in the hands of the inexperienced.

Modified 4-Square Gear Testing Machine

The design and principles of modified 4-square gear testing machines are discussed in detail elsewhere.[1-9] However, in this paper only a brief description of the following items relating to these machines will be made: mounting rigidity; mass-elastic system; torque application, measurement, and control; shut-off devices; and lubrication.

Mounting Rigidity—The mounting of gears in modified 4-square gear-testing machines has varied from a simple, overhung design (that is, supported only by one bearing) [7] to a closely supported, rigid frame design, such as used in the recently constructed high-capacity machine of the Caterpillar Tractor Co., as shown in Figs. 1 and 2. The overhung design is generally not considered to be suitable for endurance testing, since serious shaft deflections occur under heavy loads and cause load concentration on one end of the teeth. However, the General Electric Co. has found it desirable and helpful to use this design in their railway gear endurance test machine, shown in Fig. 3, since it simulates the actual mounting used in railway power units.

Most machines use straddle-mounted gears with closely coupled bearings and rigid frames to reduce shaft and housing deflections to a minimum. We consider this design necessary where metallurgical studies are of paramount importance.

Bearings should preferably be press-fitted directly onto gearshafts rather than onto intermediate bush-

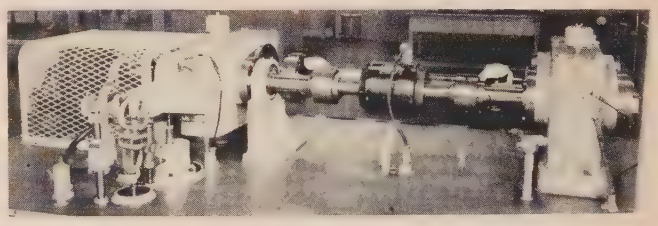


Fig. 5—Close-up view of Fig. 4, showing (a) loading gearbox mounted on trunnion bearings on front shaft, (b) strain gage torquemeter in front shaft, (c) universal joints in rear shaft to compensate for misalignment caused by applied load, and (d) test gearbox at right. Below loading frame is oil dashpot, rubber snubber, microswitch, and tension spring. Loading capstan nut and Emery load cell are not visible below machine base

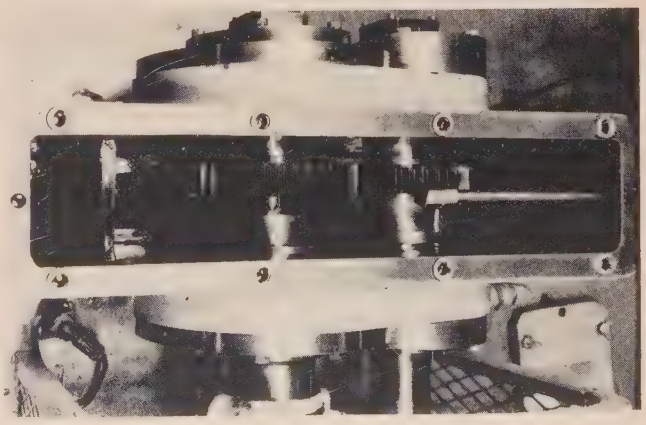


Fig. 6—Close-up view of test gearbox seen in Figs. 4 and 5. Bearing supports are replaceable for testing gears of 5.5- to 8-in. center distance. Gearshafts have labyrinth-type grooves for oil seals

4 *American Machinist*, Vol. 98, August, 1954, p. 169: "Somntag Special Machine Simulates Gear Fatigue."
5 SAE *Journal*, Vol. 62, May, 1954, pp. 60–63: "Engineering Evaluation of Gear Manufacturing Processes," by W. E. Gustin and D. C. Smiley.
6 ASME Transactions, Vol. 71, 1949, pp. 487–497: "Marine Propulsion-Gear Testing at Naval Boiler and Turbine Laboratory," by I. Monk.
7 ASME Transactions, Vol. 73, 1951, pp. 687–694: "Effect of Oil Viscosity on Power-Transmitting Capacity of Spur Gears," by V. N. Borsoff, I. B. Accinelli, and A. G. Cattaneo.
8 *ASTM Bulletin*, No. 148, October, 1947, pp. 69–73: "Gear and Lubricant Tester Measures Tooth Strength or Surface Effects," by E. A. Ryder.
9 *Machine Design*, Vol. 18, July, 1946, pp. 142–146: "Evaluating Surface Durability of Gears," by T. H. Wickenden, G. R. Brophy, and A. J. Miller.

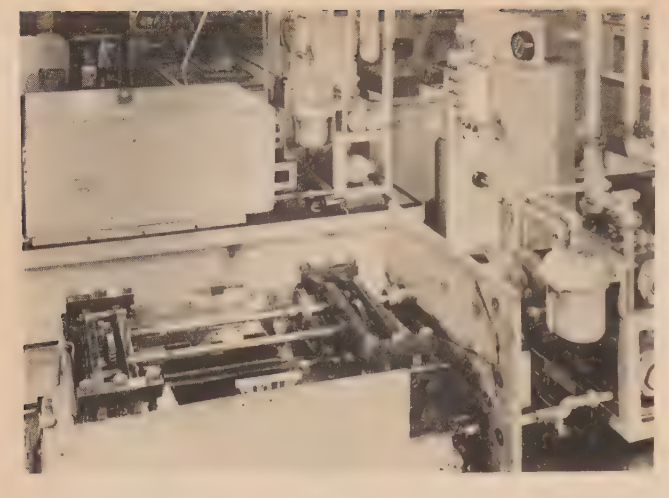


Fig. 7—Small-capacity modified 4-square gear-testing machines. Lubrication system and motor drive are located outside of sound-proofing housing for testing machine. Bearing support arms are removable for ease of assembly and replaceable for different center distance. Test gears are at left, power return gears at right, and loading coupling on far shaft. Control cabinet, including overload control device for stopping machine when tooth breaks, is behind lubrication system

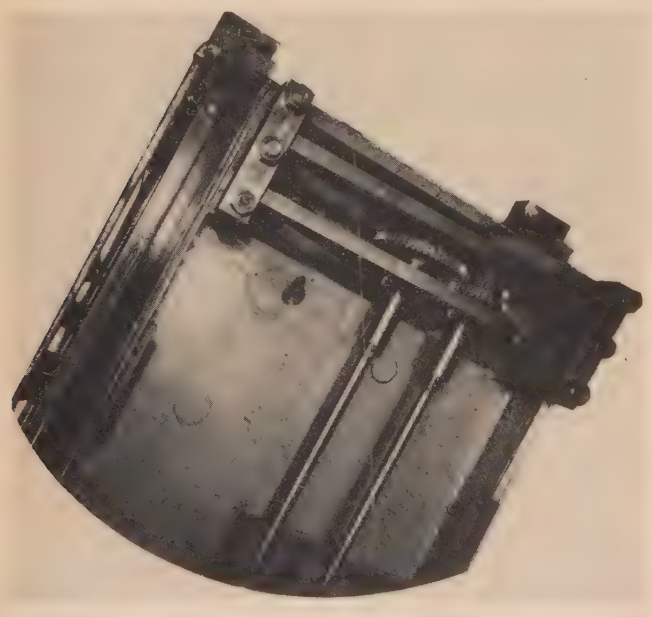


Fig. 8—Close-up view of test gear end of machine seen in Fig. 7, showing close coupling of bearings and flexible connecting shafts

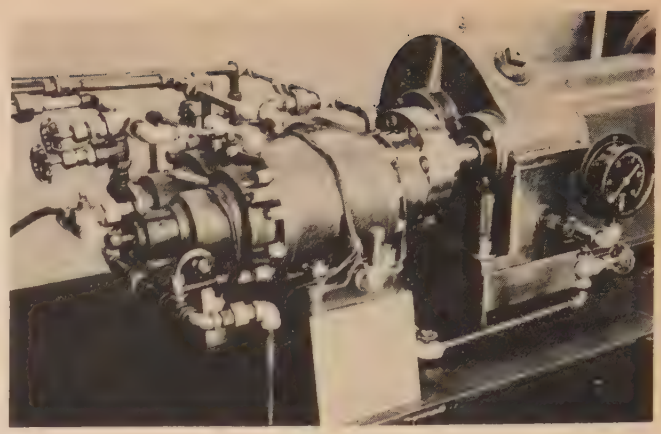


Fig. 9—General view of gear tester showing (a) driving motor and speed reducer at right, (b) common gearbox for test and loading gears and piping to separate oil supply system for test gears and loading gears. Corks in left flange cover inspection holes for test gear examination without removal of flange

ings which have slip fits on the shaft for ease of assembly. The latter design has proved to be quite troublesome in the large gear testing machine of the International Harvester Co., shown in Figs. 4, 5, and 6, because of fretting corrosion between the bushing and the shaft. Some relief has been obtained by cadmium plating the inside of the bushing and chromium plating the outside of the shaft.

Removable bearing supports, found in the International Harvester Co.'s small gear-testing machines shown in Figs. 7 and 8 have definite advantages in ease of assembly. The split mounting of the Caterpillar Tractor Co.'s large machine would seem to have distinct advantages also.

Loading gears and shafts of one-piece construction are used in some machines in order to increase the rigidity and accuracy of these parts and to eliminate the tendency of splined joints to wear due to fretting corrosion. Test gearshafts are considered to be expendable. They require replacement after becoming worn outside of specification tolerances.

Mass-Elastic System—Mass-elastic effects are as important in modified 4-square gear testing as they are in any type of gear testing. Selection of test speed must include consideration of accuracy of the gears and resonant frequency of the testing equipment as well as of expediting of the test. We have seen cases in which exceptionally high tooth loads have resulted from operation of standard 4-square testing machines and chassis dynamometer rigs near their natural resonant frequency, or a harmonic thereof. The load-measuring instrumentation, having a slow response, gave no indication of overloads.

Most modified 4-square gear-testing machines use fairly flexible coupling shafts between the test gearbox and the power return (or loading) gearbox. This flexibility helps to iron out fluctuations in tooth load caused by spacing errors, and to give low-resonant frequency to the machine. One exception is the Pratt & Whitney Aircraft gear-test rig, shown in Figs. 9 and 10, in which the test gears and loading gears are mounted within the same rigid case and on the same rigid shafts. This machine is used to test precision aircraft gears and to evaluate gear lubricants. It would seem that the extremely rigid mounting and coupling should be seriously questioned before considering this machine design for testing automotive gears having less precise manufacturing tolerance.

Application, Measurement, and Control of Test Load—The application, measurement, and control of

torsional load in modified 4-square gear-testing machines is accomplished in a number of satisfactory ways.

One simple method for manually applying load utilizes a piloted, flanged coupling in one of the parallel shafts connecting the gear sets. Figs. 11, 12, 13, and 6 show the types used at Deere Mfg. Co.,[5] General Electric Co., and International Harvester Co.,[1] respectively. Twisting of one flange relative to the other imparts the torsional load, which is locked into the system by securing the flanges together with one or more bolts.

In Deere Mfg. Co.'s design of coupling, slotted holes in one flange enable sensitive adjustment of the load. The International Harvester design has a series of holes near the flange periphery, with one flange having an additional hole. Fairly close adjustment of load is obtained by the vernier arrangement of the holes and by the use of flexible connecting shafts between the gearboxes.

The amount of torsional load in machines of this simple construction is measured either by applying a definite twisting moment to the coupling by means of a lever and spring balance or by including a mechanical indicating torquemeter, or a transducer type of torquemeter in the second shaft. Fig. 11 shows the type of mechanical calibrated torquemeter used at

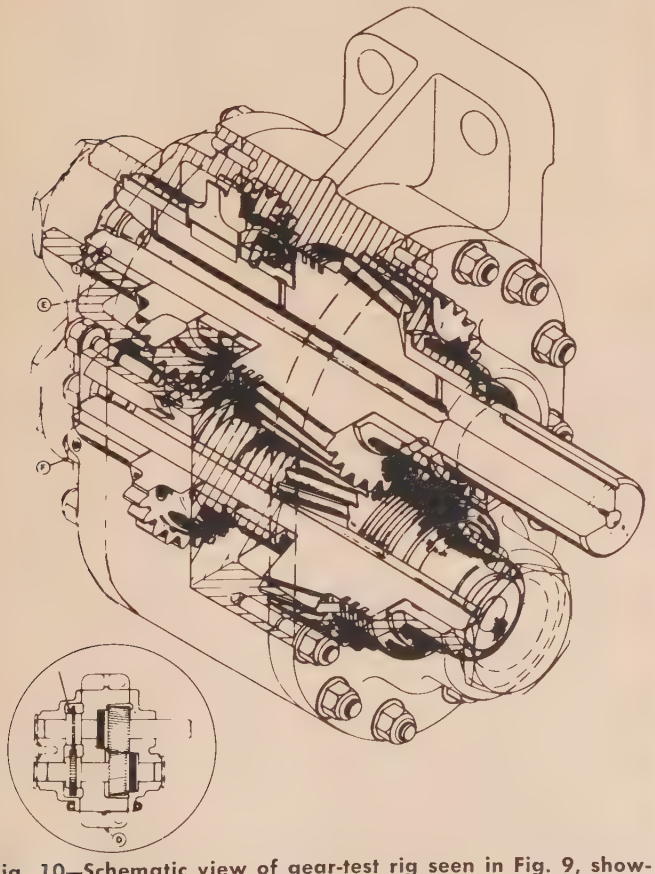


Fig. 10—Schematic view of gear-test rig seen in Fig. 9, showing rigid mounting of test gears on common shaft with helical loading gears

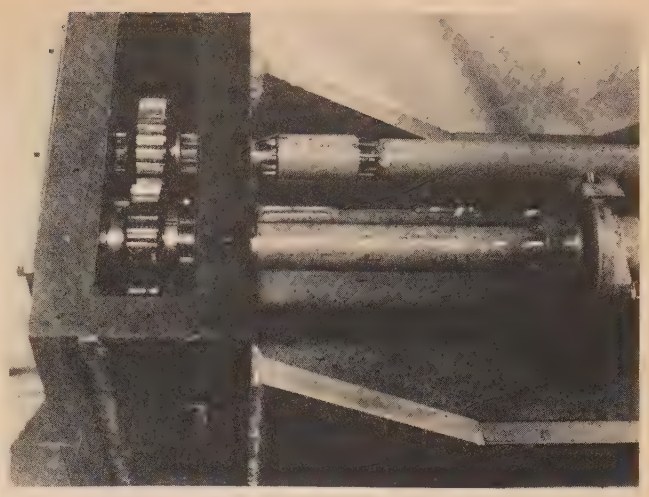


Fig. 11—Close-up view of loading coupling and test gearbox of power circulating gear testing machine of John Deere Waterloo Tractor Works. Rigid gearbox and machine frame have welded construction

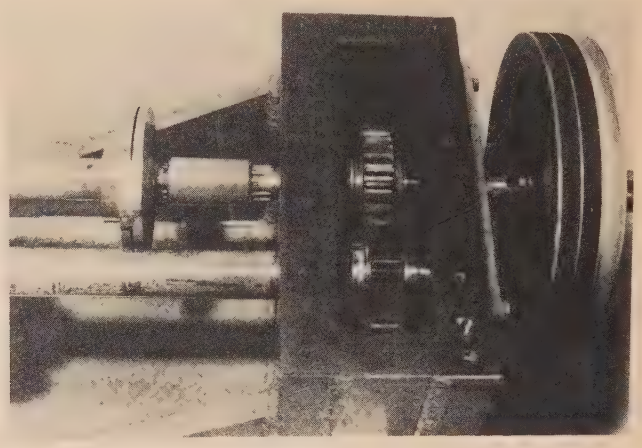


Fig. 12—Close-up view of mechanical torque-indicating coupling and power-return gearbox in modified 4-square gear testing machine of John Deere Waterloo Tractor Works, Deere Mfg. Co.

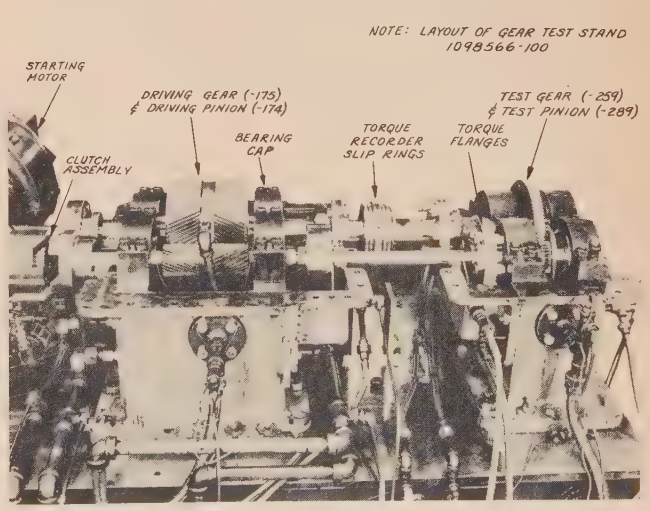


Fig. 13—Small aircraft gear endurance test machine of General Electric Co., showing (a) rigid mounting of test gears, (b) bearing caps for ease of assembly, (c) torque flanges for application of load, (d) torquemeter, and (e) large precision herringbone driving gears

Deere Mfg. Co. Fig. 13 shows the transducer type of torquemeter used at General Electric Co.

It is obvious that test loads in machines of this design must be applied or adjusted with the machine stopped. The inability to apply or adjust loads during operation is not necessarily a serious disadvantage, since the tests must be interrupted at periodic intervals for visual examination of the test gears in location. It has been our experience that the use of fairly flexible connecting shafts makes load-adjustment intervals necessary less frequently than visual inspection intervals. We have seen no evidence that starting the test under full load has any significance upon final results so long as the lubricating oil does not have borderline antiscuffing properties.

A second method for applying load to modified 4-square gear testing machines utilizes helical gears as loading gears. Figs. 14 and 15 show Caterpillar Tractor Co.'s machines of this type,[3] and Figs. 9 and 10 show the Pratt & Whitney Aircraft gear test rig.[8] In machines which are built with this loading principle, one of the helical loading gears is mounted on a shaft which is free to float axially. The other helical gear and the two test gears are mounted on axially fixed shafts. Axial thrust against the floating gear develops a twisting moment in the system in relation to the helix angle of the loading gears. A mechanical leverage system is employed in the Caterpillar Tractor Co.'s machines for supplying the axial thrust. Hydraulic pressure against the piston-like hubs on the floating helical gear is used in the Pratt & Whitney Aircraft machine.

The amount of load in machines of this type is controlled either by weights on the external lever arm or by hydraulic pressure against the hubs of the floating gear. Hence these machines are constant load machines. The test load can be applied or changed while the machines are running at test speed, thus eliminating any possibility of scuffing during starting of the machines. For lubricant testing, these machines would therefore have definite advantages.

Van Zandt and Kelley [3] reported good agreement between calculated torque (based on helix angle, mechanical leverage, and weights used) and experimentally measured torque (SR-4 strain gage torquemeter), except near the rated capacity of their machines where a correction curve was needed.

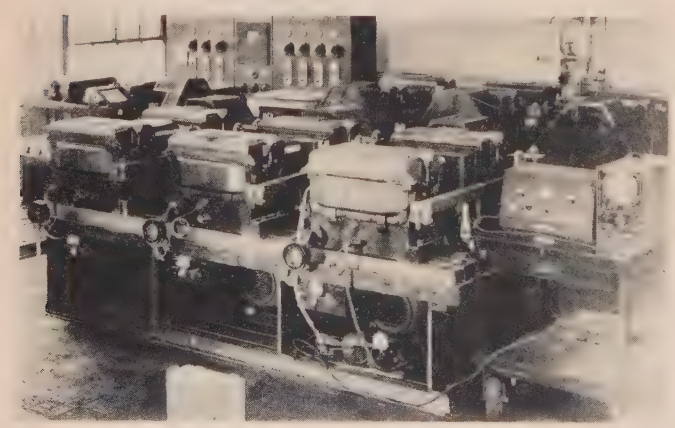


Fig. 14—Battery of single-pair gear test machines showing six machines mounted on one base and another on a second base. Lubrication systems and motor drives are located below base

Fig. 15—Close-up view of single-pair gear test machine seen in Fig. 14, showing (a) constant center distance test and loading gearboxes, (b) wide precision helical loading gears, and (c) external mechanical leverage for applying thrust load to helical loading gear mounted on axially floating shaft

Theoretical calibration curves for the Pratt & Whitney Aircraft machine are available for different machine speeds in order to correct for errors due to rotation of the oil in the loading chambers. How closely the theoretical calibration agrees with experimentally measured load has not been reported.

A third method for applying loads to modified 4-square gear testing machines is incorporated in the English 5-in. centers machine [2] and in the International Harvester Co.'s 30,000-lb-in. capacity machine,[4] which has a similar design. In this method the test gearbox is mounted rigidly to the machine base, and the loading gearbox is free to turn on trunnion bearings on one of the connecting shafts. The other shaft has a universal joint on either end to correct for the shaft misalignment which results when the loading gearbox pivots around the first shaft because of applied load. Twisting of the loading gearbox around this shaft produces an opposing bearing reaction within this gearbox and causes a locked-up torque within the system.

Attached to the loading gearbox is a frame for applying a twisting moment to this gearbox. The actual loading is done through a large capstan nut beneath the base.

The amount of load is indicated on a large spring balance in the 5-in. centers machine. In the International Harvester machine a stiff tension spring and the loading screw are connected in series with an Emery load cell. The amount of load is indicated on a large scale on a separate instrument panel. The connecting shafts must be rigid enough to keep the universal

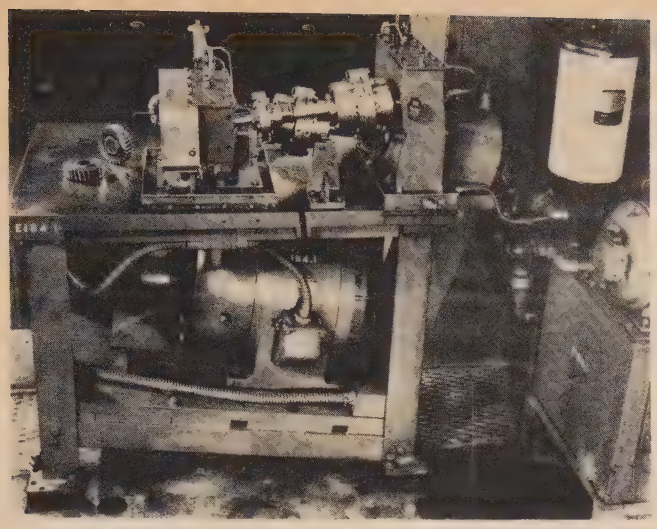


Fig. 16—Rear view of gear testing machine of International Nickel Co., showing (a) lubrication unit at right and (b) motor drive below machine base

joint eccentricity within permissible limits. The machine does not quite qualify as a constant load machine, since an occasional small adjustment in load is required as the test proceeds.

Having an external loading mechanism, these machines can be loaded after they have reached operating speed. A strain gage torquemeter in the trunnion bearing shaft on the International Harvester machine is available for calibration, for checking against internal resonance within the system and for studying variations in shaft torque due to such factors as pitch line run-out, spacing errors, and tooth interference.

A slight variation of this third method is used in the International Nickel Co.'s gear-testing equipment [9] which is shown in Figs. 16 and 17. In this machine the gearshafts in both the loading gearbox and the test gearbox are mounted on a 15-deg angle to each other. In order to operate the machine with this much shaft eccentricity, universal joints are used in both connecting shafts between two gearboxes. Loading is accomplished by hanging weights onto the framework of the loading gearbox, so that the machine is a constant load machine.

Since each pair of crossed helical gears operates on axially misaligned shafts within its own gearbox, the teeth mesh in an entirely different manner than they do in ordinary spur gears. The contact area is elongated and elliptical in shape, conducive to high compressive stresses and to sliding but not to low bending stresses. The machine is not suitable for comparing bending-fatigue strength of gears. It would seem to be satisfactory for comparative tests involving the performance of lubricants and of materials for pitting and scuffing properties. Its design is now undergoing modifications in order to increase its load capacity.

A fourth method for applying loads to modified 4-square gear testing machines is similar in principle to the second method, except that it uses mating helical splines, actuated by hydraulic pressure, in one connecting shaft in order to lock in the torque. The General Electric Co.'s railway gear testing rigs, shown in Fig. 3, use this method of loading. The helical loading splines are located inside the herringbone "power-return" pinion. The motor drives the gearshaft, and torque is measured by means of a strain gage torquemeter.

A variation of this fourth method is used in marine propulsion-gear testing apparatus,[6] which qualifies more nearly as an opposed-transmission rig. This rig is loaded through a hydraulic torque applier unit located in one shaft. The torque applier unit consists of mating helical splines and suitable hydraulic means for loading the spline which is free to float axially. Strain gage torquemeters are used to measure applied loads on the gear shafts.

The Caterpillar Tractor Co.'s high-capacity gear-testing machine uses a specially designed hydraulic torque coupler in the load shaft. The application of hydraulic pressure to cells within this coupler causes a twisting moment in the shafts. Wide precision herringbone gears, designed for trouble-free performance, are used as power-return gears. A strain gage torquemeter is used for measuring applied torque.

All of the gear testing machines which use the

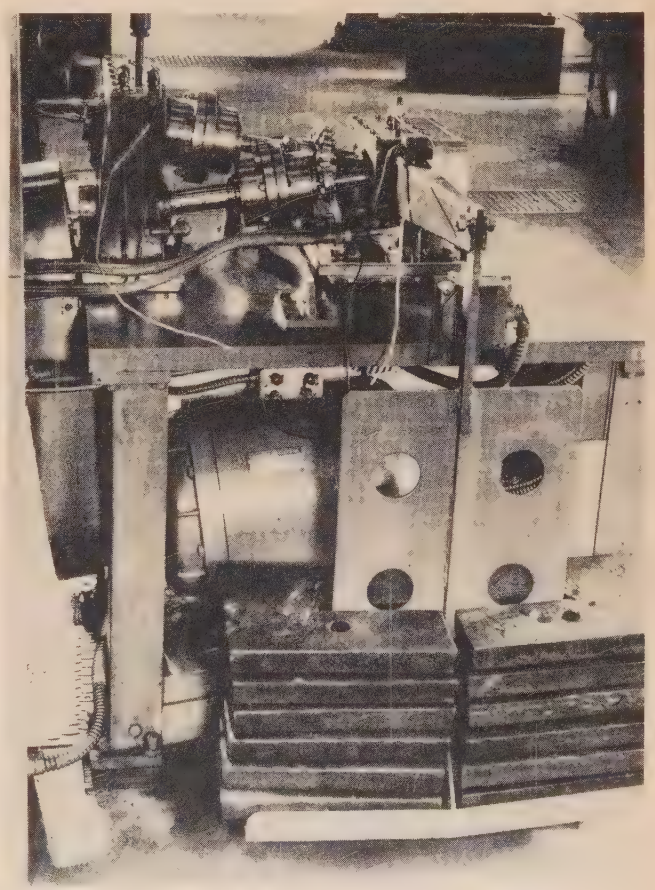


Fig. 17—Front view of gear testing machine seen in Fig. 16, showing (a) loading arm and weights on trunnion mounted gearbox, (b) axial misalignment of gearshafts, and (c) special Ajax dihedral joints in shafts connecting gearboxes

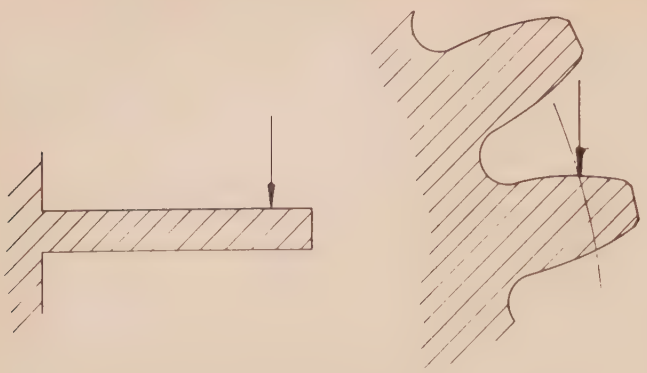


Fig. 18—Cantilever-beam principle of single-tooth testing

fourth loading method, or variations thereof, are constant load machines.

Shut-off Devices—Modified 4-square gear testing machines which are used for evaluation of lubricants and of pitting and scoring properties of gears have somewhat different requirements for automatic shut-off devices than machines which are used for evaluating bending-fatigue strength. The former tests generally produce progressive types of failures which must be visually examined at definite intervals. The only successful automatic shut-off and indicating device apparently has been the noise meter used in the International Nickel Co.'s machine. A noise level chart records the progression of failure due to pitting or scoring. Visual observation is used to detect incipient and progressive failures by most investigators.

A completely satisfactory device for stopping machines used for bending-fatigue tests has not been developed. The various devices which are being used have inadequate sensitivity for detecting the initiation of fatigue cracks. Gears with cracked teeth have been observed to run for considerable lengths of time before complete failure resulting from load transfer to adjacent teeth. We have even noted that some gears continue to run for many cycles with very little audible indication of failure even after one tooth has broken out. On the other hand, some gears fail by the rapid shelling out of adjacent teeth after the first tooth breaks.

Among the automatic shut-off devices which have been tried and found partially successful are the following:

1. Electric grid below the gears. A small mesh grid is designed to be shorted out and to actuate a relay when a broken tooth falls upon it.

2. Electronic overload control units. A commercially available electronic overload control unit, which has sufficient sensitivity and response to detect a slight momentary change in torque (and hence motor load) caused by a cracked or broken tooth, actuates a relay for shutting off the motor. At times, this control has detected fatigue cracks before a tooth has broken out, but usually a tooth breaks out before the control functions. The device requires closely controlled oil temperatures and constant loads for best results, since it is sensitive to changes in power requirements resulting from these factors. It automatically prevents operation of the machine near any resonant frequency harmonic.

3. Microswitches. Machines which have external loading arms can be stopped by microswitches actuated by the movement of these arms when failure occurs. These generally operate only after total failure of at least one tooth.

4. Thermocouple-actuated units. The navy marine propulsion-gear test rigs have automatic thermocouple-actuated devices for shutting off the power source when the outlet lubricating oil temperatures at bearings rise slightly due to incipient bearing failure. This method does not shut off the machine due to tooth breakage, however. It conceivably might be satisfactory for detecting scuffing or gross pitting failures in jet-lubricated machines, since such surface deterioration should raise the temperature of the oil being thrown off the test gears.

Lubrication—The lubrication systems in modified 4-square gear testing machines vary from simple splash types without temperature control to complete recirculating types with automatic temperature control and separate oil supply for each gear set. The latter types, which are predominant, have definite advantages for evaluating lubricants and surface deterioration due to scuffing and pitting. Full filtering of recirculated oil is common to many machines.

Gears can fail, or cease to be useful, by any of several different means:

1. Fracture due to bending fatigue.

2. Progressive surface destruction due to a) pitting, b) wear, and c) scuffing or localized welding.

Perhaps the most serious of these failures from the viewpoint of the machine operator is the first, the failure due to actual breakage of a tooth. The machine quickly becomes completely immobile due to failure of a constant mesh gear, or partially inoperative if a single-speed transmission gear fails. The customer generally takes a very dim view of bending failure even if the gear has operated in heavy-duty service for several thousand hours. However, if a gear is damaged by progressive surface deterioration, he may be unlikely to consider the damage as serious, since the machine is still operative. Bending failures must be avoided if the integrity of the product is to be maintained.

Proper design of the gear assembly can do much to prevent bending failures. Tooth width, pressure angle, diametral pitch, gear system, relative number of teeth, root fillets, involute tip relief, crowning, bearing supports, and shaft and housing deflections all play extremely important parts in determining the effectiveness of any given gear set. However, the scope of this paper excludes these mechanical factors, vital as they are.

The metallurgical aspects of gearing are equally important. Which steel is the best to use? What type of heat-treatment is most suitable? What case depth gives the best tooth strength? Is shotpeening economically justifiable? Questions of bending strength, hardenability, machinability, cost, availability, distortion, and others must be considered.

Means must be found to solve these problems. Obviously, it is impractical to run complete tests on each of the gears in a machine. However, suitably chosen tests on the most critical components, as determined either by stress calculation or field experience, can also give extremely helpful information on what can be expected from the less critical gears. In this way, a background of experience can be established.

Certainly, assembly testing can best reproduce the normal environment of a gear or any other part. However, tests of a complex assembly to determine the effect of metallurgical variations of one component can frequently be misleading. Many mating parts are involved, each of which has its effect upon other parts. In a modified 4-square gear test, rotating gears under load against mates simulate to a certain degree the normal environment of load-carrying gears operating in a machine. However, results of tests intended to detect differences in bending-fatigue strength are influenced by factors such as dimensional variations between the mating gears during rotation; wear on splines, shafts, and bearings; and deflections of shafts, bearings, and housings as well as occasional failure of other parts of the assembly. These factors will be present in the machine no matter what the metallurgy or treatment of the critical gear. In complex assembly tests it is often extremely difficult to detect variations caused by changing the metallurgy or treatment of the part being tested. Since gears can fail in many different ways, it is frequently desirable to isolate the type of failure under study: get down to fundamentals by using the most elementary test available to evaluate the desired variations in the single part and still duplicate the service failure.

What is a gear? Here it may be considered as a wheel from which project a number of carefully shaped cantilever beams called teeth. To determine the bending strength of a cantilever beam projecting from a part, the body of the part is held stationary, and a load is applied near the outer end of the beam. This, when done with a gear, is single-tooth testing as shown in Fig. 18.

What equipment is necessary in single-tooth fatigue testing? Obviously, a source of repeated load is essential. Excellent results have been obtained with universal fatigue testing machines of the rotating eccentric type. Magnetic oscillators and hydraulic loading have also been used successfully. In all types of single-tooth tests the fixture is simplicity itself, requiring only means for rigidly holding the gear while the load is applied to a particular location within the outer half of the tooth. For comparative results, the load must be applied in the same relative position on all teeth that have to be tested.

The fixtures used for single-tooth fatigue testing may be classified into two main categories. The teeth may be loaded in bending by:

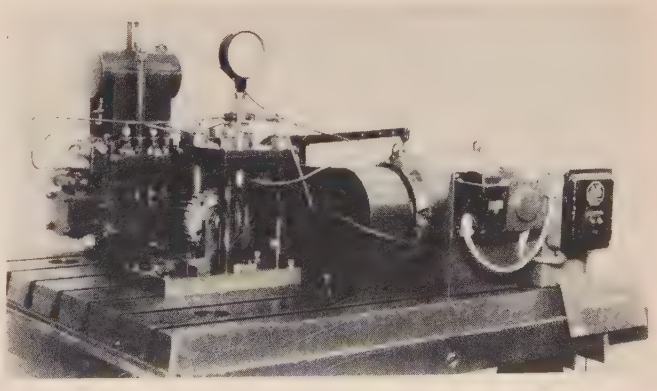
1. Direct loading on the teeth.
2. Torque applied through a shaft and resisted by the teeth.

Each of these types may be used with a variety of machines. The actual alternating load may be excited by:

1. Hydraulic pulsators.
2. Magnetic oscillators.
3. Rotating eccentrics.
4. Stroking machines.
5. Other devices.

Some examples of various combinations of fixtures and load generators will help to give a better understanding of single-tooth fatigue tests actually in operation.

Direct loading on teeth with use of hydraulic pulsators is being used by the Caterpillar Tractor Co. Their three machines operate on the same principle but are of different capacities for different size gears. An electric motor drives the pump to develop hydraulic pressure in the lines. A suitable mechanism actuates the piston and applies an alternating load varying from zero to maximum at a rate up to 1200 cpm. A general view of the smaller machine, with a capacity of 8000 lb, is shown in Fig. 19. The piston applying the load to the gear tooth is located below the pressure gage. Fig. 20 shows a close-up of the fixture itself. A short shaft supports the gear between two V-blocks. The load ram, designed to clear the adjacent tooth, applies the load to the desired position in the outer half of the test tooth. The reaction load is taken by the support block which contracts the support tooth at its base. A larger hydraulic machine used by Caterpillar, shown in Fig. 21, has a capacity of 100,000 lb. Used for coarse pitch gears, it has an identical loading system, but the load axis is arranged horizontally. The oscillograph is used for checking with SR-4 strain gages the dynamic loads applied on the loading ram. Fig. 22 shows the large 150,000-lb machine to test large final drive gears for tractors or

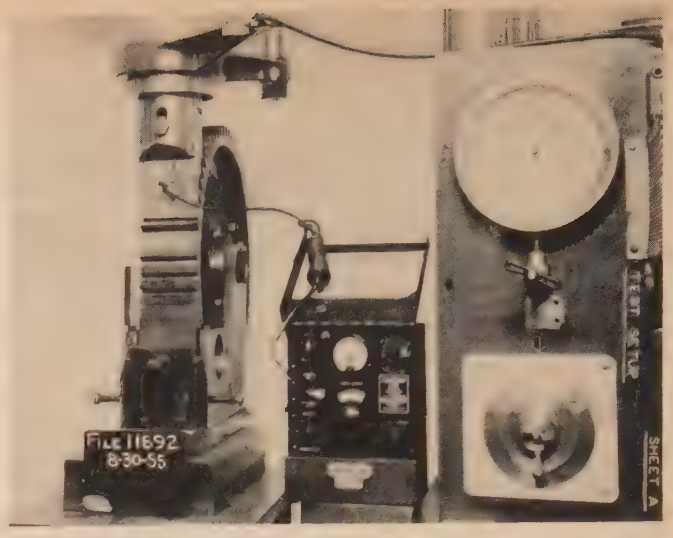


Courtesy, Caterpillar Tractor Co.

Fig. 19—General view of 8000-lb capacity hydraulic machine

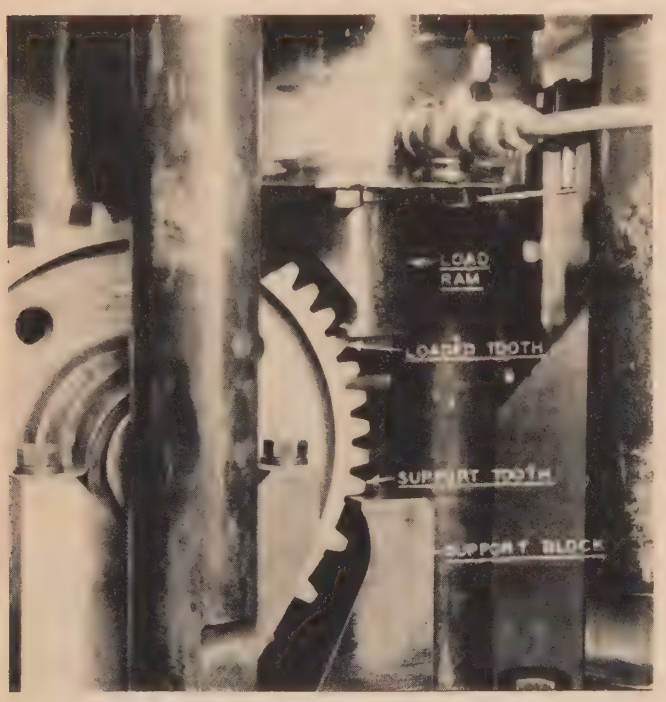
other heavy equipment. Hydraulic machines similar to these have been employed by Mansion [10] in England and by Glaubitz [11] in Germany.

Direct loading on teeth by a magnetic oscillator, as used by International Harvester, is shown in Fig. 23. The two large 12x12 "H" beams, boxed in, are each 8 ft long. They are joined at the top by a plate. When the assembly is hung from the ceiling, an inverted "U" is formed. The magnetic oscillator (not shown) is mounted near the bottom of one beam, vibrating it at a frequency much higher than most mechanical testers. Due to the familiar "tuning fork" principle, the other beam (or tine) also vibrates identically. When the test fixture is clamped between the two arms the vibratory load is resisted by the teeth, and they are loaded. The load is increased by



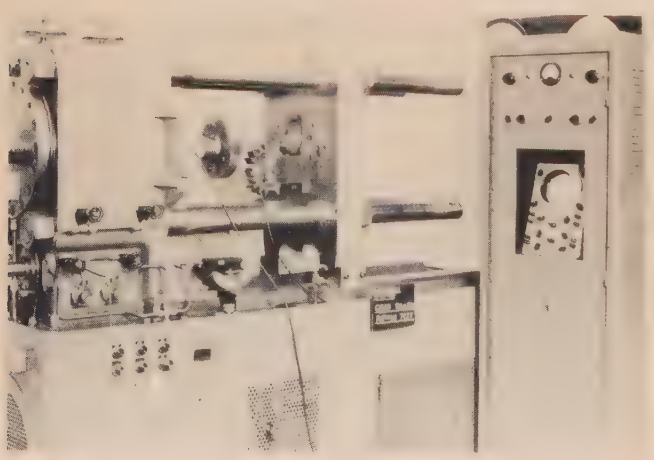
Courtesy, Caterpillar Tractor Co.

Fig. 22—Hydraulic machine of 150,000-lb capacity for testing large spur gears



Courtesy, Caterpillar Tractor Co

Fig. 20—Close-up of Fig. 19 showing gear test fixture in 8000-lb hydraulic machine



Courtesy, Caterpillar Tractor Co.

Fig. 21—Hydraulic machine of 100,000-lb capacity

sliding the fixture down the beams, and vice versa. One type of fixture is shown here although others are also possible in this versatile type of tester. The oscillator, or "shaker," used with this fixture permits 10,000,000 cycles to be attained in only about 44 hr. This is almost 38,000 cpm. SR-4 strain gages may be attached to the fixture or specimen to measure the dynamic stress in an accessible, easily duplicated position. The true stress at the point of failure is not measured but is proportional to the stress measured by the strain gage. The machine stops automatically when failure occurs. This type of setup is very versatile and adaptable for many types of fatigue testing.

A third device for direct loading on the teeth, as used by International Harvester, is shown schematically in Fig. 24. This kind of fixture has been used on rotating eccentric machines, and is illustrated later. With relatively small spur gears it is possible to machine an internal gear from a plate, the thickness of which is about the same as the width of the teeth. It has been found practical to fasten the loading block or ram with two arms to a shaft through the bore of the gear in order to maintain constant load location. The loading block contains a projection which actually contacts the test tooth. This is shown in Fig. 24.

Torsion loads applied to the shaft containing either the test pinion or the loading member are also used for single-tooth fatigue testing. In either case, the test pinion or the loading member must be held stationary while the other part is rotated against it. Because of the oscillating shaft, antifriction bearings must be used.

Torque applied to the shaft containing the test pinion is the principle of the fixture shown in Fig. 25. The alternating load moves the end of a lever arm

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[10] "Hydraulic Fatigue Testing Machine for Gear Teeth, No. 1949/4," by H. D. Mansion. Pub. by Motor Industry Research Association, Brentford, Middlesex, 1949.

[11] Instit. für Maschinenelemente, Special Report No. 124, 1950: "Influence of Tooth Shape on Root Strength of Straight Spur Gears Made from Mild, Heat-Treated, and Case-Hardened Steels," by H. Glaubitz.

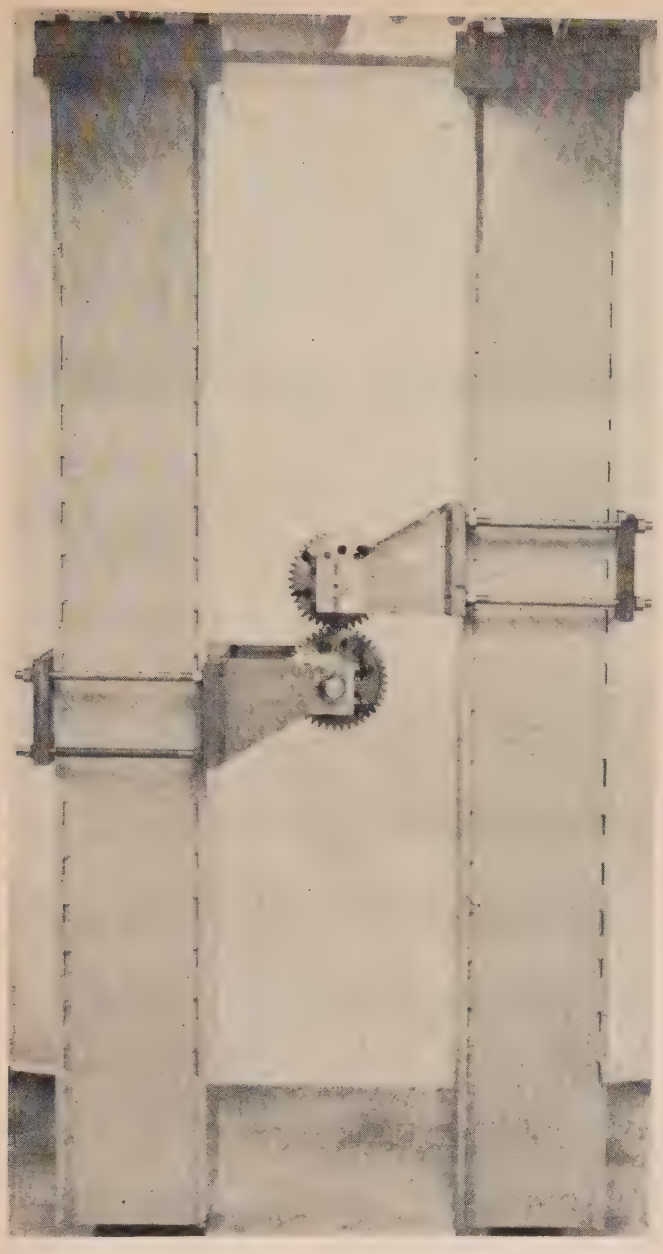


Fig. 23—"Tuning fork" fixture used with magnetic oscillator. Magnetic oscillator is not shown

fixed to a shaft. The test pinion is fastened to the fixture shaft by means of collets. The load is applied to the test pinion by a tooth of a mating gear held solidly by its shaft to the machine. Extensive testing with this type of fixture has been done by the General Electric Co.

Torque applied to the shaft of the loading member is the principle of the fixture shown in Fig. 26. International Harvester uses this fixture on a mechanically operated stroking type machine. The large final drive gear is clamped and braced in a vertical position to the base of the machine. The test tooth is positioned at the bottom. Through the link shown, a recriprocating slide at the top oscillates the vertical lever arm fastened to a large shaft running parallel to the base of the machine under the test tooth. Mounted in bearings, the shaft contains a ½-in. diameter pin, braced from behind, which contacts the test tooth along the pitch line. A close-up view of this arrangement is shown in Fig. 27. As can be seen, it was necessary to machine off part of the adjacent tooth to provide room for reinforcement of the loading pin. Measurements with SR-4 strain gages showed that this modification of the gear did not change the residual stress at the root fillet between the teeth. Some results from this test are shown later in this paper.

When a test is run on a gear, regardless of the type of fixture used, one tooth is loaded with a certain cyclic load. If no failure occurs, the test is stopped after a certain number of cycles, perhaps

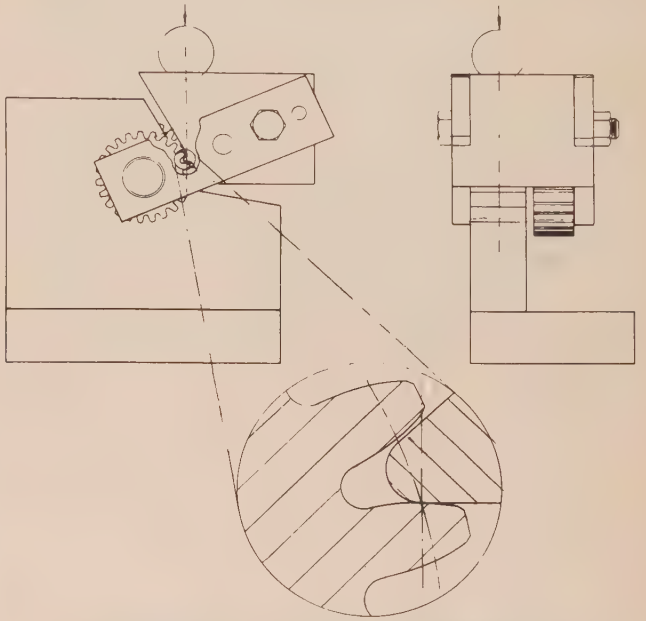
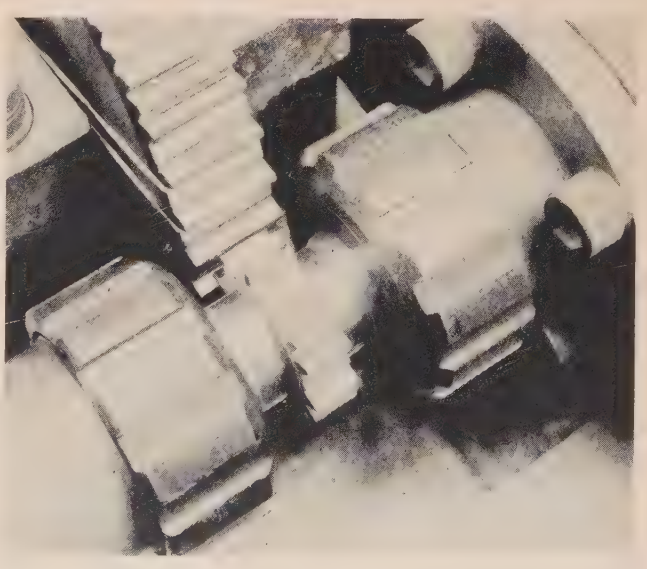


Fig. 24—Schematic view of direct-loading type of fixture



Courtesy, General Electric Co

Fig. 25—Torsional loading of small pinion (front) reacts against stationary gear

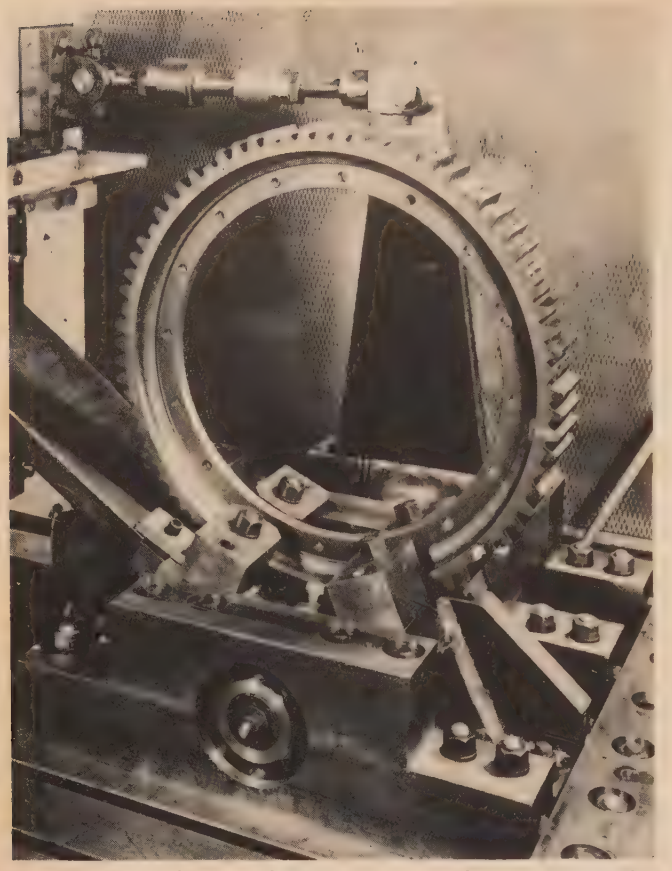


Fig. 26—General view of final drive gear loaded by oscillating shaft

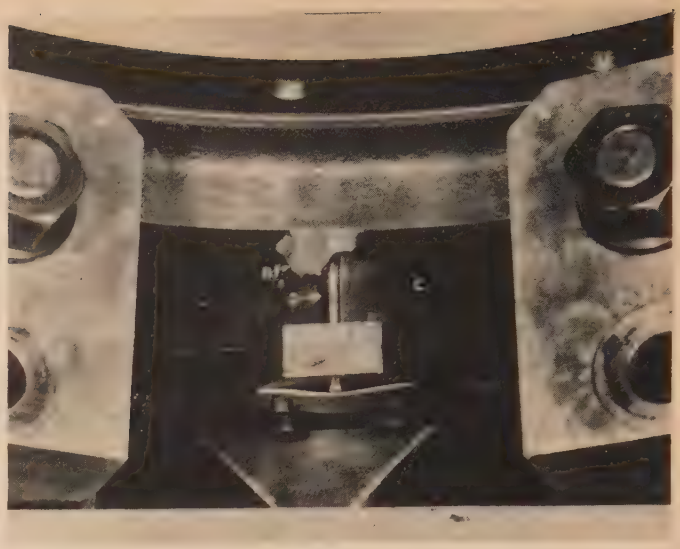


Fig. 27—Close-up of Fig. 26, showing details of test tooth and loading pin

5,000,000 or 10,000,000. After the test the gear is removed from the fixture, rotated two or more teeth, and replaced in the fixture. It is thus indexed so that another tooth is in position for test. This tooth is then loaded either above or below the load of the previous test, depending upon the earlier results. In this manner many teeth on the same gear may be tested. If desired, a good idea of the endurance limit may be obtained.

A gear containing a single metallurgical change may then be tested similarly. Whether the variable be in material or processing, the results may be compared directly with those of other gears tested, providing no other variables are present. For comparative purposes it is not necessary to know the theoretical bending stresses in the root fillet, although this information is useful for a better understanding of the results.

Interpretation of results of any comparative test can be extremely difficult if evaluation of more than one variable at a time is attempted. Suppose, for example, the object is to evaluate the effect of shotpeening. A group of identical gears must be made—same bar of steel, machined consecutively, heat-treated together—as similar as is commercially possible. Then some of these gears must be shotpeened by the desired procedure. After testing, the results of each group may be lumped together to evaluate the performance of the group as a whole, or the gears may be considered individually since separate endurance limits may be obtained. In this way the change in bending-fatigue strength due solely to shotpeening may be accurately evaluated.

The advantages and limitations of the single-tooth fatigue test as compared with running types of gear tests may now be considered. Some desirable features are that this test:

1. Eliminates test machine variables. Wear or failure of shafts, splines, bearings, and other parts of rotating-gear tests frequently confuse the test results. In the single-tooth test the load may be received directly by the test tooth. The fixure can be rigidly designed to prevent extraneous failure or damage.

2. Eliminates gear variables. Dimensional variations between pairs of rotating gears may tend to overshadow metallurgical variations. The effect of pitch line run-out, tooth spacing, and involute profile errors are eliminated since contact is made in the same position at all times.

3. Uses simple equipment. Standard fatigue testing machines of several different types may be used although special machines are sometimes employed. Since these machines usually have automatic cutoffs, the tests may be run continuously with the machine stopping upon tooth failure. For spur gears the fixtures are quite uncomplicated and are easily dismantled for changing the test gear.

4. Permits measurement of bending stress. If the gear teeth are large enough, it is relatively easy to measure bending stresses in the root fillet when the test is operating. SR-4 strain gages can be located wherever desired, and the operating stresses measured with suitable equipment. These stresses can be calibrated with the test loads as determined by the machine settings so that the actual bending stress is known for each machine setting.

5. Eliminates undesired failures. Progressive sur-

face damage due to pitting, wear, or scuffing is eliminated since no relative motion occurs against the test tooth. Only bending-fatigue failures occur.

No test is perfect. Each has its limitations. The single-tooth test is no exception with certain disadvantages that must be recognized, in that the test:

1. Limits failures to bending. If interest is not in bending failures at the root, the test is not suitable since the failure will always occur at the root fillet. Pitting fatigue may be simulated by rolling-type tests and wear or scuffing by sliding tests.

2. Lacks direct correlation with operation. In rotating service gear teeth are not loaded individually. Usually at least 1½ teeth are in contact at an average time. The tangential load carried by the tooth may be obtained from the test, then calculations must be made in terms of the contact ratio to determine the torque this would represent in service.

3. Requires special fixtures. The test may require special fixturing for each design tested. Complete fixture redesign may be necessary for relatively minor gear design changes. Gears with curved teeth such as helical, spiral bevel, and hypoid gears present special problems in fixture design, and the fixtures may become quite complex.

As can be seen, the test has some limitations. However, there are many times when it can be used as a supplement to field tests to improve the final product. It can act as a screening test to compare quickly various materials, heat-treatments, or manufacturing processes when bending-fatigue failures occur in the field. It can also serve as a research tool for investigating the factors affecting the bending-fatigue strength of gear teeth.

As an example of its use in a practical situation, field tests of preproduction tractor models containing compond 14 pitch planetary gears showed that the gear was subject to bending-fatigue failures. The field tests were time-consuming and expensive to run. The many possible variations in design, material, heat-treatment, and surface treatment made a screening test necessary. This gear is shown at the left side of Fig. 28.

On the basis of the failed field test gears, the design engineers made all the geometrical improvements practical. These included proper crown amount and location and also full fillet radii since it was not possible to change the size or basic tooth design. The balance of the improvement had to be achieved metallurgically by determining the optimum material, heat-treatment, and surface treatment.

The test fixture was simple in design, loading directly on the test tooth. The gear was held by a plate containing part of an internal gear to fit the smaller 23-tooth gear, as shown in Fig. 29. A shaft through the bore held two arms secured to the ram block. (The near arm was removed for the photograph.) The 23 teeth made it possible to test as many as 10 teeth on each gear to obtain a good idea of the endurance limit under these conditions.

One of the objectives of this test was to evaluate the effect of shotpeening. Several gears were forged from the same bar of 4118 steel, machined, carburized for 0.025–0.035-in. case, marquenched in 400 F oil for 10 min, aircooled, and tempered. All were as identical as commercially possible. Then some of these gears were shotpeened under controlled conditions while the balance was not shotpeened. Four gears of each type were tested. Comparative results of endurance limits, in terms of machine settings, for the eight separate gears are given in Table 1. Metallographic examination and microhardness tra-

Table 1—Comparison of Endurance Limits of Gears

Nonshotpeened, lb	Shotpeened, lb
2400	3300
2400	3100
2300	2900
2300	2800
Average 2350	3025

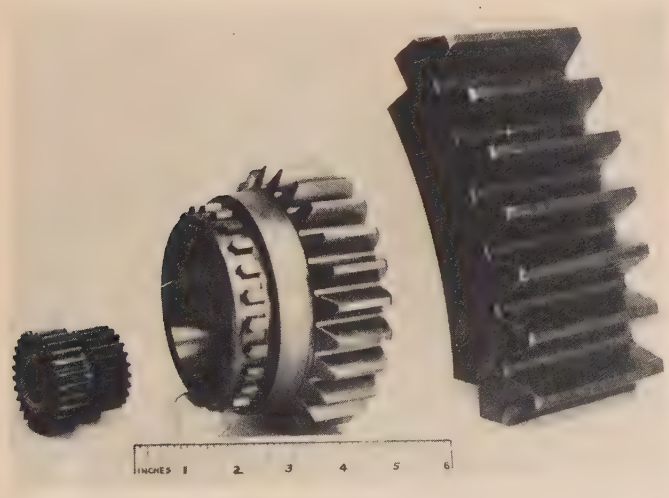


Fig. 28—Size comparison of gears for which test results are reported here. From left to right: small planetary gear, medium-size transmission gear, and section of large final drive gear

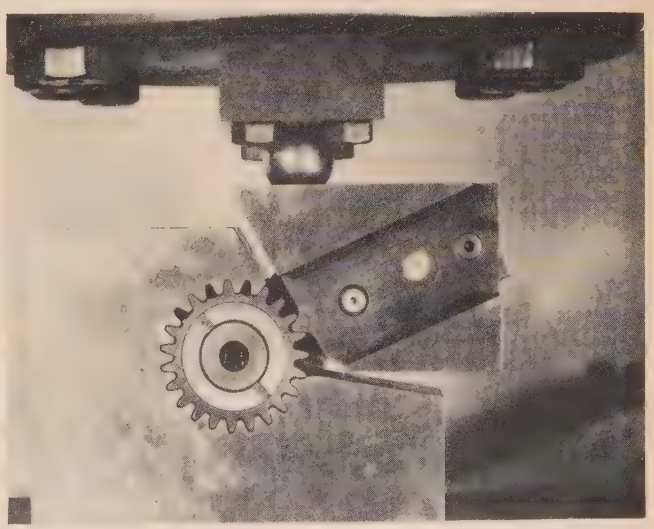


Fig. 29—Planet gear-test fixture shown schematically in Fig. 24. Arm on near side was removed for photograph

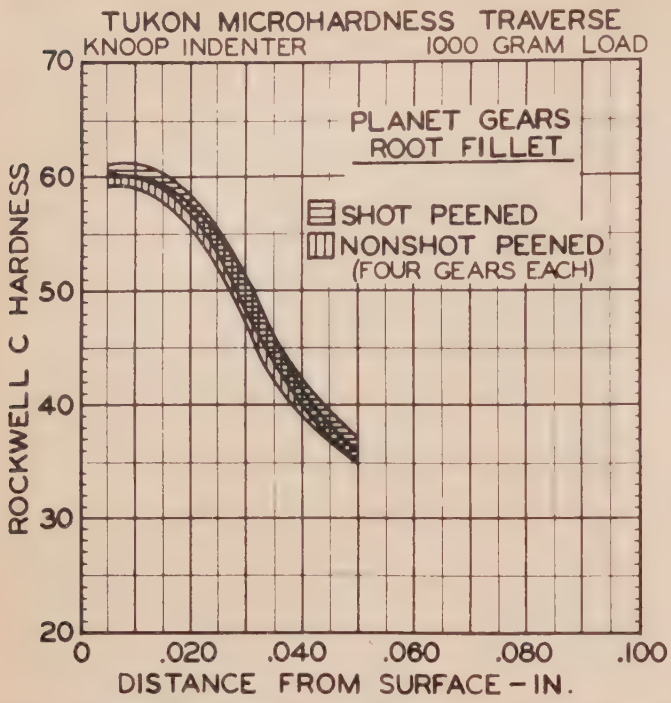


Fig. 30—Planet gear test, root-fillet hardness curves of shotpeened and notshotpeened gears for single-tooth fatigue

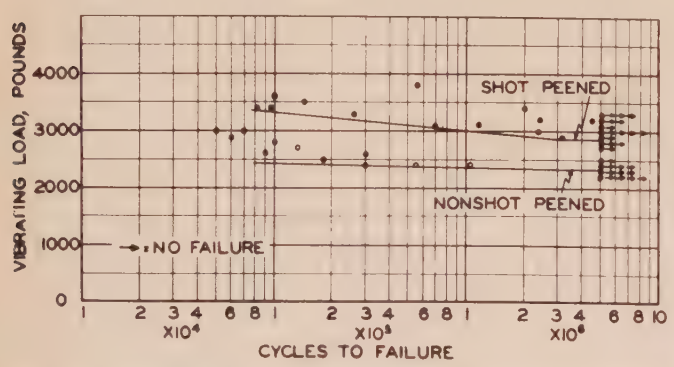


Fig. 31—Planet gear test, cumulative SN curves comparing shotpeened and nonshotpeened gears

verses at the root fillets of each gear showed that they were very nearly identical metallurgically. The root hardness traverse bands for each type were quite narrow and overlapped as shown in Fig. 30. Chemistry and microstructures were identical.

Since all possible mechanical and metallurgical factors were held constant, the one variable introduced, shotpeening, must have been responsible for the increase in bending-fatigue strength. That increase was in the order of $\frac{3025 - 2350}{2350} \times 100 = 29\%$ based upon the averages previously mentioned. Gears which had endurance limits at the 2300–2400-lb level failed on the test track, while otherwise similar gears, shotpeened as done here, ran successfully with no trouble. Shotpeening was the difference between success and failure in this gear. It was economically justified and has remained in use. The test results obtained were clear-cut and decisive. When the points in all these eight gears are plotted together, they form the two bands shown in the SN curves of Fig. 31.

Another objective of these tests was to evaluate steels. In order to compare with the 4118 steel gears, some planet gears made from 4817 steel were also tested. These were of the same design and heat-treatment as the 4118 gears mentioned previously. Two nonshotpeened gears had endurance limits of 3500 and 3600 lb, while a similar shotpeened gear attained an endurance limit of 4900 lb, the highest reached during all tests on the planet gears. The reasons for these results were not determined. While spectacular, these limited tests were not carried further because they involved the use of critical alloying elements. It was essential to obtain satisfactory results with a more practical steel.

Several other steels were tested and discarded because of poor performance. One of these was 50B44, cyanided and marquenched. The endurance limit was less than 2000 lb. The definite figure was never obtained because it was considered unwise to waste time on an unlikely course.

Heat-treat variations of these planet gears were also performed. Changes in the temperature from which three 4118 gears were quenched into 450 F salt gave the results shown in Table 2.

These gears were not from the same lot of steel as

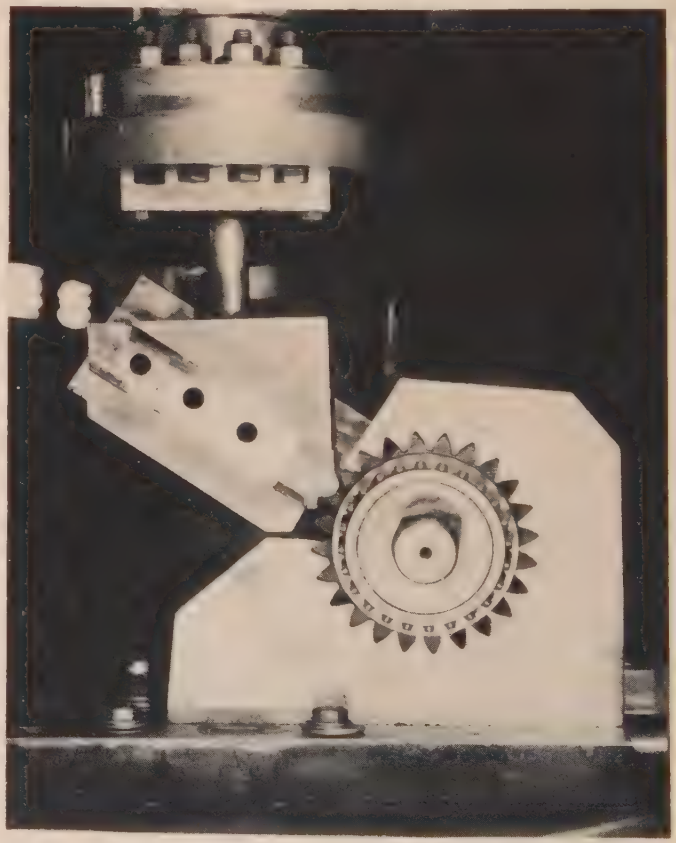


Fig. 32—Transmission gear test fixture similar in principle to that shown in Fig. 24. Arm on near side was removed for photograph

the gears previously mentioned, so direct comparisons should not be made. They were not shotpeened. The trend of increasing strength with decreasing quenching temperature is of interest here. Microhardness traverse curves in the root fillets were again almost identical for the three gears. This indicates that the differences in performance apparently were caused not by inherent strength, as measured by hardness, but by the changing residual stresses caused by the various quenching temperatures.

These results of steel and heat-treatment variations may present a somewhat distorted picture of the factors involved. These changes produced differences in the residual stress patterns which at the present time we are unable to evaluate. These results are presented here only to show the differences in performance that may be detected with the single-tooth bending-fatigue test. They do not necessarily represent optimum conditions.

Other bending-fatigue tests were run in a similar manner on a larger gear. This was a transmission gear used in a large crawler tractor. Shown in the center of Fig. 28, it was chosen as representative of gears in this transmission. Experience was desired concerning the single-tooth bending-fatigue strength of this gear.

The gear itself had 24 teeth with a diametral pitch approximately 4.8. The pressure angle was slightly over 25 deg, and the tooth width was approximately 1.4 in.

Fixturing, as shown in Fig. 32, was similar to that of the previously mentioned planet gear set except

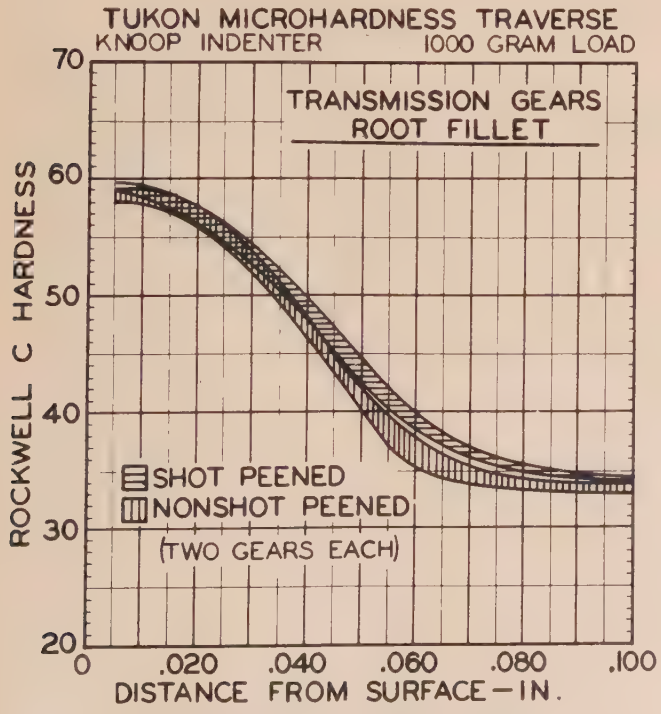


Fig. 33—Transmission gear test, root-fillet hardness curves of shotpeened and nonshotpeened gears

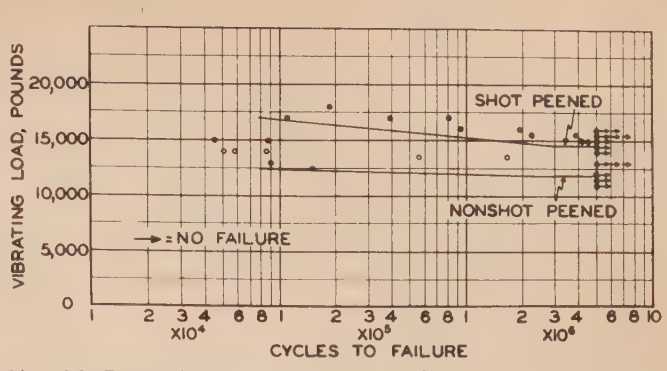


Fig. 34—Transmission gear test, cumulative SN curves comparing shotpeened and nonshotpeened gears for single-tooth fatigue

Table 2—Heat-Treat Variations of Planet Gears

Quenched from Temperature, F	Endurance Limit, lb
1600	2100
1550	2500
1500	2700

Table 3—Effect of Shotpeening on Gears

Nonshotpeened, lb	Shotpeened, lb
13,000	15,500
12,000	15,000
Average 12,500	15,250

that the line of contact was chosen midway between the pitch line and the tip of the tooth. The purpose was to weaken the test tooth by applying the load nearer the tip and to permit corresponding strengthening of the ram tooth.

Four gears were forged from the same bar of TS-8620 steel and were machined and carburized together in order to make them as identical as commercially possible. The purpose was to reduce the variables in the gears in order to evaluate more accurately the effect of shotpeening. The gears were all shot-cleaned, then two were picked at random and shotpeened. Table 3 shows the results obtained on the four gears tested.

Superimposing the shotpeening action upon the lighter shot-cleaned surfaces resulted in an approximate endurance limit increase of $\frac{15{,}250 - 12{,}500}{12{,}500} \times 100 = 22\%$. Again, examination revealed that the gears were very nearly identical within the limits of accuracy of the metallurgical tests. The microhardness traverse bands in the root fillets are shown in Fig. 33. The cumulative results of these tests are shown in Fig. 34.

Another single-tooth test currently being operated involves a very large sprocket drive gear from a crawler tractor. A section is shown at the right side of Fig. 28. Teeth could not be broken out in a massive 4-square type of machine, although failure sometimes occurred under severe field service. In order to evaluate the effect of proposed design and metallurgical

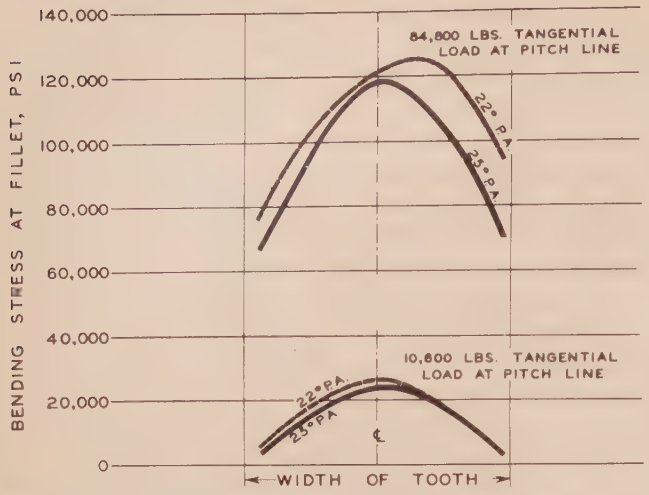


Fig. 35—Final drive gear stress traverse across root fillet of loaded tooth. Eight SR-4 strain gages side by side across the tooth were used to show the effect of 0.003-in. crown on the loading pin

Table 4—Effect of Shotpeening on Gear Life

Gear	Cycles to Failure	
	Nonshotpeened	Shotpeened
A	73,000	715,000
	66,000	385,000
	51,000	233,000
	50,000	
	49,000	
	40,000	
B	88,000	241,000
	80,000	201,000
	77,000	170,000
	76,000	143,000
C	57,000	116,000
	47,000	95,000
	35,000	89,000
D	88,000	192,000
	77,000	155,000
	53,000	106,000

changes, it was necessary to devise a test that could duplicate field failures in bending fatigue.

This gear has an OD of slightly over 31 in. with a face width of over 4 in. The 77 teeth are 2.5 diametral pitch with a 22-deg pressure angle. It has a short addendum causing the pitch line to be near the tip of the teeth. The gear is a large ring which normally is riveted to a gear carrier, or hub, for tractor installation. This gear is roll-forged from 1045 steel, normalized, machined, and induction-hardened. The test fixture, previously described, is shown in Fig. 26.

It is usually desirable to obtain the endurance limit as a measure of relative load-carrying capacity. However, other considerations may make this impractical. Quantitative measures of strength cannot be obtained with tests of life at a single load. Only trends can be established. It was not practical to obtain endurance limits on these gears since they operate quite slowly in service, and each tooth needs to withstand relatively few cycles at maximum load. Also, because of the slow speed of the testing machine, the tests necessary would be too time-consuming. Consequently the tests were run at an arbitrarily selected high load.

Shotpeening was investigated on these large gears also. In many cases both shotpeened and nonshotpeened teeth on the same gear were tested. This was done by masking certain teeth and peening the balance. In this way the variables were cut to a minimum. Some representative comparisons in life at a tangential load of 84,800 lb are shown in Table 4.

Since this gear is of ample size, it is relatively easy to measure root-fillet bending stresses. One interesting experiment was to measure the effect of the crown of the loading pin on the root-fillet bending stress. Eight SR-4 strain gages were placed side by side across the 4-in. width of the tooth. When a certain tangential load was imposed on the pitch line by the crowned pin the strain gage readings showed how the stress peaked near the center of the tooth and dropped off toward the ends. This is shown in Fig. 35. Curves for both 22- and 25-deg pressure angles are shown since some work is being done with the latter type of tooth.

The possibilities of single-tooth testing are limited only by the ingenuity of the engineers concerned. While this paper has been concerned primarily with bending-fatigue strength, the same type of fixture can also be used to determine static or impact strength, tooth deflection under various tangential loads, and, if the tooth is large enough, actual fillet stresses. If the problem involves bending failures of gear teeth, single-tooth fatigue testing can be a valuable supplement to field testing and service experience and can also serve as a useful research tool.

Summary

Single-tooth fatigue testing is used to measure the bending strength of gear teeth. One tooth at a time is tested in the manner of a cantilever beam. By successive testing, the strength of many teeth of the same gear may be tested, and endurance limits obtained if desired. The advantages of this type of test lie primarily in the simple test fixtures and clear test results.

Two types of test fixtures are possible: the gear tooth may be loaded directly, or the gear tooth may resist a torsional load applied to a shaft. Either of these types may be used with a variety of load-exciting machines.

Results of tests have shown that it is possible to determine clearly the effect of various metallurgical factors, such as material, heat-treatment, or surface-finishing treatment upon the bending-fatigue strength of gear teeth.

Acknowledgment

Mr. Halgren thanks the Caterpillar Tractor Co., Deere Mfg. Co., General Electric Co., International Nickel Co., Pratt & Whitney Aircraft Division, Allis-

Chalmers Manufacturing Co., and Chrysler Corp. for supplying information and/or photographs.

Mr. Wulpi wishes to thank many of his colleagues within the International Harvester Co. for their invaluable advice and help. These include J. D. Graham, F. Sailer, and men from their organizations, and many others. Also thanks are due to A. Cowie of Illinois Institute of Technology and D. W. Dudley of General Electric Co. for reading and criticizing the manuscript. General Electric Co. and Caterpillar Tractor Co. generously contributed photographs for inclusion here.

APPENDIX

The chemical analyses of the gears on which shot-peening tests were run are as follows:

	Planet Gear	Transmission Gear	Final Drive Gear (Range)
Carbon	0.23	0.22	0.43–0.50
Manganese	0.83	0.90	0.80–1.00
Silicon (Range)	0.20–0.35	0.20–0.35	0.15–0.30
Sulfur	0.013	0.027	0.05 maximum
Phosphorus	<0.04	<0.04	0.04 maximum
Nickel	0.11	0.46	
Chromium	0.55	0.72	
Molybdenum	0.10	0.10	
Steel Type	4118	TS-8620	C-1046 (modified)

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DISCUSSION

Compares Lab, Field Test Results

—B. W. Kelley
Caterpillar Tractor Co.

THE fact that the test machine is no better than the man behind it, is something that cannot be too strongly emphasized. I give overdue credit to the many operators and gear engineers who, with unobstructed mental outlook, conscientious analysis, and almost fanatic curiosity, have made these costly machines pay off.

We first consider our laboratory test machines as assistants to engineers who are working on failure criteria. Tooth breakage, pitting, scoring, evaluation of misalignment, dynamic loads, and lubrication represent only a few of the programs which have been run on such test machines. The machines, in effect, help the research worker to fill in or bridge the gaps still existing in our engineering knowledge today. We are convinced of the sound-

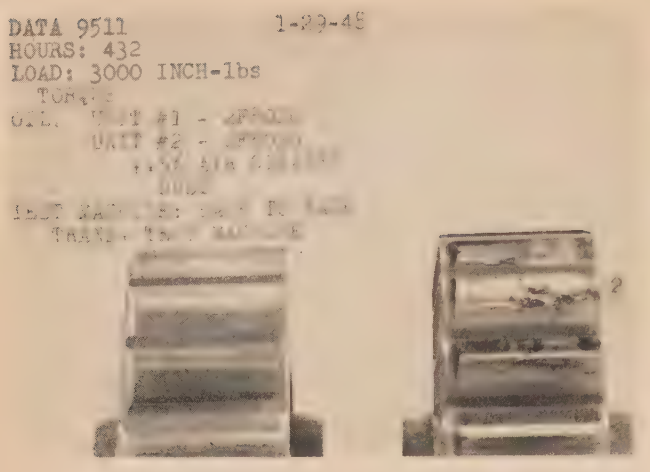

Fig. A—Field failure duplicated in laboratory testing by addition of abrasives in test boxes

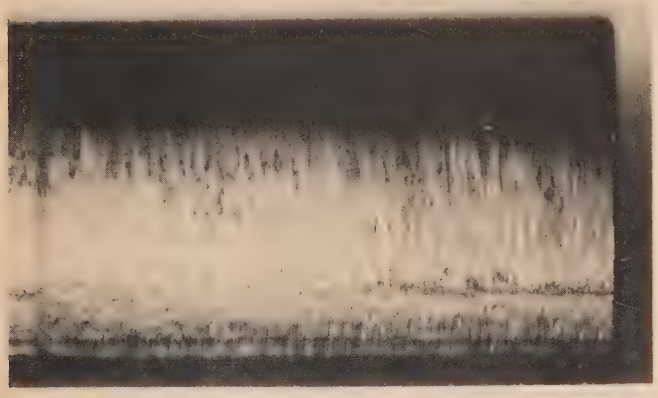

Fig. B—Wear resulting from low constant speeds reproduced in laboratory testing

ness of reason for the expenditures of such work.

Such machines are not without fault unless very elaborate pains are taken to see that they duplicate exactly running conditions or that all physical values are exactly known or recognized in reference to the actual gear in service. As Mr. Halgren has mentioned, sometimes these faults can be turned into advantages. As mentioned in a previous publication,[a] our attempts to recreate in the laboratory one form of pitting on gear-tooth surfaces which occurred normally in field service finally resulted in the addition of small amounts of abrasives in test boxes which reproduced the failure exactly as is shown in Fig. A. Both identical gears were run at the same load for the same length of time. The pitted one ran with abrasives in the oil. The recognition of this form of failure resulted in the addition of filters in transmissions and final drives of the higher horsepower tractors now in production; the results—greater pitting resistance.

All laboratory test results do not turn out as well as this. For instance, a few years ago, laboratory tests were conducted on a new steel for gears. The tests included single-tooth fatigue, single-pair running tests, and chassis dynamometer tests. Performance in every case was found to be satisfactory in the laboratory. Being naturally cautious about the wholesale adoption of such materials in production, quite a number of gears were then sent out for final testing in the field. The results were only slightly less than disastrous. Carburized case crushing due to a fairly soft core took place because of sudden high increments of load, bordering on impact, in actual operation. No evidence was seen of such a failure in the laboratory even though very high operating loads were applied. Such an example illustrates the constant need for field follow-up before production releases. It also, of course, reminded us how severely the load conditions can be in a track-type tractor.

Fig. C—Test setup using opposed transactors

Of course, the laboratory can do things which the operating machine cannot do. Sometimes this is beneficial, other times detrimental. For instance, at extremely low constant speeds, a severe wear takes place on gears whose surface hardnesses are over Rockwell C 60 as shown in Fig. B. This is representative of pure boundary lubrication conditions. Such wear is not obvious under actual field operating conditions because of the normal load cycle of the gears. Similar cycle work in the laboratory diminishes the wear considerably. The results are that when tests of some failure are desired, other than wear, such as pitting, it is necessary that some proper load cycle sequence be selected for the test, if the normal operating speeds of such gears are extremely low.

Any laboratory test machine which contains an external stationary method of loading may be cycled either with a changing hydraulic pressure, a cam arrangement, or a changing electrical field. We are now using such cycling devices on our chassis dynamometers and gear test machines. Some time ago, however, we had a back-to-back tractor arrangement shown on Fig. C, where the torque was statically applied and locked through the transmissions, final drives, and chains of the two tractors. Since the load could not be applied through any external loading devices, the load had to be readjusted periodically to compensate for normal wear. Its primary purpose was final drive tests since, by taking the load out of only one side of the tractors, the final drives could be overloaded without failing the transmission parts. The inability to cycle the machine under such conditions severely restricted its use since severe wear occurred at the lower speeds.

A problem arises with single-tooth testing which Mr. Wulpi did not mention, and that is the effect of frictional force between the loading device and the tooth being loaded. As the load is applied, the tooth and rim or web of the gear will deflect considerably. If the hub of the gear is rigidly fixed or is not free to move, then a frictional force will be exerted toward the root of the tooth, reducing the maximum fillet stress. This reduction may be fairly high, perhaps up to about 15% because of a high static coefficient of friction in the contact area. Because of this, all of our gears in this type of test machine are mounted on shafts encased in rubber bushings, as shown in Fig. 20, allowing the entire gear to move laterally as the tooth deflects downward. The amount of movement is visually significant. The necessity of this movement is also offered as a reason for keeping the speed of loading down particularly for heavier gears.

These few illustrations show that laboratory testing has its drawbacks, but we have found these are more than offset in terms of compensations in rapid development, evaluation, and fundamental research.

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[a] SAE *Quarterly* Transactions, Vol. 3, April, 1949, pp. 354–368: "Gear Testing Methods for Development of Heavy-Duty Gearing," by R. P. Van Zandt and B. W. Kelley.

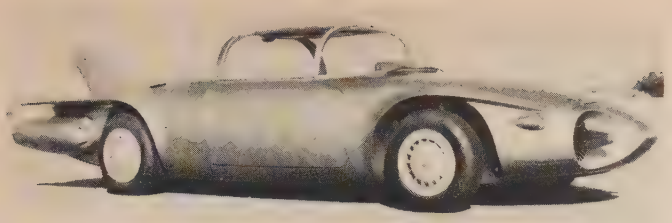

Fig. 1—Overall view of new design of Firebird II

The Experimental Chassis for the FIREBIRD II

Joseph B. Bidwell and Robert E. Owen,

Research Staff, GMC

This paper was presented at SAE Summer Meeting, Atlantic City, June 5, 1956.

THE general arrangement of the major components of the Firebird II (Fig. 1) are shown overlaid on the outline of a large current production car in Fig. 2. The gas-turbine engine is located forward between the frame side rails and front suspension arms. The propeller shaft is driven from the output of the power turbine through integral reduction gears. The 3-joint propeller shaft is enclosed in a narrow tunnel from which the center joint is supported. The transmission and final drive gears are mounted to the chassis behind the passenger compartment. All wheels are independently sprung, and the rear wheels are driven by two axle shafts from the transmission differential unit. The passenger space is located further forward relative to the wheels than in present cars. The short distance between the centerline of front wheels and the dash is possible as a result of the small tire diameter and the rear transmission position. This arrangement permits an almost clear front compartment floor unobstructed by the usual transmission hump.

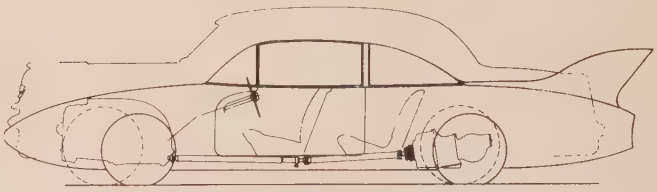

Fig. 2—Location of major components in Firebird II as compared to current production car components

THE FIREBIRD II represents an interpretation of future trends in powerplant and chassis design.

It was conceived as a high-speed, turbine-powered vehicle to carry its four passengers comfortably for long distances on smooth highways.

The chassis and suspension provide a smooth level ride, the brakes give adequate stopping power, and the hydraulic system furnishes the means for powering accessories for convenient and easy car control.

Accessories

The alternator, air conditioning compressor, and the hydraulic-system oil pump are mounted on and

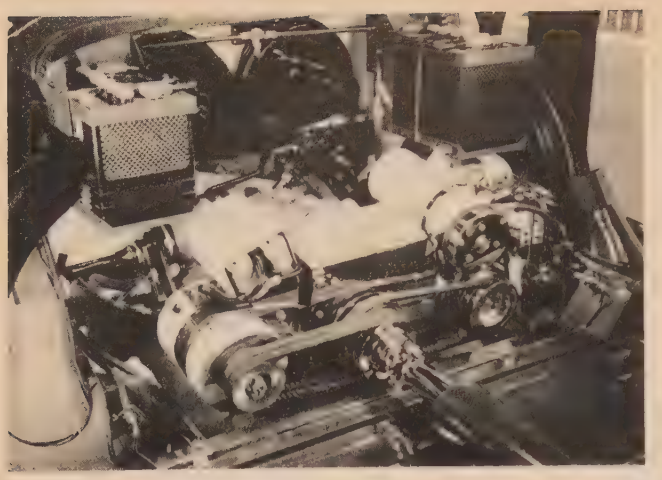

Fig. 3—Accessory-drive arrangement on transmission

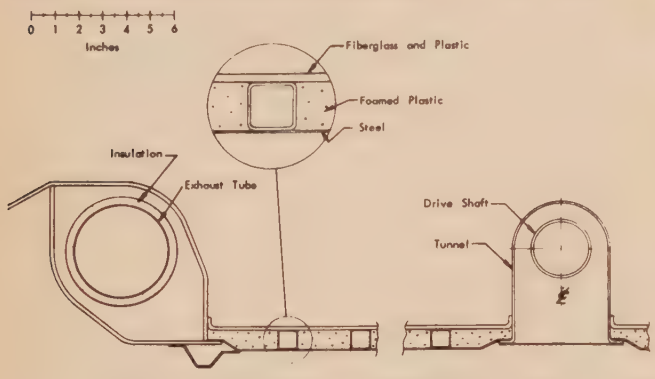

Fig. 4—Section of frame and floor construction at center of car

driven by the transmission input as shown in Fig. 3. The power turbine is coupled to the transmission through a fluid coupling. At idle, slip of this coupling permits the power turbine to continue to drive the accessories even though the car is not moving. The speed ratio between maximum speed and idle is somewhat less than in conventional cars, thus easing the accessory-drive problem. Batteries, blowers, and rectifiers are also located in back of the seat. Fuel tanks are mounted outside of the body proper under the rear fenders.

An air conditioning system, provided for passenger comfort, was supplied by Harrison Radiator and is conventional in principle. The transmission-driven axial piston compressor and blower-cooled aluminum condensor are in the rear compartment. The aluminum evaporator core is located under the dash near the fresh-air intakes. Heat is provided in the passenger space by an oil-to-air heat exchanger since cooling water is not used in this engine. Complete dimensions of the car are tabulated in Table 1.

Frame and Body Structure

The large transparent canopy dictated a frame and body structure capable of providing the required rigidity without appreciable deflection of the roof.

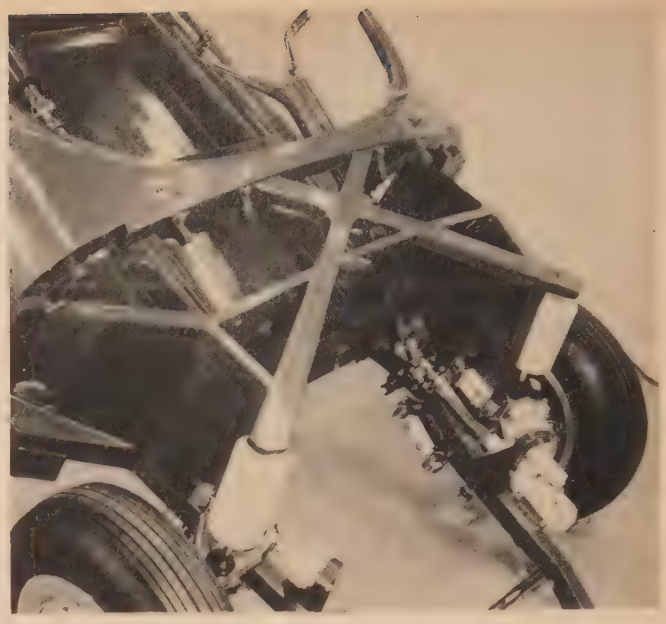

Fig. 5—Front suspension structure

This was accomplished by making the frame and body structure integral and bonding the body panels directly to the primary structure. Two cars were built, one with titanium skin and the other with fiberglass reinforced plastic body panels.

One problem associated with a low car design is that of providing space for an exhaust system and for power and control elements which must connect the front and rear of the car. Fig. 4 is a section through the frame and floor near the center of the car which illustates our solution to this problem. Large sill members and a structural tunnel provide the necessary bending and torsional stiffness through the center section of the car. The sills contain the exhaust tubes which enter near the dash and leave the sill ahead of the rear frame extension to join the exhaust outlets on top of the body behind the passenger space. The 4-in. diameter stainless steel tubes are insulated with an asbestos aluminum blanket and are spaced away from the inner walls of the sill by brackets. A small blower in the rear compartment draws cooling air through the clearance space around the insulated tube.

The flat floor is made up of a thin steel bottom plate to which are attached longitudinal steel tubes of square cross-section on 3-in. centers. The intermediate space is filled with foamed plastic and the whole assembly is covered with a layer of fiberglass reinforced plastic. The total floor thickness is 1 in. In addition to providing a rigid assembly and a smooth underbody, the square tubes provide protected channels for wires, fuel, and hydraulic lines between the front and rear of the vehicle. The propeller shaft is enclosed by a cover, and its center bearing is supported by rubber mounts attached to the tunnel.

The frame structure for supporting the front suspension is shown in Fig. 5. The frame extensions are of conventional box section. The side rails are spliced

ahead of the front cross-member to permit removal of the nose section (not shown here) for easy access to the engine. The body panels from the centerline of the front wheels forward are attached to the nose section. Housed within the nose section are the intake silencer and the oil coolers.

The front suspension arms are mounted to a vertical member which is attached at its upper end to an "X" shaped diagonal cross-brace, the ends of which support the spring seats. The cross-brace is continuous through the dash and is welded to the cowl bar and hinge pillar which is in turn attached to the sill members. Shear panels are used between the suspension arm supports and the pillar as well as over the top of the dash and cowl bar to add rigidity. The dash-to-cowl cover sheet is not shown in Fig. 5.

The rear frame structure shown in Fig. 6 is similar to the front in that the lock pillar, a rear cowl bar, and the spring support hoop are joined to form a rigid structure. The box section frame extensions join the sill members behind the reinforced exhaust tube outlets. A cross-member is provided just behind the rear seat to support the front transmission mount. The square tubes through the floor terminate at this member. Another cross-member behind the rear wheels serves as a support for the rear of the transmission and for the rear suspension arms.

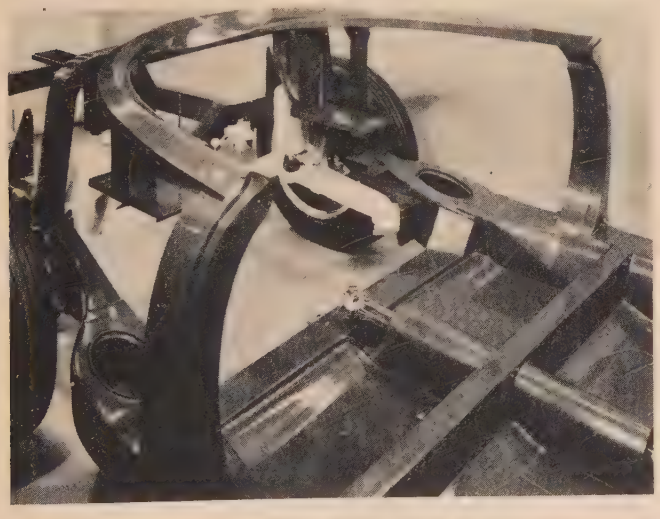

Fig. 6—Rear frame structure

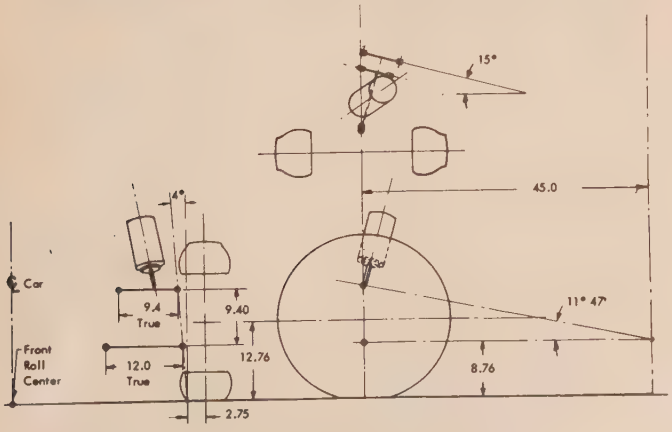

Fig. 7—Front suspension geometry

Front Suspension

The geometry of the front suspension is shown in Fig. 7. A short and long parallel arm arrangement is used which has its roll center at the ground. Caster, camber, and tread change are shown in Fig. 8. The relatively short arms used were dictated by the engine space requirements but they are adequate in view of the small ride clearance provided. This clearance is ± 2¾ in. both front and rear. Though this may seem somewhat low, it provides sufficient clearance for operation on the smooth roads over which the car is intended to travel. The integrated design of brake, knuckle support, and wheel made possible the relatively low king-pin inclination of 4 deg and 2¾-in. ground offset.

The axis of the upper arm in side view slopes down

Table 1—Chassis Dimensions

No. of Passengers	4
Wheelbase, in.	120
Front Tread, in.	60
Rear Tread, in.	57
Overall Length, in.	235
Overall Width, in.	70.6
Overall Height, in.	52.8
Front Overhang, in.	49
Rear Overhang, in.	66
Cowl Height, in.	36.8
Ground Clearance, in.	5.5
Turning Radius, ft	26.7
Curb Weight—Oil and Fuel Tanks Full, lb	5300

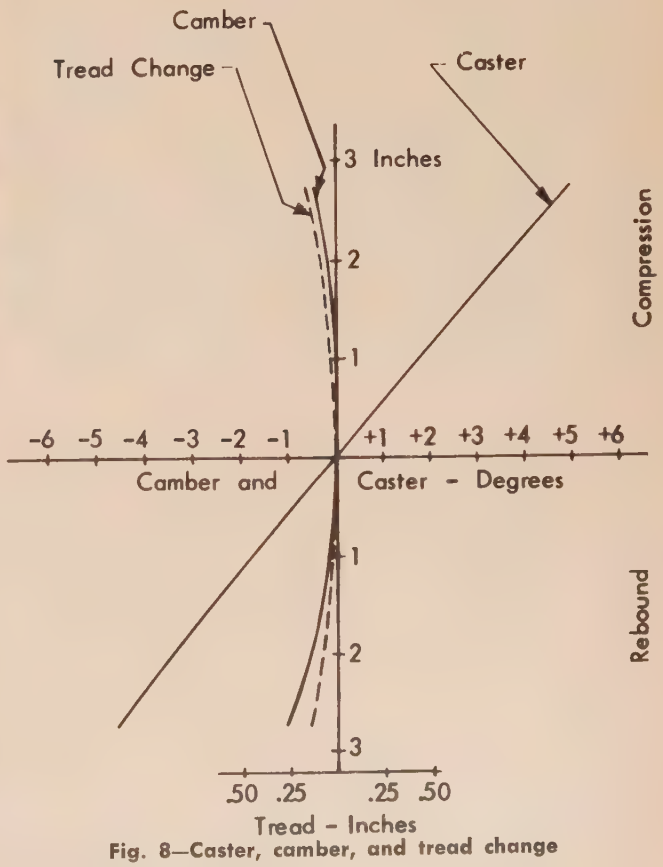

Fig. 8—Caster, camber, and tread change

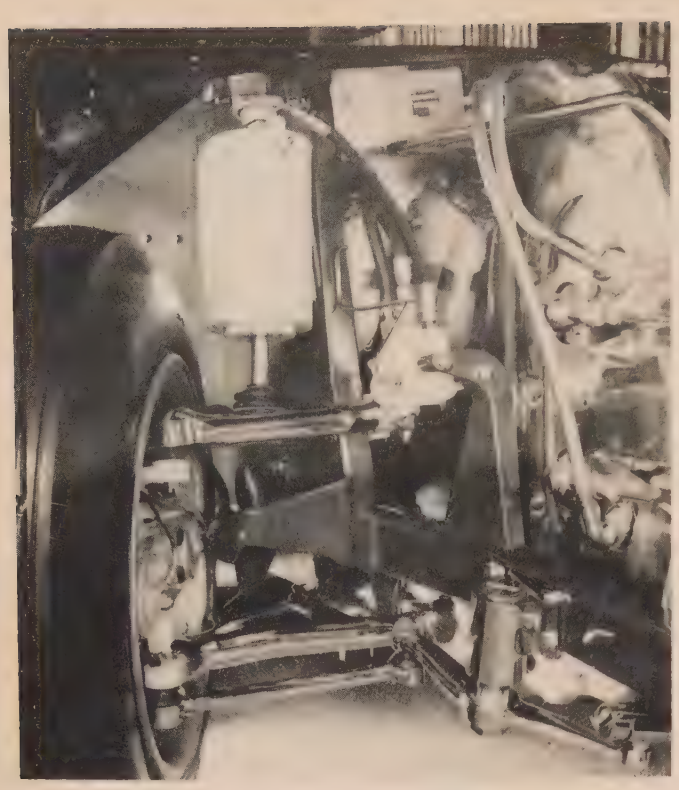

Fig. 9—Front suspension details

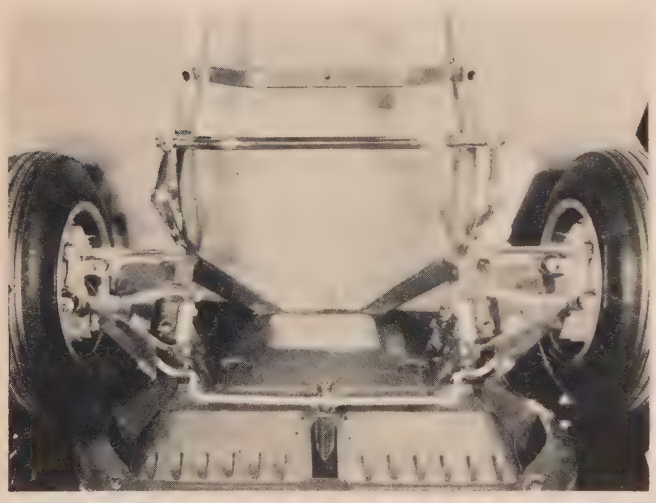

Fig. 10—Underside of steering linkage

toward the rear of the car providing a force-resisting brake dive. The arms (Fig. 9) are machined from 75 ST 6 aluminum. The forged steel steering knuckles are supported on plastic-coated balls which bear directly in ball seats machined in the aluminum arms. These joints are permanently lubricated at assembly. The inboard ends of the arms are pivoted on cylindrical bushings of similar material.

Caster and camber adjustment is provided by shims and serrated plates on the inboard support of the lower arm. The spring reacts on the upper arm through another plastic coated ball and aluminum seat. The joint is protected from dirt by a rubber boot. The upper end of the spring reacts on the frame cross-brace through a knife-edge pivot, the seat of which is supported on a rubber pad for noise insulation. Hydraulic leveling valves for the suspension are actuated by the upper arm. Bump and rebound stops are mounted on a frame projection between the two arms. The brake mechanism is attached to the steering knuckle and surrounds the live spindle. A roll bar is connected between the lower arms to obtain the desired front roll stiffness.

Steering System

The forged steel steering arms project forward of the wheel center as shown in Fig. 10. The hydraulic power steering gear is mounted on the left side rail and an idler arm is located symmetrically on the right. Tie-rods join the rear ends of these arms to the steering arms while the forward ends are connected by the idler tube. This double-ended arm arrangement was utilized to provide clearance for the engine and produce the desired steering geometry. The steering gear is a hydraulically boosted ball nut gear made by Saginaw Steering Gear. Closed center valving which closes off both the power piston and the supply ports when the spool is in its neutral position permits the use of this gear with the central hydraulic system to be described later. Two universal joints are used in the steering column to clear the engine and permit the desired steering wheel attitude.

Rear Suspension

Diagonal swing arms, as shown in Fig. 11, are used in the rear suspension. The swing axis intersects the axle shaft in plan view 3 in. from the centerline of car (Fig. 12). In elevation, the axis is parallel to the ground but 2.5 in. below the wheel centerline. The wheel hub is supported on box section arms. The forward ends of the arms are pivoted on brakets attached to the sill members while the rear pivots are on a cross-member which also supports the rear

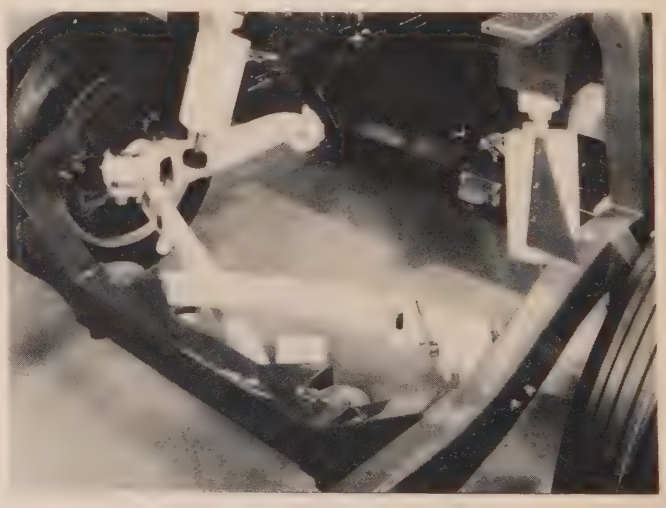

Fig. 11—Rear suspension elements

transmission mounts. The rear arms cross one another behind the transmission which prevents simpler positioning of the rear pivot. The arms turn on teflon-coated bushings which are mounted in rubber for noise isolation. A boss at the inside of the wheel hub serves as a spring seat. A leveling valve mounted between the frame and each arm serves to maintain a level attitude. As in the front suspension, the spring bears in a ball seat at the lower end and in a knife edge at the upper. The upper spring reaction is carried on a box section hoop welded to the frame and the rear cowl bar. A bump stop is provided between the frame and wheel hub. Rebound is limited by a strap around the arm.

The transmission mounts at a single point in front and on two arms which go over and under the suspension arms in the rear (Fig. 13). Relatively wide spacing of mounts is thus provided for torque reaction. Short drive-shafts with two universal joints are used between each wheel and the differential. A ball and trunnion joint is used at the inboard end because of the relatively large angles at full bump and rebound. A single Cardan joint is used at the wheel.

Table 2 is a tabulation of suspension characteristics for the fully loaded car. The front and rear equivalent static deflection are similar to those employed today and provide satisfactory ride within the limits of the ride clearance allowed. The roll stiffness is appreciably greater than most current cars and coupled with the low center of gravity, results in little roll in cornering. The front and rear geometry maintains level attitude in braking.

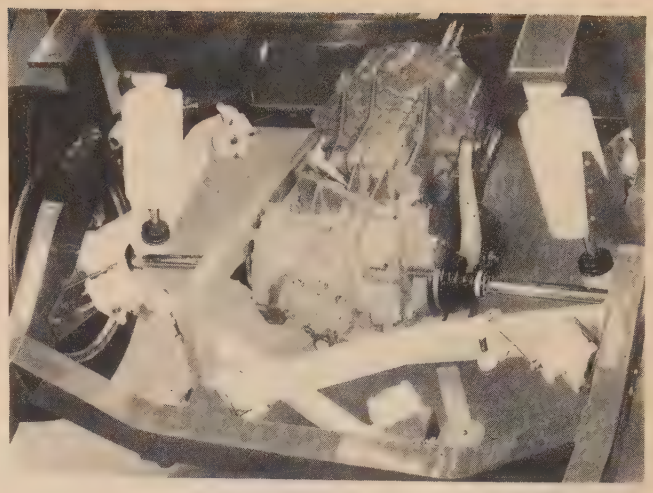

Fig. 13—Transmission and axle shaft arrangement

Table 2—Suspension Characteristics

	Front	Rear
Type	Parallel short and long arm independent with antidive	Independent diagonal swing arm
Springs	Fixed air mass with hydraulic leveling	
Weight (Curb), lb	2700	2600
Weight (4-Passenger Load), lb	2980	2960
Ride Clearance, in.	±2.75	±2.75
Equivalent Static Deflection (4-Passenger Load),[a] in.	13.25	9.35
Antidive, %	66	
Antilift, %		126
Roll Center Height, in.	At ground	11.3
Roll Stiffness, lb-ft per deg	549	327
Steering Ratio	18/1	

[a] Equivalent static deflection calculated from wheel rate and wheel load with four passengers and full fuel and oil tanks. Springs are nonlinear, and rate changes with load.

Springs

The springs utilized on the Firebird II were made by Delco Products and are shown schematically in section in Fig. 14. These units employ a fixed air mass as the spring medium. Oil is used to make up volume change of the air with load to maintain a constant car attitude. In this way, the need for an air pump is eliminated, and the required power is supplied by the hydraulic system. A leveling system was considered essential in view of the limited total ride clearance.

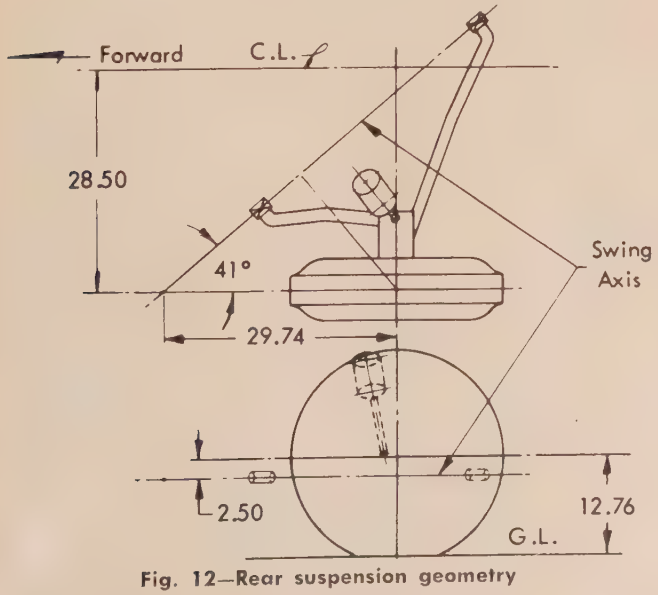

Fig. 12—Rear suspension geometry

The spring is designed so that air is retained between the outer shell and a cylindrical rubber diaphragm. The pressure is transmitted through oil to the top of a piston which supports the load. The oil passes through internal shock valving to get from the cylinder into the surrounding space. Thus, the oil in the unit is under pressure, and aeration of the oil passing through the shock valves is prevented. Oil pressure at 4-passenger load is of the order of 750 psi. A separate leveling valve which senses suspension displacement permits oil to either enter or leave the spring as required. Leveling is only permitted at car speeds below 10 mph. The springs are locked out by transmission governor oil pressure which closes a cut-out in the leveling valves at higher speeds.

Brakes

Exceptional braking capacity is certainly one of the most important features required in a high-speed car, especially in view of the lack of engine braking from the gas turbine powerplant. The disc brakes with

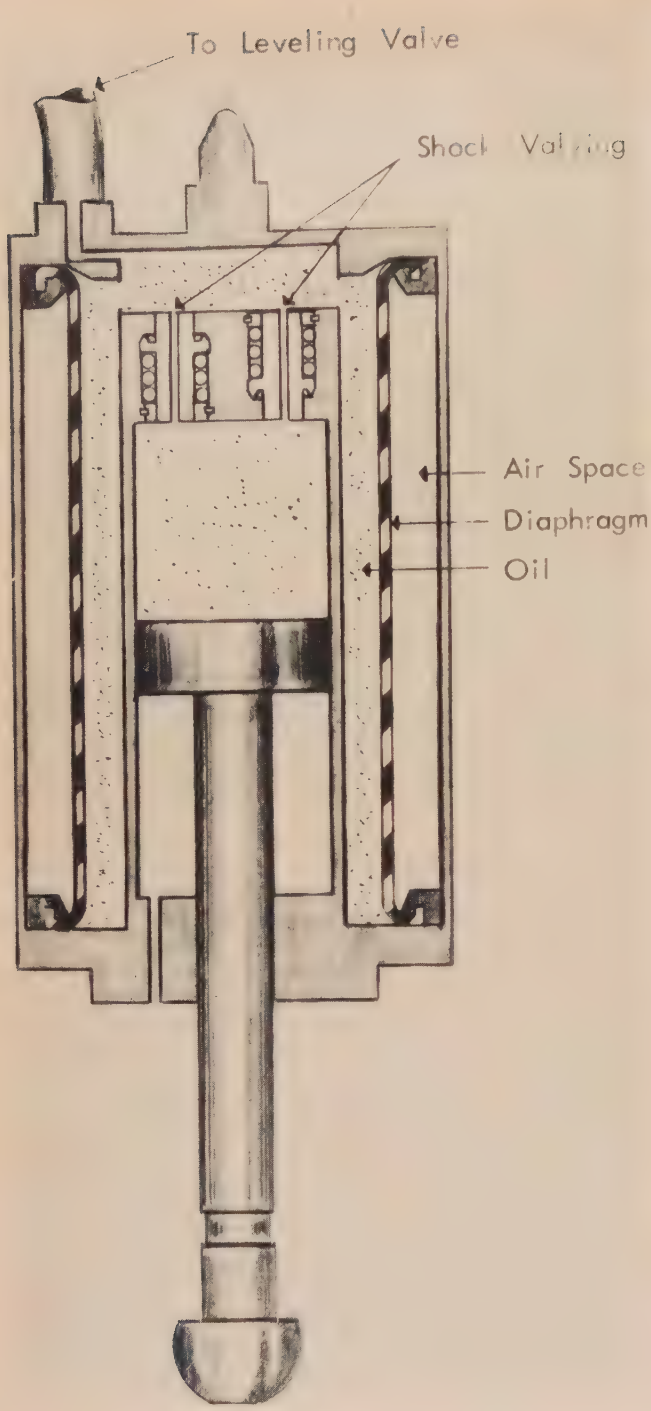

Fig. 14—Schematic section showing air-oil spring

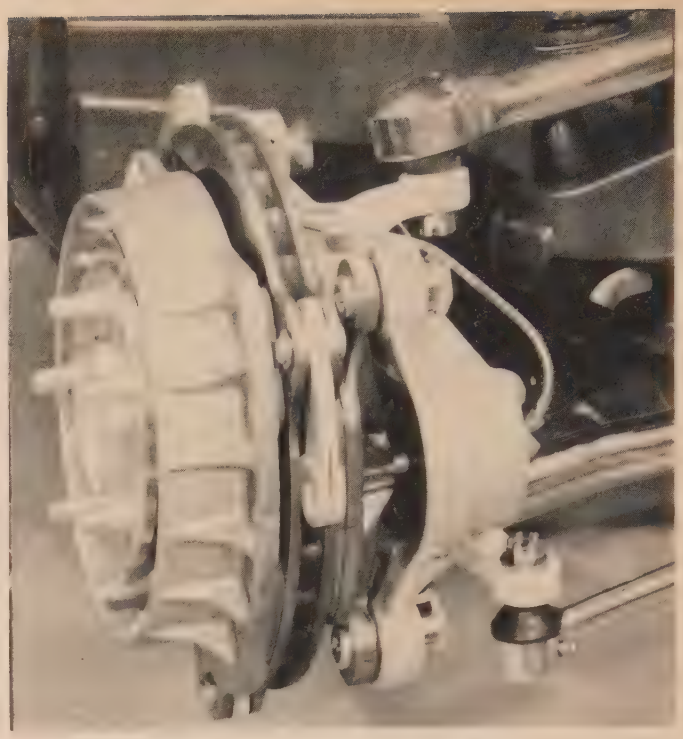

Fig. 15—Disc-brake design

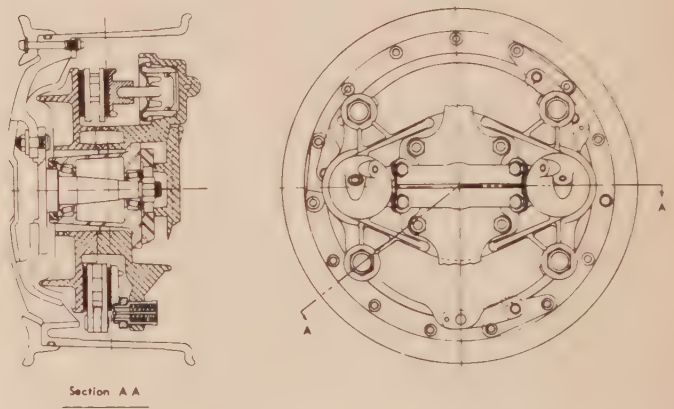

Fig. 16—Section through brake and wheel showing design

metallic linings and hydraulic booster used on the Firebird satisfy this need. These brakes were built by Moraine Products and are an adaptation of one of their designs. The compact hydraulic brake booster was made by Saginaw Steering Gear and provides a 3/1 boost ratio. Figs. 15 and 16 illustrate both the brake and wheel design. The wheel has a 16-in. rim with a separate flange. Elimination of the drop center makes a large diameter available for the brake. In this design, the iron brake disc with cored radial air passages is driven by three tangential links from pins in the wheel. This construction permits expansion of the disc without distortion of the wheel. The inner and outer aluminum brake structure is joined inside of the disc rather than outside, thus permitting a somewhat larger disc diameter. Two wheel cylinders are provided for each brake. Each actuates a separate set of metallic pads. The wheels are designed to circulate air over the disc, and the central vanes in the disc circulate air and produce cooling from the center.

The extensive cooling provided coupled with the temperature-resisting characteristic of the lining material and the absence of self-actuation results in the desired fade resistance.

Tires

Wheels and tires for the Firebird II were designed to provide adequate space inside the rim for brakes

and steering linkage and to allow brake cooling without an excessively large overall diameter. Figs. 16 and 17 illustrate the final design. A 16-in. diameter x 7.5 in. wide rim section is used. The radial tire thickness is reduced so that the overall diameter of the 8.20 tire is 27.3 in. This tire section provides good steering response with some sacrifice in softness. In order to maintain satisfactory sidewall deflections, inflation pressures of 42 psi are used both front and rear.

Wheels

The wheels are made of cast magnesium and are chromium plated. The separate rim is sealed with a large "O" ring to permit the use of tubeless tires and is held in place with bolts. The wheel spokes and the rim flange act as a blower for cooling the brake disc. Three steel pins bolted to the inside of the wheel engage and drive the disc links.

Hydraulic System

A central hydraulic system using Type A transmission fluid provides the necessary power for suspension leveling, brake booster, power steering, and windshield wipers. The system is shown schematically in Fig. 18. The pump, built by Saginaw Steering Gear, operates whenever the system pressure drops below 850 psi and bypasses when the pressure reaches 1100 psi. The pump has a capacity of 1.4 gpm at 2000 rpm of the propeller shaft. Two accumulators, one at the front and one at the rear of the car, serve to store energy to meet peak load demands. The high pressure and return lines to the pump and integral sump are installed in the fore and aft square tubes through the floor to provide service to brake booster, wipers, steering, and front suspension at the front of the car.

Conclusion

The Firebird II chassis is shown in Fig. 19 complete with engine, transmission, and accessories. The novel chassis features employed in this vehicle contribute to its performance and suit it to the operating conditions for which it was designed. The chassis and suspension provide a smooth level ride with little roll or change in attitude when braking or cornering. The brakes supply adequate stopping power for high-speed operation of a gas-turbine-powered vehicle with its lack of engine braking. Passenger compartment temperature control and ventilation assure comfortable travel. The hydraulic system provides a means for powering accessories for convenient and easy car control.

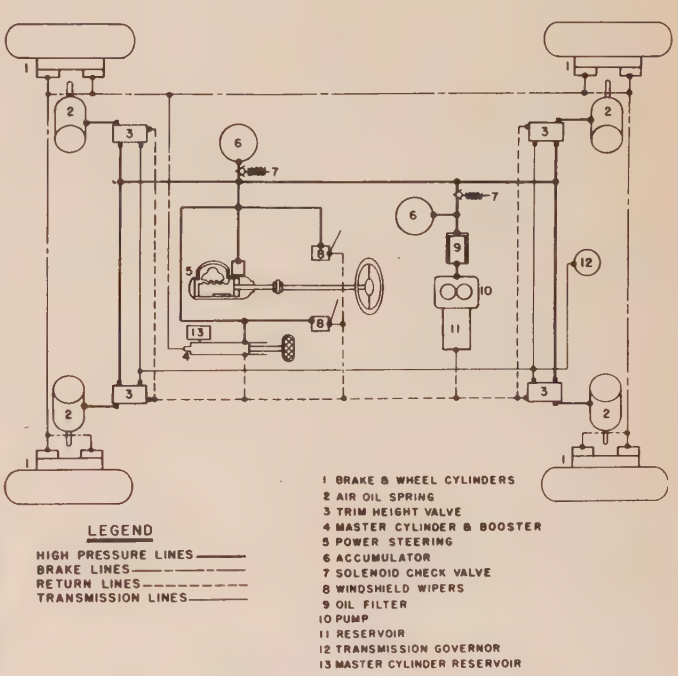

Fig. 18—Schematic diagram of hydraulic system

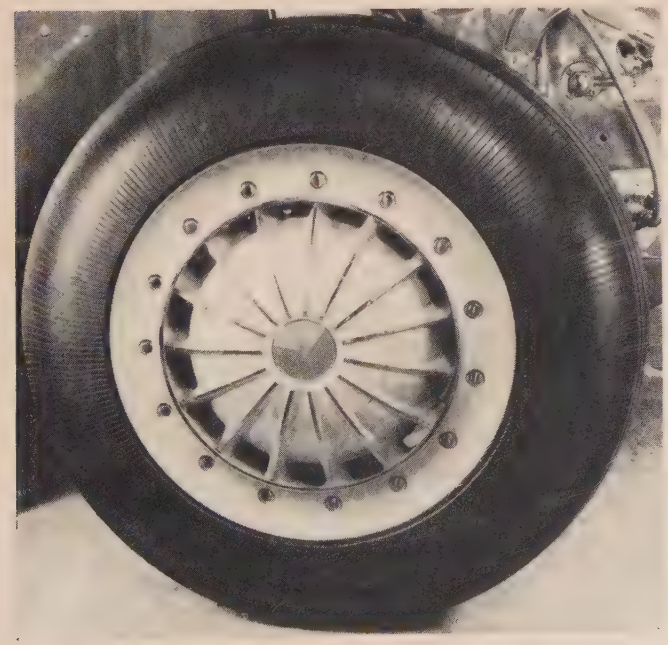

Fig. 17—Wheel and tire design

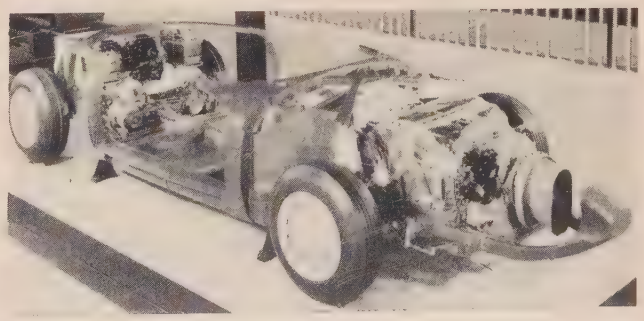

Fig. 19—Firebird II chassis with engine and transmission installed

The REGENERATIVE WHIRLFIRE

THE subject of automotive gas turbines continues to receive a great deal of attention not only in the technical journals but also in the laboratories of many of the major manufacturers. The public interest that has been aroused justifies reporting on the progress of these experiments. In two previous papers [1,2] our Research Staff presented first a theoretical analysis of the automotive gas turbine and then a progress report on early experiences with actual vehicle installations. These first installations demonstrated the feasibility of gas turbines as vehicular powerplants. In an effort to overcome the shortcomings of these earlier models, a new engine, the GT-304 Whirlfire incorporating fuel-saving regeneration, has been built. Basic design considerations and preliminary operating experience are the basis of this third report on vehicular gas turbines.

In order to establish the reasoning behind the design concept of the model GT-304 Whirlfire engine, it is well to review the results of the earlier installations in the "Firebird I," the first American gas-turbine automobile, and the "Turbocruiser," the first gas-turbine-powered bus. Both of these vehicles are shown in Fig. 1. Firebird I is a single-seat automobile with the engine mounted in the rear. This high-performance vehicle was designed for test track work rather than highway use. The Turbocruiser on the other hand has accumulated most of its mileage on the public highway. In slightly over two years of operation, a total of 9000 miles of operating experience has been accumulated, all on the original engine installations.

Experience with these two vehicles indicated many desirable features of a gas-turbine installation; smooth power application, excellent acceleration above idle, favorable weight/power ratio, low oil consumption, no special fuel requirements, and durability. Some of the problems that appeared very formidable in the initial paper study were not as serious as anticipated. Included in this group were engine noise, high speed of rotation, engine-compartment heat, and engine controls. The high fuel consumption was expected when the simple cycle was adopted for the first design. Not so apparent during the early design stages was the rather sluggish acceleration characteristic of the engine between idle and 70% gasifier speed. After many miles of test driving, only these two disadvantages remained to prevent a favorable performance comparison of the gas turbine with contemporary vehicular powerplants. It was toward improvement of these shortcomings that the design of the GT-304 was directed.

THE GT-304 Whirlfire engine, incorporating fuel-saving regeneration, has been built to overcome shortcomings of earlier vehicular gas-turbine models.

The overall power package was given prime consideration. Attention was focused on design and location of each component to achieve a compact, well-integrated assembly.

These basic design considerations and preliminary operating experience are the subject of this paper.

Cycle Analyses

Gas-turbine cycle analysis has been the subject of many papers already presented. This is at least partly due to the fact that the gas turbine is quite amenable to thermodynamic analysis and leads to the "production" of many paper engines. In the past, the bulk of information has represented only design-point conditions, since obtaining part-load data was prohibitively

[1] *SAE Journal*, Vol. 57, July, 1949, pp. 18–23: "Gas Turbine Possibilities for Cars Analyzed," by W. A. Turunen.
[2] SAE Transactions, Vol. 63, 1955, pp. 72–83: "Pinwheels or Pistons?—A Progress Report on Automotive Gas Turbines," by W. A. Turunen.

ENGINE for FIREBIRD II

W. A. Turunen and J. S. Collman,
Research Staff, GMC

This paper was presented at the SAE Summer Meeting, Atlantic City, June 5, 1956.

tedious and time consuming. Now, however, electronic computers bring part-load cycle analyses within the realm of possibility.

A computer program has been developed by our Research Staff that will simulate any engine operating condition. The effect of component characteristics on engine performance can be studied throughout the load range. By means of this program, over 90 engines were evaluated in arriving at the design specifications of the GT-304.

Fig. 1—Gas-turbine-powered Firebird I and Turbocruiser

Component Efficiencies

Some of the cycle parameters must be established at the highest attainable values within the design limitations. These include component efficiencies, internal leakage, and pressure losses. The importance of each of these factors on a typical regenerative cycle is illustrated in Fig. 2. Since these factors have the same qualitative effect at all load conditions, only design-point data are shown for simplicity. It can be seen that a change in the gasifier turbine efficiency is twice as significant as a corresponding change in power turbine efficiency, while a change of similar magnitude in the compressor is nearly three times as effective on thermal efficiency.

The effects of system losses are also illustrated in Fig. 2. For the assumed conditions, a change of one percentage point in either pressure loss or leakage has the same effect on thermal efficiency as a one percentage point change in compressor efficiency.

Increasing the turbine inlet temperature will also improve the efficiency of the gas-turbine cycle. The turbine inlet temperature is limited by the strength-temperature characteristics of the bucket alloy. The

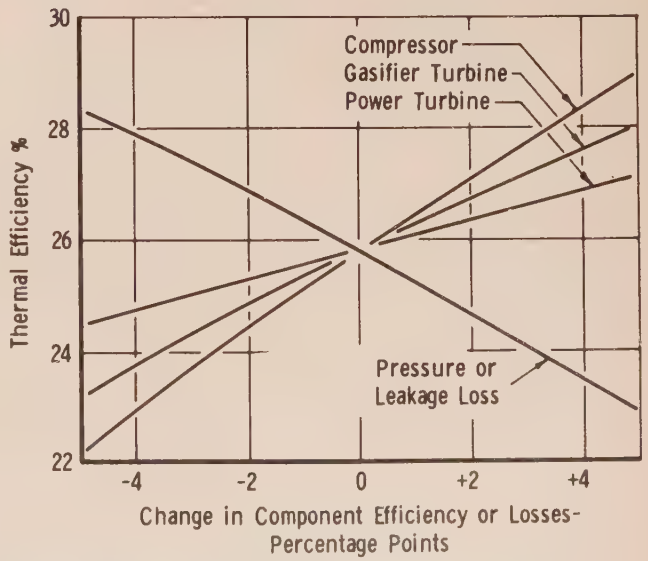

Fig. 2—Effect of component efficiencies and system losses on regenerative cycle

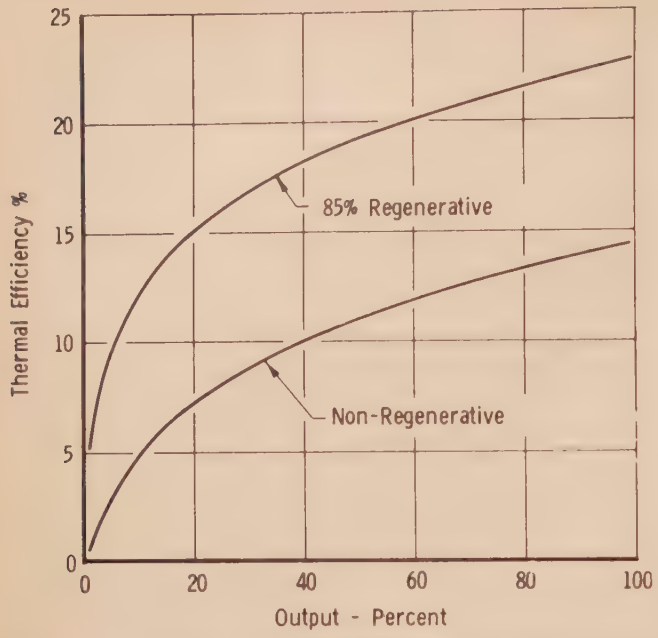

Fig. 3—Comparison of thermal efficiency of regenerative and nonregenerative cycles

earlier models of the Whirlfire engines operated at 1500 F turbine inlet temperature with buckets made of GMR-235, an alloy developed by the Metallurgy Department of Research Staff. Experience with these and other test engines indicated that higher turbine inlet temperatures could be used. On this basis, a design-point turbine inlet temperature of 1650 F was chosen for the GT-304 engine.

Regeneration

The addition of regeneration to the open-cycle gas turbine can have a great effect on overall thermal efficiency. Regeneration also tends to reduce the optimum pressure ratio to a point well within the range of the radial flow compressor. This factor is highly desirable, especially where component simplification is important. In Fig. 3, the thermal efficiency of a nonregenerative cycle, with appropriate losses and efficiencies, operating at design-point pressure ratio of 4.5/1 is compared with a typical regenerative cycle operating at 3.5/1 pressure ratio, 85% regenerator effectiveness, and 5% leakage. The regenerative cycle shows an efficiency margin of eight percentage points from full load to very low output. The advantage of regeneration is more graphically demonstrated in Fig. 4. At full load the regenerative engine uses only 65% of the fuel used by the nonregenerative engine, while at 10% load, less than 40% is required. This part-load characteristic is especially attractive for automotive applications where the powerplant operates most of the time at a small fraction of full power.

Another benefit of regeneration is reduction of the exhaust gas temperature. This is illustrated in Fig. 5, where the previous cycles are again used for comparison. The exhaust temperature of the regenerative cycle is about 600 F lower than the nonregenerative cycle. At full load the regenerative engine exhaust is about 550 F, and at idle it is only about 250 F. This is a significant reduction in the amount of heat discharged to the atmosphere and reflects the increased thermal efficiency of the regenerative design.

There are two basic methods of regeneration that may be considered for the gas-turbine cycle. The first is the stationary type, in which heat is transferred directly from the hot turbine exhaust to the compressor discharge air through the flow passage walls of the heat exchanger. The second is the rotating type, wherein heat is carried by a rotating matrix from the exhaust to the compressor discharge air. Characteristic features of each method must be evaluated in selecting the type that is most likely to succeed in a given application.

The stationary heat exchanger is a very familiar piece of equipment. Vast amounts of data are available for its design. There is no leakage from the high- to the low-pressure fluid as long as the structural integrity is maintained. Reasonable effectiveness and acceptable pressure loss usually results in a bulky unit requiring large ducts that further complicate the space and arrangement problem.

The rotating heat exchanger, on the other hand, presents high effectiveness coupled with small size. Offsetting these advantages, however, is the problem of leakage of high-pressure air past the regenerator seals. If this leakage, which bypasses the turbines, exceeds practical limits, it can more than offset the gain of higher effectiveness.

The results of a typical rotating regenerator cycle are compared with a stationary regenerator cycle in Fig. 6. Design-point effectiveness of 85% and 5%

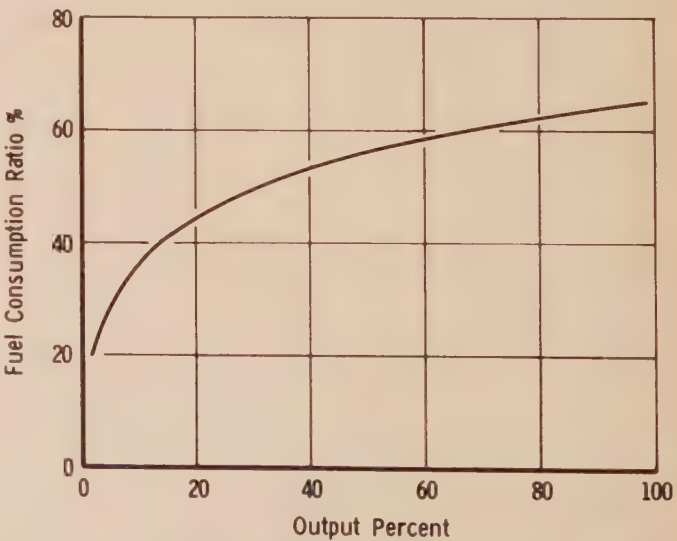

Fig. 4—Ratio of fuel consumption of regenerative and nonregenerative cycles

leakage is assumed for the rotating regenerator. Effectiveness of 60% and no leakage is assumed for the conventional regenerator. Not only is the engine assembly incorporating the stationary regenerator unit larger and heavier than the one employing a rotating regenerator, but also it shows lower thermal efficiency throughout the load range. On the basis of the results typified here, the potential of the rotating regenerator was sufficiently attractive, despite the seal leakage problem, that the design of the GT-304 engine was based on its use.

Acceleration Delay

The torque characteristic of a 2-shaft gas turbine, in which the power turbine is independent of the gasifier turbine, is considered an advantage not enjoyed by reciprocating engines. It is well known that the stall torque can be somewhat in excess of twice the torque at full power. This torque is not produced, however, unless the gasifier is turning at rated speed. A finite time interval is required to accelerate the gasifier from idle to the point where sufficient torque is developed to provide satisfactory vehicle acceleration; in short, there is an acceleration delay. In many types of applications a slight delay is acceptable, but it would not fit in with the get-away performance demanded of today's passenger cars.

Acceleration delay is a direct function of the moment of inertia of the rotating parts of the gasifier and an inverse function of the amount of energy in the gas stream available for acceleration. Excess energy required for acceleration results from increasing the turbine inlet temperature. Thus, the use of a bucket alloy that will tolerate high temperature and

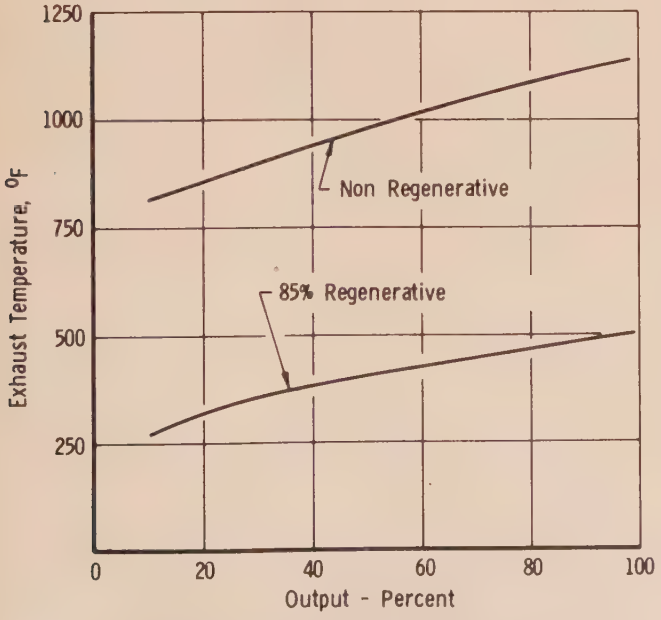

Fig. 5—Comparison of exhaust-gas temperature of regenerative and nonregenerative cycles

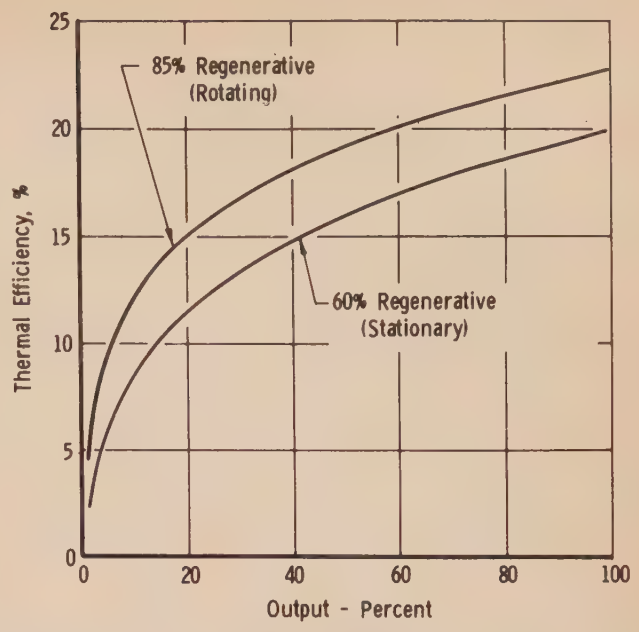

Fig. 6—Comparison of thermal efficiency of rotating and stationary regenerative cycles

repeated thermal shock is essential for good acceleration. The weight and diameter of the rotating parts of the gasifier, especially the turbine and compressor rotors, must also be held to a minimum. As these diameters are reduced, rotational speeds must be increased, and stresses go up accordingly. For best acceleration these parts must be designed for the highest speed and temperatures that stress limits will allow.

Coupling the power turbine to a transmission that has several speed changes will also improve acceleration performance. The torque characteristic of the gas turbine will allow fewer gear ratios for acceptable power delivery, but this requires large changes in power turbine speed during shifts. For smooth shifts, small speed changes as well as low engine inertia are desired. By careful design, it was possible to reduce the moment of inertia of the power section of the GT-304 engine to the point where it was equivalent to a modern V-8 engine. Four gear ratios were then chosen to cover as broad a range as possible and still have acceptable shift characteristics. With these efforts, the initial acceleration delay was reduced to a minimum while superior vehicle performance was available at road load speeds. The 4-speed transmission can be modified to provide three or two shift points only. Thus it will be possible, in our future test program, to determine the best engine-transmission combination for acceleration and performance.

General Engine Design

In designing the GT-304 engine, the overall power package was given prime consideration. Considerable thought was given to the design and location of each component in order to achieve a compact, well-inte-

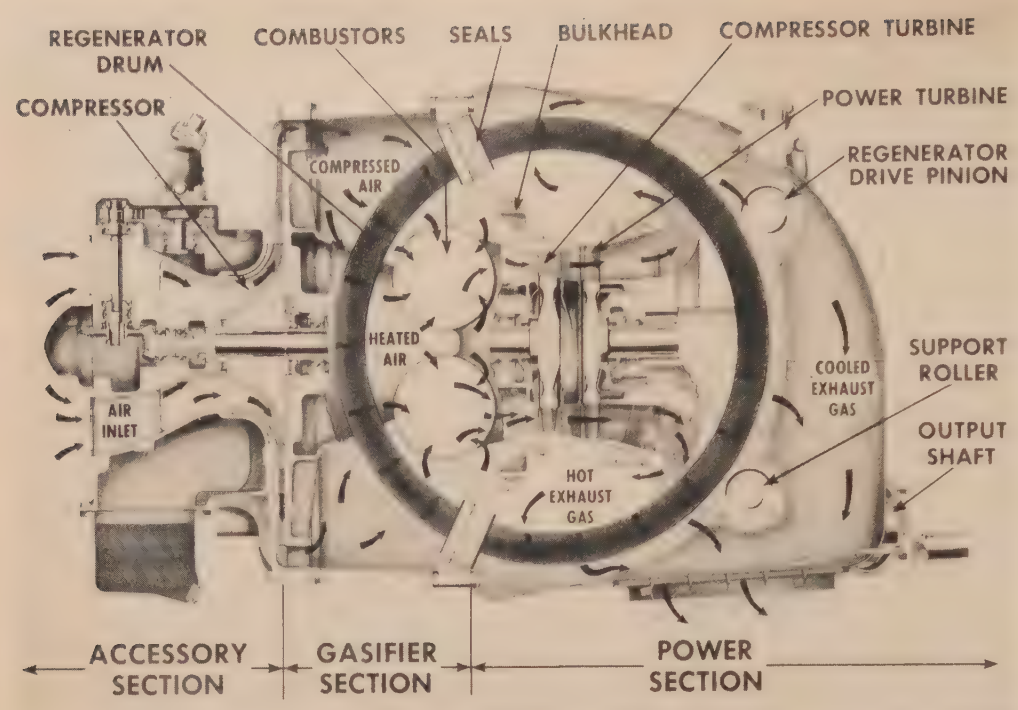

Fig. 7—Schematic arrangement of GT-304 Whirlfire engine

grated assembly. A schematic engine arrangement is shown in Fig. 7.

Air enters the radial flow compressor from the left. After passing through the diffuser, it is diverted 90 deg and discharged axially into a plenum chamber housing the regenerators, combustion chambers and turbines. The plenum is divided into a high-pressure and low-pressure section by the seals and center bulkhead assembly. Two drum-shaped regenerators rotating about a horizontal axis pass from the low pressure or exhaust side of the plenum to the high-pressure side through floating seals. The compressor discharge air passes radially through the regenerator drums picking up heat before entering the combustion chambers. The four can-type combustion chambers, two on each side of the engine, are arranged parallel to the regenerator axis. Fuel is sprayed into the chambers through nozzles located in the outer ends of the cans. The heated gas leaving the chambers is diverted into the turbine inlet annulus by symmetrical transition sections. The hot gas is expanded through two mechanically independent turbine stages located in the center cylinder. The first-stage turbine drives the compressor and accessories, and the second-stage turbine is connected to the vehicle wheels through reduction gears and the transmission. Hot gas exhausted from the power turbine is diverted into the center part of the low-pressure plenum. From there the gas passes through the regenerator drum, to which it gives up heat. The heat transferred results in a considerable reduction in temperature before the gas is exhausted to the atmosphere.

One of the important features of this integrated engine arrangement is the direct airflow path and complete absence of interconnecting ducts. The minimum number of bends in the airflow path and large plenum-type air passages contribute to very low pressure losses.

Another very important advantage of the engine arrangement arises from the fact that those parts of the engine which are exposed to, or contain, the hot gases are confined within the inner diameter of the regenerator drums. The top, bottom, front, and rear surfaces of the engine are exposed to compressor discharge air or exhaust gas cooled by passage through the regenerators. As a result, no thermal shielding or insulation is required for the engine except on the circular regenerator end covers. This reduction of heat loss from the engine minimizes engine-compartment ventilation requirements and is undoubtedly reflected in at least a slightly higher thermal efficiency.

Component Development

Because of the unusual arrangement of the GT-304 engine, several of the components were developed by means of independent test programs. Major test setups were made for compressor, combustion system, and regenerator development.

The compressor test setup is shown in Fig. 8. Inlet and discharge geometry are carefully reproduced. Various rotor and diffuser designs were evaluated to provide the best component performance and proper airflow to match the turbine requirements.

The combustion system is complicated by severe space limitations and the high burner inlet temperature which results from regeneration. Furthermore,

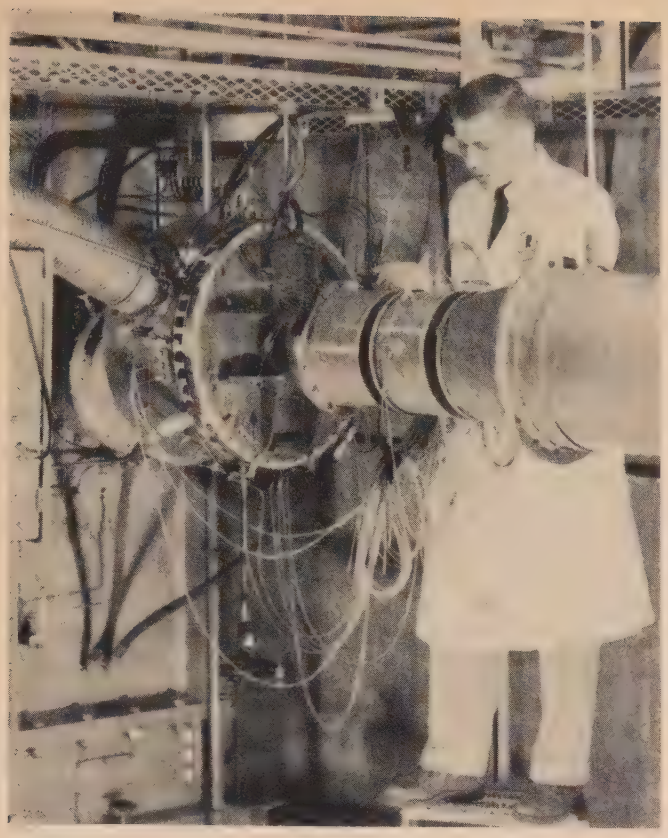

Fig. 8—Compressor test setup

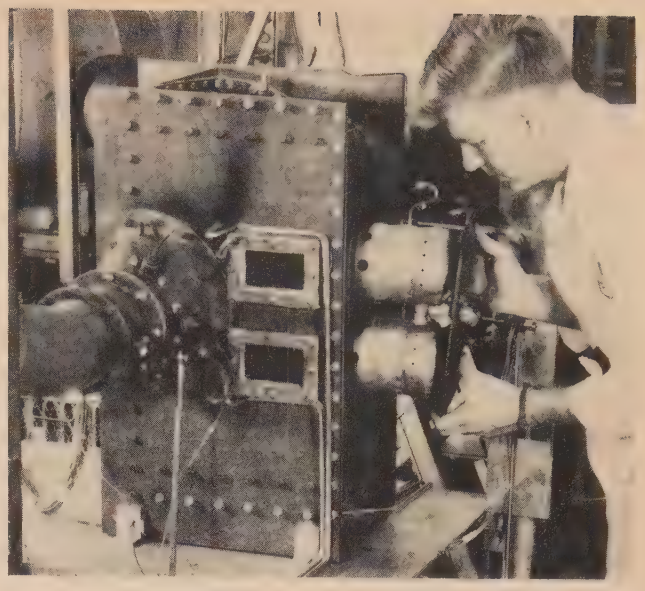

Fig. 9—Combustor test setup

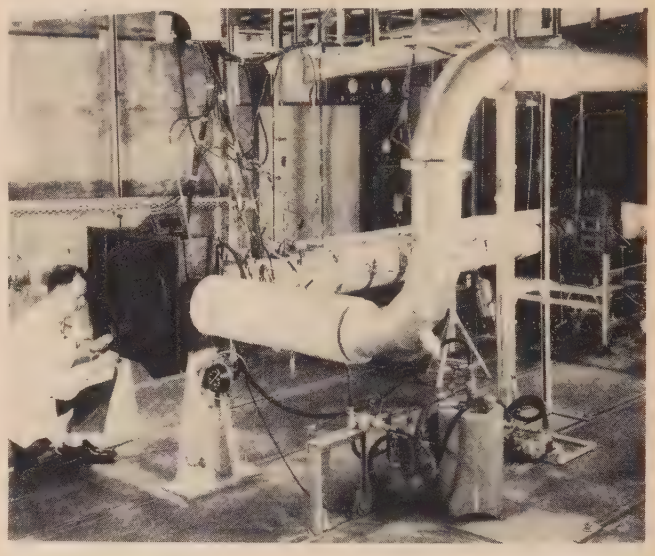

Fig. 10—Regenerator test setup

the fuel flow is reduced to such an extent that effective atomization at idle is difficult to achieve. The test setup used for combustion system development is shown in Fig. 9. Actually, only one half of the engine is simulated, requiring two instead of four burners. Compressed air supplied to the rig is heated to simulate engine operation. Several changes in liner, transition section, and nozzle design were required to provide clean burning, short flame length, and adequate stability.

The third, and perhaps most important area of testing, was regenerator development. The rig where this was carried out is shown in Fig. 10. The test regenerator housing with cover removed is in the left foreground. Air from the laboratory supply enters on the left and passes through the regenerator drum. The air then enters the loop to the rear of the test rig where temperatures and pressures corresponding to actual engine operation are simulated. The hot gas then passes through the right-hand portion of the matrix and is exhausted. Different types of matrix construction and several supporting and sealing systems have been evaluated. Seal designs have been developed with leakage of less than 5% that will accommodate the most severe operating transients.

Component tests such as these were essential to the successful development of the GT-304 engine. Each of the critical components was developed to meet design specifications that would result in satisfactory engine matching. As a result, the complete engine ran successfully and operated according to the predicted cycle analysis without modification.

Mechanical Details

The integrated design of the GT-304 provides excellent service accessibility and easily handled components. The engine comprises three major subassemblies: the accessory section, the gasifier section, and the power section.

The accessory section shown in Fig. 11 incorporates the air inlet duct, the compressor front cover, the oil sump, and drive pads for the gasifier-connected accessories. These include the automotive-type starter, fuel pump and governor assembly, built-in lube scavenge and supply pump, and the fluid power pump for the regenerator drive motor. Power for the accessories is taken from the front end of the gasifier shaft through spiral bevel reduction gears. The lube

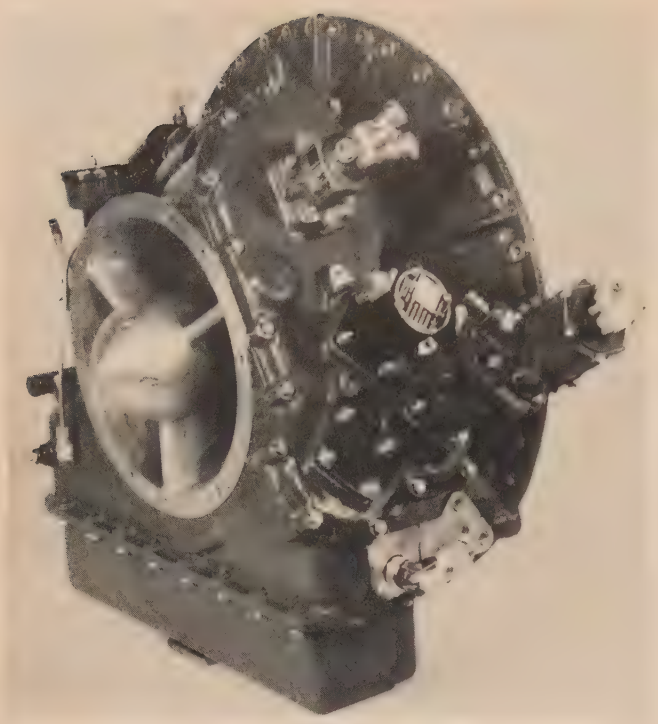

Fig. 11—Accessory section

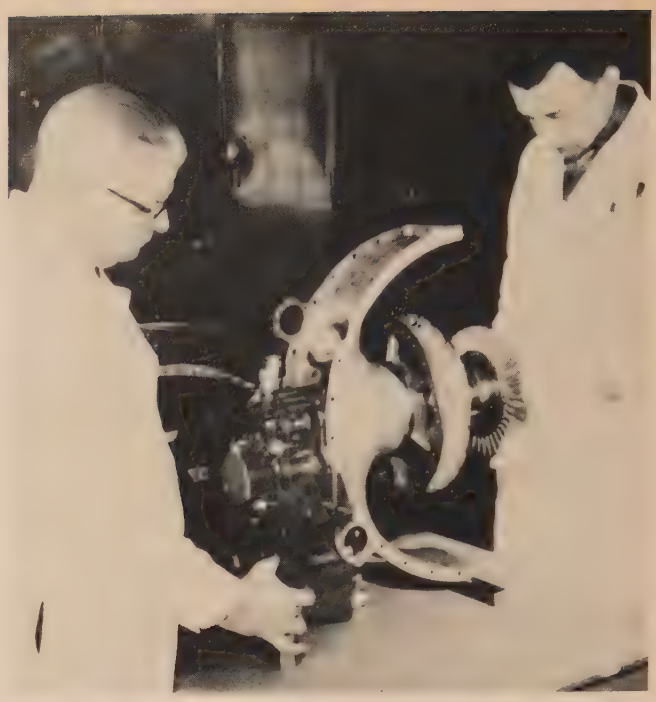

Fig. 13—Power turbine and reduction gear case

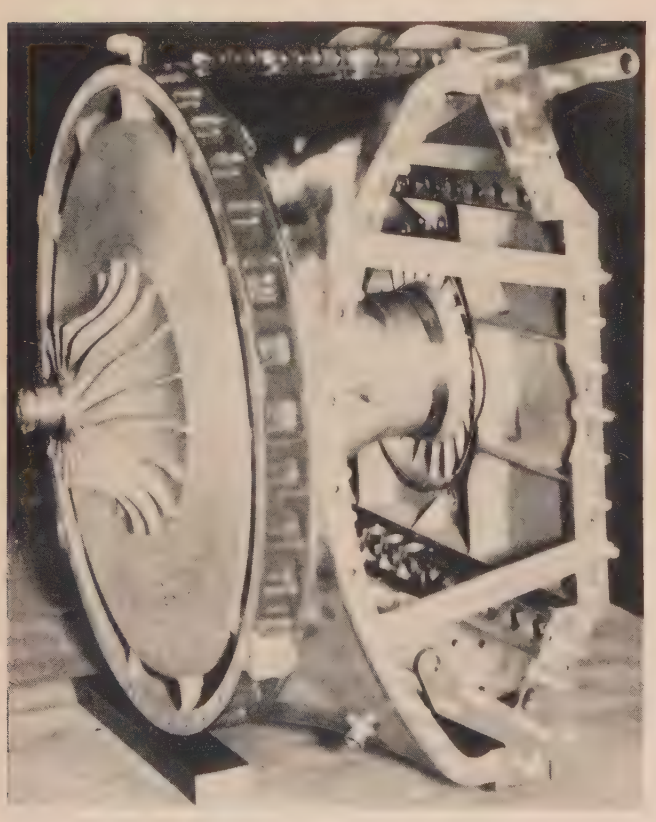

Fig. 12—Gasifier section

pump assembly is located near the bottom of the housing within the oil sump. Oil is supplied from one supply pump to all parts of the engine including the reduction gears and the fluid power pump. The arrangement of the accessories around the compressor inlet duct lends to a compact design and provides a cool environment for these components.

The gasifier section, shown in Fig. 12, includes the gasifier turbine, compressor rotor, bearing housing, compressor discharge casing, and the bulkhead and seal assembly. The rear compressor housing is an integral part of the bearing housing. The compressor discharge passage around the circumference of the housing provides an exceptionally desirable compressor outlet condition. The sheet-metal compressor discharge casing serves as a pressure vessel and also as a structural component of the engine assembly between the compressor housing and the bulkhead and seals assembly. The bulkhead and seals separate the high-pressure portion of the air plenum from the low-pressure section. The gasifier turbine nozzle assembly is attached to the forward side of the bulkhead and is also piloted on the turbine end of the bearing housing. Thus, differential expansion can be accommodated axially while radial alignment is maintained between the turbine rotor and stator components.

The power section includes the power turbine, reduction gears and the exhaust housings. The power turbine and the reduction gear case are shown in Fig. 13. A mounting bracket with two mounting pads is attached to the rear of the gear case. Exhaust housings are attached on each side to complete the power section. The power section is then attached to the gasifier and accessory section by securing the exhaust housings to the upper and lower seal bars as shown in Fig. 14. The exhaust diverter, shown in this view, prevents direct impingement of the turbine discharge onto the regenerator drums.

The regenerator drums are supported by fixed

roller shafts located in and supported by the exhaust housings. The geared rims of the drums engage pinions on the upper roller shafts to drive the regenerators from a hydraulic motor. A drum being installed in the engine is shown in Fig. 15. The regenerator seal assemblies are movable to provide perfect alignment with the regenerator drums.

The combustion chambers are installed in the space between the drums and the forward side of the bulkhead. Two liners and a transition section are shown in Fig. 16.

The completed engine assembly is shown in Fig. 17. The engine side covers complete the engine plenum enclosure and also serves as structural components, tying the compressor housing, bulkhead assembly, and exhaust housings together. Aluminum radiation shields cover the engine side panels which are the only hot exterior surfaces.

Engine Specifications

The design rating of the GT-304 engine is 200 hp at a gasifier speed of 35,000 rpm and turbine inlet temperature of 1650 F. Allowance is made for all engine accessories, inlet and exhaust losses, and reduction gear losses. Rated power turbine speed is 28,000 rpm with an overspeed allowance to 35,000 rpm. A 7.27/1 three-step helical reduction gear reduces output shaft speed to a normal engine range.

The weight of the original engine buildup is 850

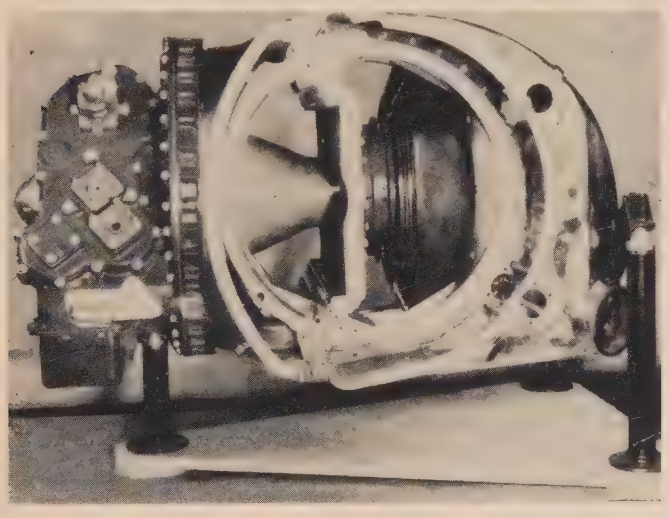

Fig. 14—Assembly of gasifier and power section

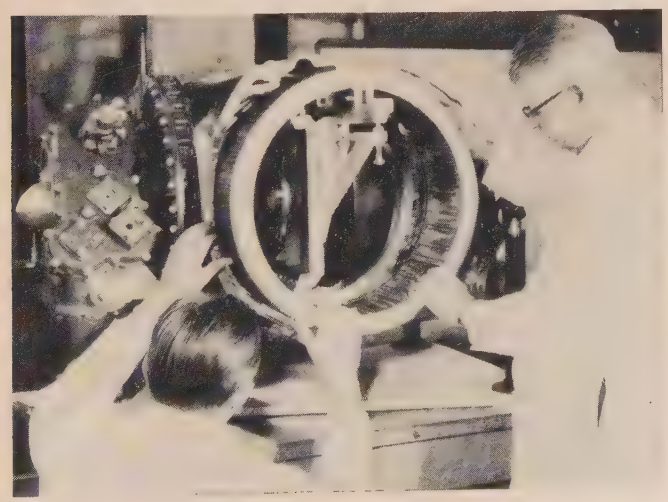

Fig. 15—Regenerator installation

Fig. 16—Combustion chambers

Fig. 17—GT-304 engine assembly

Fig. 18—Firebird II in which the GT-304 engine is installed

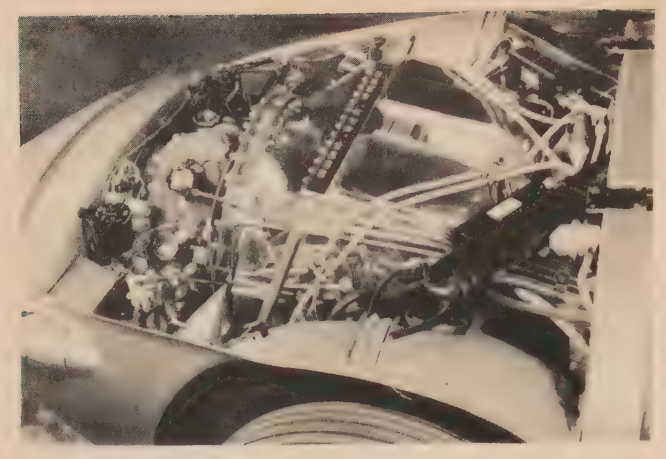

Fig. 19—Engine installation

lb. This weight can be reduced by further design refinements of the engine structure and components. The regenerators presently weigh approximately ¾ lb per hp.

Vehicle Installation

The Firebird II, in which the GT-304 engine is installed, is shown in Fig. 18. This installation incorporates several unusual features. The engine is mounted forward in the conventional location but the transmission is located behind the passenger compartment in line with the rear wheels. A rubber-mounted 3-joint driveshaft connects the engine output shaft to the transmission input. The larger power-absorbing accessories are mounted on the transmission and driven from the input shaft. An acoustic intake silencer is mounted forward of the engine and is an integral part of the body. A dual range exhaust system is provided with a total of four outlets.

The engine installation is shown in Fig. 19. Three mounting points are provided for the engine, one below the accessory section, and two at the rear of the reduction gear case. The low lines of the car leave only limited space for accessory, instrumentation, and control linkage installation. Heater and air conditioning components are mounted on the bulkhead above and behind the engine. The ignition packages are mounted on the diagonal frame members. Engine oil coolers, mounted on the rear of the front engine compartment bulkhead, are cooled by fan-induced airflow. The fans, one of which can be seen at the lower left, discharge rearward along the sides of the engine for compartment ventilation. Just above the fan outlet is the fuel pump and governor assembly and the other gasifier-driven accessories. Retractable headlight assemblies can also be seen on each side at the front of the engine space.

In the earlier Whirlfire installations, all accessories were driven from the gasifier turbine. However, design calculations showed that the accessory loads for the Firebird II installation would overload the gasifier turbine at idle. Therefore, it was advantageous to drive some of the larger accessories from the power turbine. Because of space limitations in the engine compartment, these units were mounted on the transmission, which was placed behind the passenger space and in line with the rear wheels. A view of the transmission and accessory installation is shown in Fig. 20. The Delco a-c generator can be seen to the left of the propeller shaft. Above it and slightly to the right is the rectifier and further to the right is the current regulator. The two batteries are mounted at the top of the compartment on each side. To the right of the propeller shaft is the airconditioning compressor and directly above is the pump and reservoir for the central hydraulic system. In this view the transmission itself is almost completely hidden by the attached components.

The transmission was designed by the Transmission Development Section of our Engineering Staff. This unit is frame-mounted and incorporates the differential for the swing axles. Electric solenoids permit driver selection of park, neutral, reverse, and drive. The planetary-type transmission provides four automatic gear ranges in the drive position. A further feature of the transmission is the fluid input coupling which is a distinct departure from the direct-connected transmissions used with the previous Whirlfire engine installations. Thus, the power turbine turns at all times to drive the transmission-mounted accessories. "Softness" of the coupling is also an assist during shifting when energy of the rotating mass must be absorbed to prevent roughness.

Ducts for the engine exhaust system can also be seen at each side of the chassis in Fig. 20. These ducts run from the exhaust housings on each side of the engine, rearward through the chassis sill members and upward to vents on each side just behind the passenger-compartment canopy. Actually these are the engine idling exhaust vents and are augmented under road load conditions by two additional vents directly under the engine. These augmentor vents are opened by hydraulically actuated shutters when the accelerator pedal is depressed. The resulting decrease in engine back pressure is very desirable from a performance and economy standpoint.

Operating Experience

A limited amount of operating experience has been accumulated since the first model of the GT-304 engine was completed. This includes test cell operation as well as running experience in the Firebird II. The close agreement of engine performance with design calculations has been very encouraging. Those problems which have come up are of such a nature that they can be resolved with our present engineering "know-how."

The calculated performance data shown in Fig. 21 agree closely with test results obtained to date. Leakage of the early regenerator seals was above the

design limit and resulted in higher turbine inlet temperatures. This limited maximum output to 160 hp, or 80% of design power. The latest seal design reduces this leakage to a value well below the initial design goal.

One of the most surprising features of the GT-304 engine is the very low noise level. This is noticeable in the test cell as well as in the Firebird II installation. The regenerators are undoubtedy responsible for much of this improvement. The plenum-type design with low air velocities, as well as the attenuating effect of the regenerator matrices, act as sound barriers to muffle the turbine exhaust noise. A very effective inlet silencer further contributes to quiet operation of the Firebird II.

Quieting the turbines and compressor has unmasked another source of noise, however. The reduction gears create the predominant sound of the Firebird II installation. Ground, helical gears were used in an effort to avoid this problem, but it is obvious that other means must be employed. Divided loading and lower pitch-line velocities make planetary gearing very attractive for this job.

A very distressing condition was discovered when the engine was first operated with an open exhaust system. The combustion system, which had been developed for low carbon formation, produced "aldehydes" that were more acrid than any diesel-engine exhaust. This was apparently a combustion phenomenon like carbon or smoke formation but, unlike these, could not be visually detected. Spray pattern and liner design proved to be the means of attacking this problem. The odor is barely noticeable now and with further development can be completely eliminated.

Performance of the GT-304 in the Firebird II shows significant improvement over the Firebird I and Turbocruiser installations. The engine noise level is greatly reduced, and can be further improved with better reduction gearing. The fuel consumption has been reduced by at least 50%. A full evaluation of this factor is still dependent on further tests. Acceleration performance is greatly improved over the earlier engine designs. The acceleration delay is notably less, and low-speed acceleration is enhanced by the 4-speed transmission. Thus, it appears that the fundamental design objectives of the GT-304 engine, reduced fuel consumption and improved low-speed performance, have been accomplished. Full evaluation of the engine must await more comprehensive test cell and proving ground work.

Future Program

Development of the GT-304 engine will proceed along three general lines: thermodynamic improvements, increased durability of components, and a reduction in the amount of critical material required for fabrication. It is not likely that components such as the turbines and compressors are at their highest state of development, and it certainly should be possible to improve both effectiveness and pressure loss of the regenerators from the first design incorporated in the engine. Many parts of the engine are fabricated from better materials than are actually required. This is partly due to lack of information and partly due to the fact that material cost is soon obscured in hand-made prototypes. Test engines can now be instrumented and operating conditions determined to help in more logical materials selection. The goal of this program is an engine weighing less than 3 lb per hp, having thermal efficiency approaching that of the reciprocating engine, and containing less than 10 lb of nickel or other critical material. Our work to date does not show any reason to believe this cannot be accomplished.

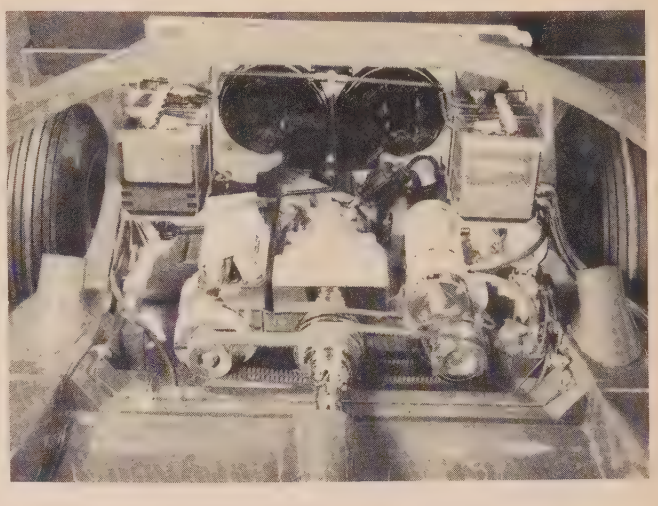

Fig. 20—Transmission and accessories installation

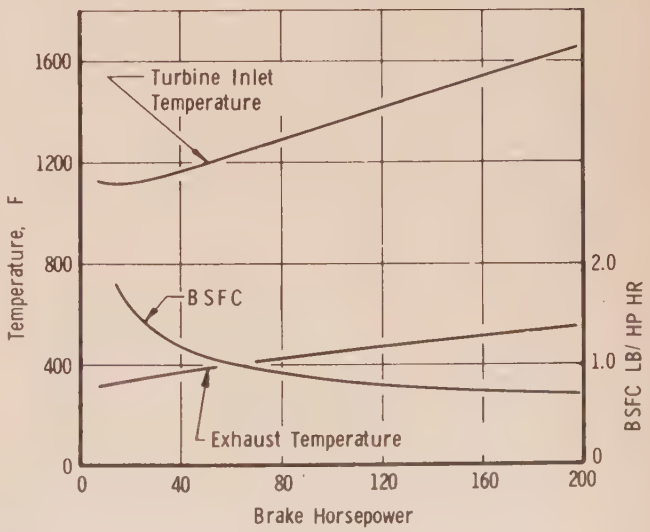

Fig. 21—Typical performance characteristics of GT-304 Whirlfire engine

HUMAN FACTORS OF CRASH

THE number of military personnel killed and injured each year in ground-vehicle accidents is a constant source of concern for the Armed Forces as it represents a needless waste of man power to the defense effort. In the Air Force, motor vehicle accidents have been the leading cause of death, injury, and noneffectiveness.[1] The majority of these men injured and killed were in private vehicles off-base, and most of the accidents occurred during the week ends. This can be attributed to the fact that most bases rigidly enforce a relatively low speed limit for vehicles on base, whereas the off-base speed limits are determined by the particular state where the base is located. The on-base accident then is more apt to occur at lower speeds than those off-base which would tend to reduce the severity of the accident.

Whether an airman is killed or injured off-base or on-base in a private vehicle or government vehicle, his services and training are nevertheless lost to the Air Force. Driver education will help somewhat to prevent accidents, but concrete steps should be taken to protect the vehicle occupant when the accident occurs.

Vehicles have shown constant improvements in design and mechanical functioning, but their relative crash-protection characteristics have not increased along with the other improvements.[2] Since a vehicle can be considered a package to deliver specific goods to a destination, more attention should be focused on the design of the package from the safety aspect of the passenger in crash situations. With force-attenuating structures incorporated in vehicles, injury severity will be reduced with a resultant reduction of man days lost to the service due to recuperation from these accident injuries.

The primary factors to consider in protecting vehicle occupants from crash forces are the limits of human tolerance to these forces. The parameters of human tolerance to experimental application of crash-

THE problem of crash protection for ground-vehicle occupants relates not only to the general public but involves the Armed Forces inasmuch as there is a tremendous loss of man power to the defense effort from motor vehicle accident injuries.

Evidence for the need of safety devices in ground vehicles in order to improve their crash-protection characteristics is recommended.

Specifications for lap belts and lap belt installations are described, and the use of these belts in priority vehicles to evaluate their effectiveness is recommended. The three fundamental harness configurations are briefly discussed.

[1] "Human Variables in Motor Vehicle Accidents, Review of Literature," by R. A. McFarland, R. C. Moore, and A. B. Warren. Pub. by Harvard School of Public Health, Boston, 1955.

[2] "Study of Crash-Injury Patterns as Related to Two Periods of Vehicular Design," by J. O. Moore, B. Tourin, and J. W. Garret. Pub. by Cornell University Medical College, Automotive Crash-Injury Research, Department of Public Health and Preventive Medicine, New York, March, 1955.

PROTECTION IN AUTOMOBILES

John P. Stapp and Sidney T. Lewis,
Holloman Air Development Center

This paper was presented at the SAE Summer Meeting, Atlantic City, June 4, 1956.

type mechanical force have been determined with respect to rate of onset, magnitude, duration, and direction.[3,4] These limitations can provide a basis for designing adequate enclosures and restraints for occupants of vehicles. The problem then remains to determine the forces encountered in automobile collisions and methods for attenuating these forces so that they will be within the human tolerance limits.

The forces generated in automobile collisions will depend on a number of factors. A few of these are speed at impact; type of impact, whether rear-end, head-on, broadside, or roll-over; type of vehicle or vehicles involved in the collision; and object impacted. Severy and Mathewson [5] have investigated forces encountered in rear-end collisions and collisions with fixed objects. For the impact conditions, they give deceleration patterns, frame deformation, and automobile impact analysis. This work indicates that forces will have to be attenuated in order to reduce injuries to vehicle occupants.

There are many accident configurations which can cause serious injury to occupants of the vehicles involved. The most serious accident configurations for USAF motor vehicles by order of precedence [6] are first, those vehicles impacting objects other than another vehicle, that is, an impact with a fixed object. This type of accident configuration has been the cause of more death and injury than any other type. The second most serious configuration is that in which a vehicle overturns or rolls over. This is classified broadly as the roll-over type of accident in which occupants of open-top vehicles are placed in an extremely hazardous condition. The third type of accident configuration is the head-on collision in which there are two vehicles involved in the accident. In this type of accident the ratio of drivers killed to those receiving temporary injuries is one to four, whereas the ratio of passengers killed is one to seventeen.[6] This indicates that there is a higher force concentration on the drivers as compared to passengers. This is due possibly to the fact that the driver is positioned such that the steering wheel and column can be struck with such force as to cause fatal injury, whereas the passenger is less apt to strike such an imposing force-concentrating object. By installing restraint systems in vehicles, the driver can be protected against striking the steering wheel with as great a force as when unprotected, and passengers would be prevented from being thrown about within the vehicle.

Studies have been conducted to correlate passenger injury to impact areas within the car and to accident configuration.[2,7,8] From these studies it is apparent that

3 *Journal of Aviation Medicine*, Vol. 26, August, 1955, pp. 268–288: "Effects of Mechanical Force on Living Tissues, Part I: Abrupt Deceleration and Windblast," by J. P. Stapp.

4 Air Force Technical Report No. 5915, WADC, Dayton, Ohio, December, 1951, "Human Exposures to Linear Deceleration, Part II: Forward-Facing Position and Development of Crash Harness."

5 "Technical Findings from Automobile Impact Studies," by D. M. Severy and J. H. Mathewson. This is an unpublished report.

6 Reports, Analyses, and Survey Division, Assistant for Ground Safety, DCS/P, Headquarters USAF, "USAF and Private Motor Vehicle Ground Accident Experience, Driver and Passenger Crash Injury Analysis, January through June, 1955."

7 "Annual Report for Period April 1, 1955 to March 31, 1956," by J. O. Moore, Director of Division. Pub. by Cornell University Medical College, Automotive Crash-Injury Research, Department of Public Health and Preventive Medicine, New York, 1956.

8 "Study of Automobile Doors Opening under Crash Conditions," by J. O. Moore and B. Tourin. Pub. by Cornell University Medical College, Automotive Crash-Injury Research, Department of Public Health and Preventive Medicine, New York, August, 1954.

some types of protective devices are necessary to decrease the injury-producing characteristics of vehicles. The automobile manufacturers are becoming more cognizant of the fact that some steps must be taken to protect the occupants of their product when a crash situation occurs. One of the principal automobile manufacturers has already incorporated padded dashboards, seat belts, safety door latches, and energy-absorbing steering wheels as devices for reducing injuries to passengers in accident conditions. The studies conducted and the subsequent incorporation of safety devices can be indications to ways of increasing the crash protective characteristics of Armed Forces ground vehicles. This can be accomplished by modifying the vehicle itself and by adopting restraint systems for the vehicle occupants.

Ground-Vehicle Modification

Moore and Tourin [8] have shown that under impact conditions, ejection from the vehicle doubles the risk of injury. These ejections were primarily due to the doors opening upon impact and the occupant then being thrown from the vehicle. One method of preventing these ejections is to strengthen door latches so that they will not be forced open upon impact. This will not only prevent the ejection of the occupants and reduce the risk of injury, but will also strengthen the occupant compartment. Ground vehicles should be equipped with safety door latches so as to prevent ejection of occupants in accident situations.

Many injuries to occupants are caused by the occupant being thrown against various objects within the vehicle. Some of these lethal objects are instrument panels, steering components, windshield frame and glass, rear-view mirrors, emergency hand brake, and door components.[2] Interiors of vehicles should be delethalized in order to increase their crash-protective characteristics with specific attention given to any object such as the above-listed items which could cause injury if impacted by the occupant.

Open-top vehicles should have retrofitted roll-over structures to protect occupants in roll-over-type accidents. These structures would prevent the occupants from being crushed by the vehicle when overturned. For Armed Forces vehicles, the device should be retrofitted so as to be easily removed in instances where the structures would interfere with the combat operation of the vehicle. This device would have to support the weight of the vehicle in the overturned position with an additional safety factor incorporated to withstand the forces if the vehicle overturns at speeds up to 40 mph. Such a device is being designed and constructed by J. J. Ryan at the University of Minnesota under Air Force Contract AF 29(600)–831 which is monitored by the Aero Medical Field Laboratory, Holloman Air Development Center.

Bumpers should be utilized that will adequately attenuate the crash forces so that the force transmitted to the occupant compartment will be within the human tolerance limits. Under the same contract previously mentioned, Professor Ryan is designing a hydraulic and spring-mounted bumper that will meet the above specification. These bumpers will be retrofitted so that a minimum of time and material will be necessary to afford their immediate installation after the device has been accepted.

Dials and hand controls should be arranged for maximum efficiency of operation so that motor vehicle operators would not have to divert their attention from the control of the vehicle to check these instruments. In this manner accidents can be prevented which are caused by time delays in operating a control due to poor positioning and design.[9] This arrangement should be such that the injury-producing potentiality of the instruments would be reduced in instances when an accident does occur.

Tracks and locks on movable seats should be stressed for a minimum force of 5000 lb. This will reduce the possibility of seats tearing loose from their anchorage points and adding to the impact weight of the occupant. This strengthened installation will serve as a better foundation for seat belts, so that eventually belts can be anchored directly to the seats without additional material required for linkage to a stronger member in the vehicle.

Steering wheels should be collapsible or present broad flat surfaces parallel to the thorax so that upon impact the force will be distributed over a larger area rather than concentrated in one location. This would help to reduce injuries to drivers who are secured by lap belts and who are impinged on the steering wheel. Under actual tests, a steering wheel with ductile spokes which bend under a force of 1000 lb or more through a distance of about 6 in. produced less injury than the rigid-spoked standard wheel.[10] The injuries sustained with the standard wheel were caused by impacting the steering column and concentrating the force in one area, whereas the energy-absorbing wheel prevented the subject from impacting the steering column and distributed the force over a larger area.

All structures that can be impacted by the head when an individual is secured by lap belt should be covered with energy-absorbing foam plastic. This material should be of such quality that it would not lose its energy-absorbing characteristics upon long exposure to environmental conditions within the car or upon repeated impacts. By using such a material the severity of head injuries to occupants on impact with these structures would be reduced.

Headrest structures should be incorporated on seat backs to reduce the possibility of whiplash injury to the neck. This would be beneficial in rear-end-type collisions when the neck is severely snapped or in situations where occupants are restrained by lap belt and shoulder harness.

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[9] "Human Factors in Highway Transportation Safety," by R. A. McFarland and A. L. Moseley. Pub. by Harvard School of Public Health, Boston, 1954.

[10] "Test Report on Biodynamics of Human Factors in Aviation," by S. T. Lewis. Pub. by Holloman Air Development Center, New Mexico, August, 1955.

Lap belts should be incorporated to prevent occupants from being ejected from the vehicle or thrown violently against objects within the automobile. During the manufacturing process, adequate seat belt fastening should be constructed in the vehicle and included in the basic cost, but leaving the actual seat belt itself optional.

Lap Belt Specifications

The following specifications are recommended for lap belt installation in Armed Forces ground vehicles.

The webbing utilized for the fabrication of the belts should be either nylon or dacron. The nylon webbing specifications are MIL-W-4088B which are as follows:

1. Nylon warp, nylon filling.
2. Type VII.
3. Double plain weave.
4. Breaking strength of 5000 lb.
5. Abrasive treated with latex rather than resin.
6. Width of 1¾ in.

By using this webbing the installation of the belts can be expedited since it has already been field tested and would eliminate an extensive testing program which would have to be conducted on any other type material. This webbing is being produced and can be easily procured. Any difference in cost between this webbing and other materials is more than offset by its long wearing characteristics. This would eliminate the replacing of the webbing due to fabric fatigue as often as compared to a cheaper material.

The dacron webbing specifications are MIL-W-25361 which are:

1. Dacron warp, nylon filling.
2. Type II.
3. Breaking strength of 5500 lb.
4. Abrasive treated with latex.
5. Width of 1¾ in.

This webbing has been most recently adopted by the Air Force as standard for aircraft.

Buckles, hardware, and attachments should be stressed for 6000 lb of force. The buckles should be metal-to-metal fitted with a positive cam-type lock capable of being opened with either hand. It should weigh no more than ¾ lb and be constructed of stainless steel or equal. There should be friction-type link adjustments on either side of the buckle positioned such that there will be no pull applied to any stitches. The hardware connections which the link the webbing to the anchorage point within the vehicle should be those utilized in aircraft shoulder harnesses. These are classified as link connectings for shoulder harnesses with specifications as per Air Force drawing number 49B7414.

The belts should be anchored by the following methods:

1. Attach the belts to an adequately stressed metal angle piece which is anchored to the back of the seat with 10,000-lb strength aircraft control cable linkages continuing to the frame through holes in the floor and positioned so as to give minimum interference with floor foot-space in the rear compartment of the vehicle.
2. Reinforce the floor panel with a metal bar and attach the link connectings to the metal bar.

In both of these installations the seat must be secured so as to prevent tearing loose on impact. This can be done by using aircraft cable as a linking between the seat and frame.

Recommended Application

Seat belts should be eventually installed in all Armed Forces ground vehicles, but for the present time they should be installed in the following type vehicles:

1. Emergency vehicles authorized to operate in excess of usual speeds, such as ambulances, fire trucks, air police vehicles, and crash vehicles.
2. Vehicles authorized for off-base operation at highway speeds such as staff cars, pickups, carryalls, and panel trucks.
3. Vehicles where occupants may be thrown about and lose control, such as earthmoving equipment, tugs, and tractors.

Seat belts are not recommended for open-top vehicles of the weapons carrier and jeep class until adequate roll-over structures are provided which will protect occupants of these vehicles from being crushed in roll-over-type accidents.

In regard to privately owned vehicles, it is recommended that experimental installations of belts and hardware at government expense in automobiles of all Ground Safety Officers be evaluated with quarterly reports from these officers.

Summary

Because of the number of military personnel killed and injured each year in ground-vehicle accidents, the problem of crash protection for vehicle occupants has become an integral part of the Air Force research program. By enhancing the crash-protective characteristics of vehicles, we are protecting the service man against serious injury or possible death with a resultant reduction of man power lost to the defense effort. In order to accomplish this task, there are two basic areas to consider. One is the modification of the vehicle and the other is the development of restraint systems.

From the evidence available on the statistical evaluation of crashes, it is apparent that design modifications are necessary in order to obtain an equilibrium between appearance and safety. The interior of a vehicle must be delethalized in order to assure the protection of the occupant should the accident occur. This delethalization can be accomplished so that the appearance of the vehicle will be acceptable

to the public, yet the protection is available when the situation arises where it is necessary.

It is generally agreed that some type of restraint system should be available to the vehicle occupant to protect against the crash forces which are generated. The problem arises as to what specifications and configuration should be adopted. The three fundamental configurations of restraint are the lap belt, chest belt, and shoulder loop. The argument for the chest belt and shoulder loop is that they will afford the best protection as compared to the lap belt by restraining the vital parts of the body from impacting the interior of the vehicle. These two configurations permit the knees and legs to be thrown into the dashboard thus increasing the possibility of joint injury. Joint injuries are more permanent as compared to slight concussions. These two types of restraints will interfere with the normal driving movements required for control of the vehicle and with the comfort of the passengers. For this reason, there will be a low degree of acceptability by the public which is an important factor to consider in adopting a configuration. These particular configurations are not suitable remedies for whiplash injuries since a headrest must be incorporated on seat backs so as to provide support for the head when crash forces are applied.

The lap belt configuration has been recommended for use in military vehicles as it restrains a larger mass of the body from moving due to the fact that it is nearer the center of gravity of the body. It does not restrict driving movements or interfere with passenger comfort which makes this configuration more acceptable. It will not completely prevent head impacts but will lessen the severity of the impacts. The only appropriate remedy for eliminating head impacts is adequate clearance for seat occupants. There is adequate clearance for rear-seat passengers whereas front-seat passengers are apt to impact the instrument panel or windshield.

More attention must be focused by the engineer on the design of vehicles from the safety viewpoint. Safety devices must be incorporated which will have a high degree of acceptability by the public so that these devices will be constantly utilized. In this manner the number of those killed and injured on our highways each year will be greatly reduced.

ORAL DISCUSSION

Reported by J. G. Haviland

General Motors Corp.

R. McLean, GMC: In the seat belt attaching method, can the horizontal seat adjustment be used as usually provided in passenger vehicles?

Col. Stapp: Yes, enough slack in the cable can be provided, but army vehicles have few seat adjusters, thus minimizing this problem.

R. H. Fredericks, Ford Motor Co.: Have you run lap belt tests which give indications of whiplash injuries in a forward-type collision stop?

Col. Stapp: The Institute of Transportation and Traffic Engineering, University of California, has published informative data on this subject.

T. S. Gillick, Jr., American Felt Co.: Assuming a 46.2-*g* impact force, how much is this reduced with a lap belt with respect to the head hitting the dash panel?

Col. Stapp: A deceleration of 46.2 *g* on a car frame would be high, but the passenger would have considerably less. An actual 70-*g* deceleration force on the frame gave a 35-*g* deceleration to the properly strapped-in passenger. In a case where the crash is foreseen, the strapped-in occupant should sit erect, brace against the wheel (although a place to grip underneath the wheel would be better), and lean his head forward.

D. J. Schrum, Studebaker-Packard Corp.: At what maximum speeds of impact can crash-padding material be expected to be of value?

Col. Stapp: Energy-absorbing pads will absorb about 50% of the force with which the head strikes. We believe, if properly cushioned, the head will take 150 *g* for not over 0.15 sec, but simulated head tests indicate 220-*g* deceleration would be lethal. On the other hand, we have found 25-*g* forces can be fatal if impacting small areas such as the end of the control stick or half of a steering wheel.

A. Haynes, Ford Motor Co.: Do the specifications requiring 6000 lb for seat belts mean static load? It seems that the present lower specifications belts are working at much higher loads dynamically.

Col. Stapp: The requirement of 6000-lb tensile strength for seat belts is static load and is certainly sufficient for automotive use. However, since the Air Force has large quantities of this webbing available, there is little point in reducing the requirement.

G. Hannes, Libbey-Owens-Ford Glass Co.: Do you feel you are better off without a shoulder harness when you do not have a headrest or support?

Col. Stapp: Not having any data on effect of crashes with a shoulder harness without a headrest, we cannot answer this question. However, for a really effective installation, one should have all three, the seat belt, the shoulder strap, and the seat back headrest.

R. Brodie, American Cord and Webbing Co.: What is your opinion of the commercial auto lap belt now being installed? Would the best of these have your approval for armed forces installation?

Col. Stapp: The armed forces have the specifications, but belt manufacturers will have to evaluate their belts to see how they stand in relation to the specifications themselves.

J. Gordon, Gordon Chapman Co.: How effective is reverse seating found to be in both aircraft and group vehicles?

Col. Stapp: The British have had some success in this. In one aircraft crash, the crew were killed, and the rearward-facing passengers were unharmed, except for one passenger whose leg was injured by a boulder coming up through the floor. In automobiles, rearward-facing passenger seats should also be good, but they open psychological problems.

R. Watts, Detroit News: Do you still plan to make runs through the sonic barriers?

Col. Stapp: We do plan more runs with the sled, but time of making and speed of runs planned are, at present, classified information.

Motion Sensitivity as a Guide to Road Design

William A. McConnell, Ford Motor Co.

This paper was presented at the SAE Summer Meeting, Atlantic City, June 5, 1956.

SHORTLY after the first automobiles were built, suspensions were employed, not so much for comfort in the beginning as for survival—to keep the passengers from being thrown out or maimed. As cars and road surfaces improved, we became more critical of ride, and our standards of judgment changed. We sought the comfortable instead of the tolerable, then luxury in preference to comfort. Today, ride engineers are a special breed, pursuing the diminishing return, using instruments and apparatus to measure motions so slight and differences so subtle that their own physique can no longer register them with sufficient accuracy or objectivity. Quantitative riding comfort and vibration limits have been formulated which are useful design tools.

HUMAN perception thresholds to motions and the various characteristics of motions in six degrees of freedom are presented.

Experiments made to establish these thresholds for disturbances of the duration or frequency that might arise from highway geometry—durations in the range of from one second to several hundred seconds—are described.

Applications of motion sensitivity criteria in the design of vertical and horizontal highway curves and transitions are developed, with examples from company proving grounds in Arizona and Michigan. The resulting geometric features of the high-speed road systems differ in several important respects from conventional highway practice and previous automotive test-track designs.

Highway engineering has evolved along much the same lines. At first it was enough to clear the trees and scrape a path. Then surfaces were smoothed, summits shaved, valleys filled, and alignment eased to speed the passage. Today highway technology has progressed to where standards are defined for gradients, curvature, and other features. With the increased understanding of the dynamic aspects of highway geometry, it is no longer a question of whether transitions, superelevations, vertical curves, and easements should be used, but rather what are the best forms and what design criteria can be used.

In designing high-speed test road systems for an automotive proving ground, these questions become especially critical. It is important in test work to be able to operate under stabilized conditions at all possible vehicle speeds. What we really want is a straight road, indefinitely long; yet this must be built in a limited space with curves sharper than any highway designer would consider feasible and for speeds up to twice what might ordinarily be considered reasonable and proper. We would like these curves designed in such a way that neither the car nor its occupants can tell that the vehicle is in a turn and not on a straight level road. There should be no steering drag, no undue tire wear, no perceptible motions, forces, or changes in force above those which would be experienced on a straight level road.

It is probably natural that automobile engineers faced with a highway design problem of this sort would consider highway geometry problems as an extension of the ride problem. Dips and turns are just big bumps in the road. Traveling across mountains and valleys is just a vibration—although a very large amplitude low-frequency one. If our suspensions can't smooth such bumps, maybe the bumps themselves—since they are intentional ones—can be shaped to

Table 1—Human Perception Thresholds for Possible Modes of Motion[a]

Degree of Freedom	Motion Characteristic: Position x	Velocity $\frac{dx}{dt}$	Acceleration $\frac{d^2x}{dt^2}$	Rate of Change of Acceleration $\frac{d^3x}{dt^3}$
Translational				
Longitudinal	Not sensed	Not sensed	±1.0 ft/sec^2	±0.5 ft/sec^3
Lateral	Not sensed	Not sensed	±0.6 ft/sec^2	±0.3 ft/sec^3
Vertical	Not sensed	Not sensed	±4.0 ft/sec^2	±0.8 ft/sec^3
Rotational				
Yaw	Not sensed	± 5 deg/sec	±2 deg/sec^2	±1 deg/sec^3
Roll	± 1.1 deg[b]	± 8 deg/sec	±4 deg/sec^2	±2 deg/sec^3
Pitch	± 1.9 deg[b]	±12 deg/sec	±6 deg/sec^2	±3 deg/sec^3

[a] Conditions of Measurement
1. Subjects seated in standard automobile seats.
2. Subjects blindfolded; kinesthetic and labyrinthine cues only.

[b] Roll and pitch position sensitivity due to lateral and longitudinal components of gravity force. Not sensed when balanced by lateral or longitudinal accelerations.

give the ride motions we prefer. Possibly comfort criteria established for smaller highway faults, irregularities, and roughness can be extrapolated to govern the character of the slower motions and disturbances designed into the highway.

We were confident we could develop a surface shape to produce any desired redirection of a vehicle, within any given limits to the velocities, accelerations, or rates of change involved. Analysis of existing high-speed road designs showed several approaches. Most were aimed at neutralizing lateral forces; some were concerned with vertical accelerations and rates; and others considered roll motions as well. There were some published data showing how particular motion characteristics could be minimized to make best use of a given space. But a superelevated spiral curve involves a complex motion in six degrees of freedom. It is not clear which are the critical motions, or what the relationship between them should be to achieve a balanced design within a given space; nor could we find out how much of this given space was needed. A rational determination of curve dimensions and straight-to-curve transition geometry, therefore, would seem to start with an evaluation of human sensitivity to all the perceptible characteristics of motions for the six possible degrees of freedom and with the establishment of suitable working limits to the magnitude of these characteristics.

Perception Threshold Data

Two different parameters for measuring body sensitivity are commonly used. One employs a scale with different degrees of comfort from luxury through varying stages of disturbance to pain. The other ranges from perception through various degrees of injury to death. Note that the first, or psychological scale, calls for an opinion or conclusion on the part of the witness; the second, or physiological evaluation which we prefer, does not. As engineers, we preferred to think of our sensory mechanisms as instruments: accelerometers, both angular and linear, in the inner ear; some force, pressure, or load cells in the skin; and some stress and strain gages in the muscles. Since people come in only two models, we felt we would find better consistency in determining the limiting sensitivity of their instruments, that is, the point at which awareness of motion first occurred, than we would in establishing some vague value of exposure which, in the group opinion, was judged comfortable. We also decided to ignore visual stimuli, since it didn't appear feasible on a highway to eliminate the impression of motion gained by watching the changing landscape.

Our first search for perception threshold data was disconcerting. Our human instruments did not have a very flat frequency response. There are apparently many distinct types of neural receptors, each tuned to its own range and variety of stimulation—seven specific kinds of nerve endings in the skin alone, for example. These receptors can also adapt to certain levels of stimuli under continuous nonchanging excitation, and under these conditions cease to give their characteristic sensation. In this respect, they are much like some of our test instruments which will measure strain or acceleration under dynamic conditions, but won't respond at all to steady or low-frequency signals. A force change on the body equivalent to 40 g's which crash people say will produce "irreversible injury" (that is, death) if sustained for a few milliseconds, can be endured with no especial harm if developed gradually over a long time, for instance 20 years, as any 5-lb baby who is now a 200-lb man can testify. It was also apparent that the skin and kinesthetic or muscle sensations would be very much influenced by the manner of support of the body. Seated, the unit pressures and muscular tensions are less, and we can withstand forces and accelerations which we couldn't take standing up.

The table of thresholds shown in Table 1 began with a correlation of values obtained by other investigators, although these appeared inadequate in many respects. Ride motion analyses had been confined generally to frequencies above 1 cps, and extrapolations to slower motions could be misleading, as there are infinitely more motions differing by a given time or length interval below 1 cps than there are above this frequency. Therefore, the data would have to be checked for motions of the time and amplitude range encountered in highway geometry disturbances. The subjects should be seated in an automobile seat, preferably in an automobile. Particular attention would have to be given to rates of change of acceleration or forces, as the first perception of these slow motions would probably come from detection of a change even though the level of the force stimulation itself might be unrecognized.

The procedure for checking the tentative table of thresholds was quite simple. We simply drove as many different examples of highway curves and transitions as were available to us, gradually increasing the speeds until a blindfolded passenger perceived that he was changing direction. From the known geometry of these road features, we could calculate the veloci-

ties, accelerations, and rates of change of acceleration (hereafter referred to as "jerk," for want of a better name) which were present at the critical speed for each of the six modes of motion. Our table was revised whenever an imperceptible maneuver showed a particular characteristic with a higher value than that already tabulated. We reasoned that if no characteristic on a new design exceeded the values reached without perception on other existing road features, the new design likewise should be imperceptible.

Among the road facilities used for this purpose were some superelevated transition curves at the ends of our straightaway at Dearborn, some flat curves of varying radii, a 350-ft radius parabolic section curve or "bowl," and some crowned roads or side slopes, all with precisely known geometry. We have also had the opportunity of making limited observations on some of our competitors' tracks, and on the larger radius and longer transitions of our recently completed Arizona Proving Ground track.

In most cases we were able to obtain further road checks of threshold limits for individual motion characteristics, uncomplicated by cues from motions in other degrees of freedom, either by eliminating these other motions, balancing them, or keeping them below the appropriate threshold. Thus, longitudinal acceleration was checked on a level straightaway. With the perceptible acceleration determined, various rates of change were tried by uniformly increasing and decreasing acceleration without allowing the maximum to approach the value previously established as a force limit. Likewise, lateral acceleration limits were found by driving slowly in a large-radius flat turn until the subjects felt the pull of centrifugal force. The vertical threshold was confirmed by very slowly gathering speed in a large-radius banked-bowl, allowing the car to find its equilibrium path so that lateral forces would be balanced, or until the passenger noticed that he felt heavier.

At slow speeds and large radii, centrifugal forces are relatively large and angular motions relatively negligible. By shifting to the skid pad and operating very slowly in small radius turns, yaw motions were investigated with centrifugal forces reduced to a negligible value. In addition to road tests in automobiles, some checks on angular motions were made in the laboratory with an automobile seat mounted on a teeter-totter for roll and pitch and on a rotary table for turning movements.

Some caution is in order in accepting the values shown in Table 1. To begin with, the number of subjects used in our own checks was unfortunately small. A maximum of 12 people was used in estimating vertical, roll, and yaw thresholds which were of most concern in our special design problem. Even with so few subjects, the scatter was wide, with thresholds reported by different individuals varying by factors as large as two to one. Even on the smoothest roadways, extraneous motions and vibrations of the car produced a masking effect—somewhat like trying to feel the rise and fall of the tide in a rowboat on a choppy sea. A complex motion, with all components below the threshold values might also produce a resultant larger than any of the limits shown, and so might be sensed. However, the manner of support in a car seat makes the sensitivity to vertical motion and pitch substantially less than the sensitivity to lateral or roll motions. There is enough difference between thresholds in the different modes that any resultant will not be significantly larger than the limits shown.

Despite some shortcomings, we feel that these data afford safe design criteria for highway curves, and that further checks will raise rather than lower these limits. We hope that others with more subjects, more time, more facilities, and more ingenuity in devising ways of isolating particular characteristics for measurement will be interested in further developing a table of perception thresholds.

Application to Track Design

The manner of applying motion threshold data to highway design can be illustrated in Fig. 1 by the design of a high-speed test track at our new proving ground in Romeo, Mich. We knew from the type of tests to be performed, that we wanted a track 5 miles long. We had also determined that it should be suitable for speeds up to 140 mph without side force on the tires, and we wanted an oval form with two straight legs as long as possible. The first step was to determine the approximate shape and the radius of the turns.

Fig. 2 is a diagram showing the forces acting on a

Fig. 1—Earthmoving operation at Michigan test track showing shape and radius of the turns

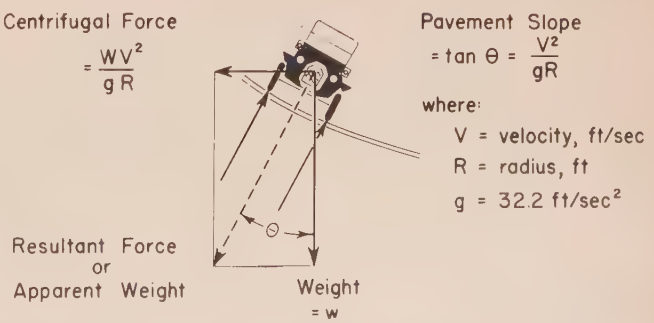


Fig. 2—Forces on vehicle in superelevated curve at equilibrium or neutral speed

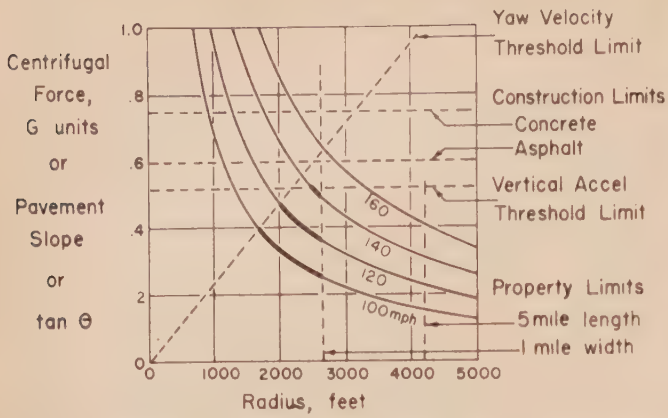


Fig. 3—Limitations on turn radius and bank for various design speeds

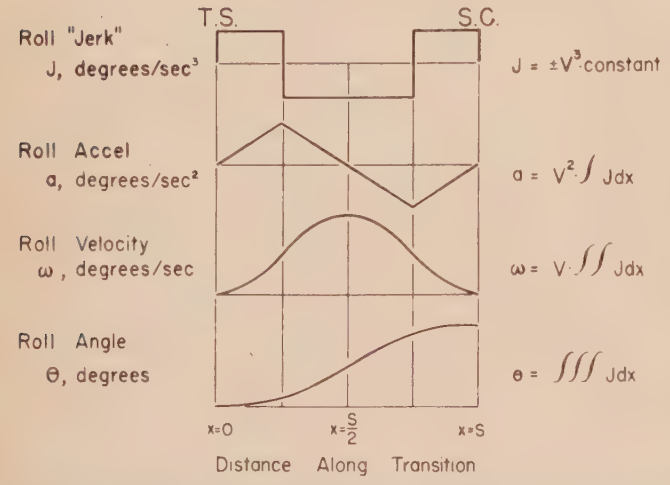


Fig. 4—Roll characteristics of transition designed for minimum roll jerk

car at equilibruim speed in a banked turn. The equilibrium or neutral speed for a banked turn, of course, is that speed where centrifugal force and the vehicle weight combine to produce a resultant force which is perpendicular to the pavement. Wheel reactions on both sides of the car are equal and likewise normal to the road surface. The passenger in such a situation perceives the resultant force as his apparent weight, and has no tendency to slide laterally on the seat. There are no transverse forces between the tire and the pavement. The difference between his true weight and his apparent weight appears to the car occupant as a vertical acceleration. If this is to be imperceptible, it must be kept below the threshold value for vertical acceleration of 4 ft per sec^2; that is, the apparent weight must not be more than about 12½% larger than the acceleration of gravity. This limit is reached with an angle of about 28 deg, or a pavement slope equal to the tangent of this angle, or 0.52. Some of the restrictions on the choice of turn radius are shown in Fig. 3, which is a plot of the pavement slope required for equilibrium as a function of curve radius

Fig. 5—Construction of transition curve at Ford Dearborn tes area

for four different speeds. The slope limit of 0.52 required to keep vertical acceleration below perception is the governing limit on slope. The construction limits or maximum practical slopes which can be formed with asphalt or concrete are not approached. A yaw velocity threshold of 5 deg per sec also limits the choice of radius. This angular velocity is reached at 100 mph in a 1700-ft radius, or at 160 mph in a 2700-ft radius. Everything to the left of a diagonal line defining the yaw velocity limit involves a turning velocity which will be perceptible. For upper limits on radius we have a radius of 4200 ft, which is the most we can get in a 5-mile circumference. A lower limit was fixed in our case at 2600 ft in order to keep within certain geographical limits. For a 100-mph track, our radius could be between 1700 and 2600 ft. For the 140-mph design speed, we have the choice of any radius between 2500 and 2600 ft. A 160-mph track with imperceptible curves can't be built in the space available. A radius of 2500 ft was chosen for our 140-mph tracks at Romeo and Arizona.

Transition-Spiral Design

The next step was to determine the shape of the transition spiral to connect the straight and circular portions of the 5-mile oval. Several test roads have been built recently, including a large high-speed automotive test track similar to the one we planned, which have used as a basis for laying out the transition curves a mathematical form known as Cornu's spiral.

The pertinent property of the Cornu spiral is that the radius varies inversely as the distance along the spiral. The significance of this property in a highway transition is that the lateral acceleration, or centrifugal force, will increase at a constant uniform rate for a vehicle traveling along the spiral path at a constant speed. The lateral "jerk" will be at a minimum. This is a logical form for railroad and highway curves and is the basis for several transition formulas in common use. Highway traffic and railroad trains will seldom travel at the exact neutral speed of a curve, and such design will minimize the lateral movements to which we are especially sensitive. A train

cannot choose different track superelevations appropriate to its speed. Highways, as long as they must carry 2-way traffic and make both left and right turns without low- and high-speed vehicles changing lanes, are restricted to straight sections with constant superelevation slope.

On automotive test tracks, however, curved cross-sections are usually used, so that there is an infinite variety of superelevations for any speed up to the design maximum. In the transitions, the cross-sections are matched to the instantaneous radius at each station. Regardless of the horizontal shape the transition assumes, the vehicle will find an equilibrium path where lateral forces are perfectly balanced. In view of this, it would appear that variation of lateral force and even the amount of lateral force is irrelevant, since this is the one thing we expect to neutralize completely by our test-track superelevation. It will be imperceptible to the vehicle and its occupants.

What cannot be balanced by superelevations, and what may, therefore, become perceptible are the roll acceleration and roll jerk about a longitudinal axis of the vehicle as it passes from the level attitude on the straightaway to the banked attitude on the curve. These were the critical factors which limited the speed for imperceptible operation on every other track we studied—except one which avoided the problem because it was built as a circle without transitions. In the transition design for our new test tracks, we therefore rejected the Cornu spiral, and developed equations to control the roll motions instead.

The method of developing the transition equation is shown in Fig. 4. Curves representing the roll characteristics are drawn such that the acceleration increases at a uniform minimum rate for one-fourth of the distance, decreases at the same uniform rate to a negative peak three-fourths of the way through, and finally increases again at the same rate to zero value. Velocity is found by integrating the roll acceleration curve, and finally, roll angles for various stations along the transition are obtained by integrating the velocity curve. The final bank angle of the car in the circular curve is known, from the design speed and curve radius previously selected. Peak values (threshold limits) are also established for the jerk, acceleration and velocity characteristics. With this information, it is possible to evaluate the integration constants, solve for a length for the transition which will keep the characteristics within the desired limits, and develop equations giving the best roll angle for the vehicle at any distance along the curve. Derivation and solution of these equations for the Ford track transition at Romeo are given in Appendix I.

With roll angles calculated and fixed for each station along the transition, the necessary radius at each station is determined so as to provide a centrifugal force to keep the car in equilibrium on that slope at the design speed. By some straightforward but tedious step-by-step arithmetic, the coordinates of each station and deflection angles for use in the field can be calculated.

Station lengths in the transition and curve are chosen so as to avoid resonance with either the suspension natural frequency (1–2 cps) or the tire hop frequency (10 cps). A 25-ft spacing avoids any resonances in the 35- to 140-mph range and was used in our design. Figs. 5, 6, and 7 show Ford Dearborn transition curves based on this design plan.

Fig. 8 shows the radius of a spiral, determined by

Fig. 6—Ford Dearborn test area showing construction of transition curve

Fig. 7—Ford Dearborn test area transition curve

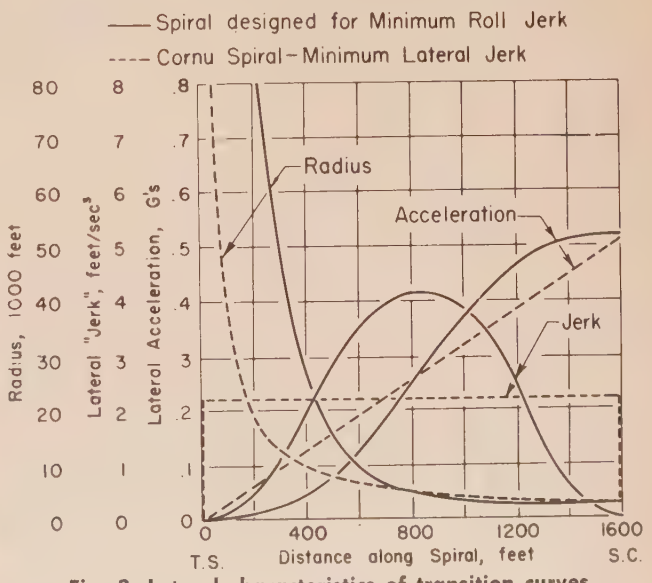


Fig. 8—Lateral characteristics of transition curves

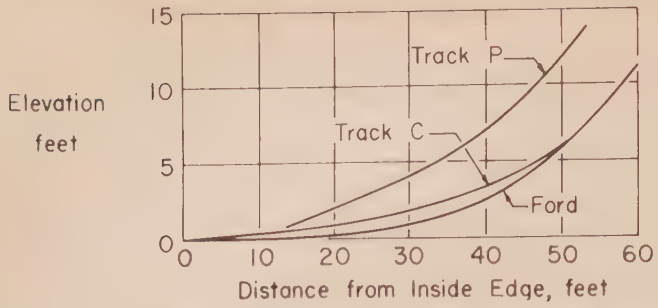


Fig. 9—Cross-section comparison of three high-speed track curves

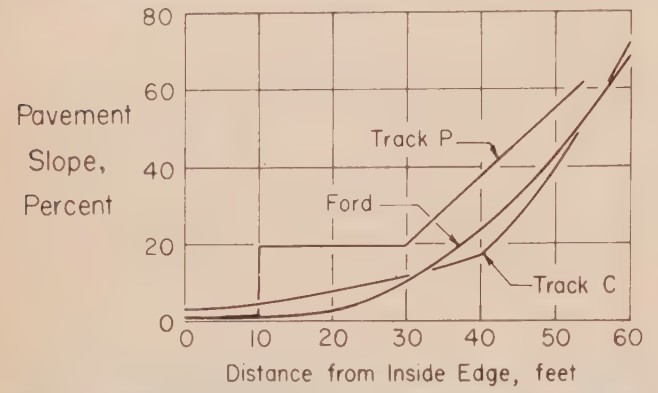


Fig. 10—Pavement slope and width relationship of three high-speed track curves

minimizing roll motions, at various distances along the curve, together with the lateral acceleration and jerk values which would occur on a flat curve of this shape. For comparison, a Cornu spiral is also shown. Note that although the Cornu spiral shows a minimum jerk value, lateral acceleration and jerk are completely balanced by superelevation in both cases and will not be perceptible. Roll characteristics of the Cornu spiral are less desirable, for the roll angle is nearly a linear function of distance along the spiral with the result that roll velocity is nearly constant, but roll acceleration and jerk are theoretically infinite at the beginning and end of the curve. In practice, roll velocity is developed in the space of one station, so that accelerations are finite, but very high, and can be controlled only by keeping the velocity small, and making the transition excessively long.

Cross-Section Selection

The final design step in developing the transition is to calculate cross-sections for each station. An equation is first written for the cross-section in the curve, and the same equation used for all stations in the transition but with the constants changed to give the proper slope at design path as determined from the required roll angle. In this manner, the slope for each lower speed lane or path will be accurately proportioned to the angles of the design path, and the motion characteristics for lower speed vehicles will show the same form but with diminished values. These calculations again are simple but tedious and are given in sample form in Appendix I.

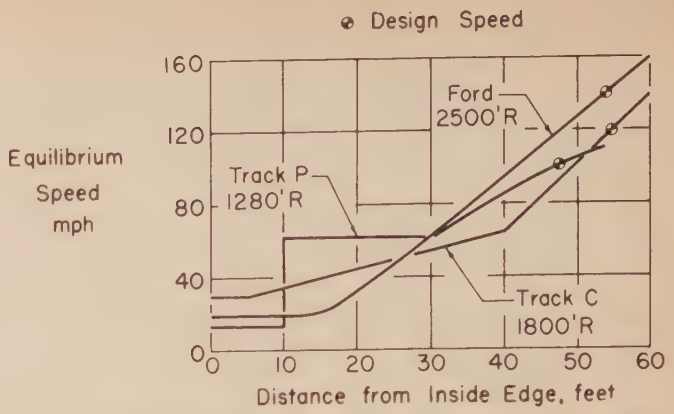


Fig. 11—Equilibrium speed and width relationship of three high-speed track curves

The choice of cross-section form depends largely on anticipated usage and the speed distribution desired across the width of the track. Cross-sections of the curves of three automotive tracks are shown in Fig. 9. The actual section reveals little except that track P required considerably more earthwork than the other two. A plot showing the slopes of these cross-sections, Fig. 10, is more revealing. Track P has a higher slope over most of its width. Going one step further and plotting the equilibrium speeds corresponding to these slopes, we get the most informative comparison in Fig. 11. If we overlook the inner lane which is nearly flat in all cases, we note that track P was evidently designed for a preponderance of high-speed traffic, although its maximum design speed happens to be lowest because of the shorter radius. The center lanes are a straight slope with a constant neutral speed, and the outer lanes are a quadratic form. Track C shows two straight lines, indicating that two cubic equations were used, each giving a uniform velocity distribution across part of the track. Two-thirds of the width have been devoted to low-speed operation and a relatively narrow portion at the outer edge for very high-speed use. The Ford track uses a single cubic cross-section with a uniform velocity distribution, except for the inner lane which must have a minimum 1% slope for drainage. This section was chosen since we anticipated about equal volumes of extreme high-speed and low-speed (truck) traffic on our track. With narrower pavements, higher power equations for the cross-section are commonly used. Thus our Arizona track has a quartic or fourth power cross-section (Figs. 12 and 13). The speed distribution of the quartic form approximates that of track C, in that while most of the track width is available for normal speed driving, a considerable margin for higher speeds is obtained within a limited width near the outside edge. (See Figs. 14 and 15.)

Once a cross-section form is decided upon, one

Fig. 12—Rolling superelevation at Ford Arizona track to specifications

Fig. 13—Paving superelevation at Ford Arizona track

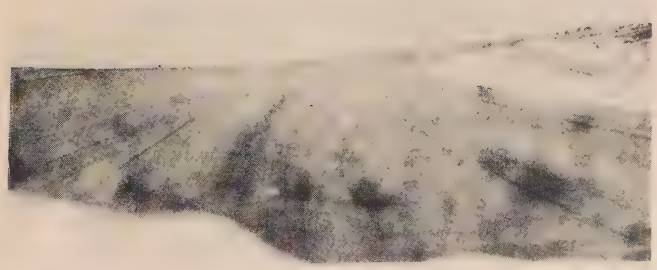


Fig. 14—Ford Arizona track at 140-mph speed

Fig. 15—Ford Arizona track at 140-mph speed

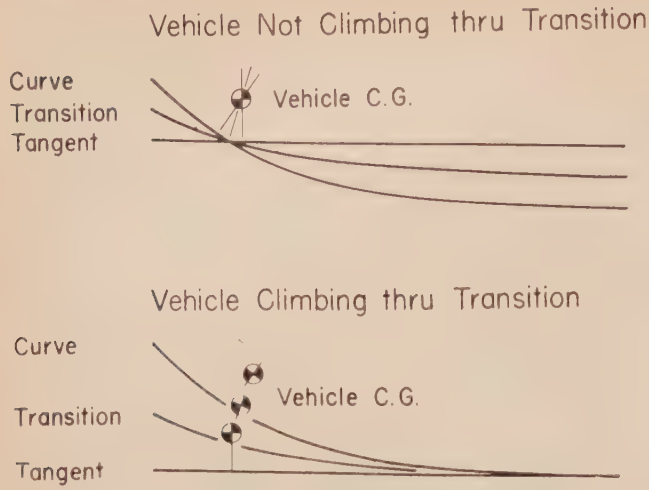


Fig. 16—Possible vertical motion in transition

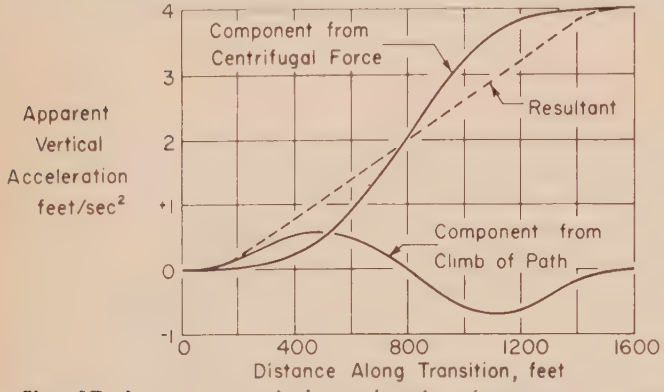


Fig. 17—Apparent vertical acceleration (components perpendicular to pavement) of superelevated spiral transition curve designed for minimum roll jerk

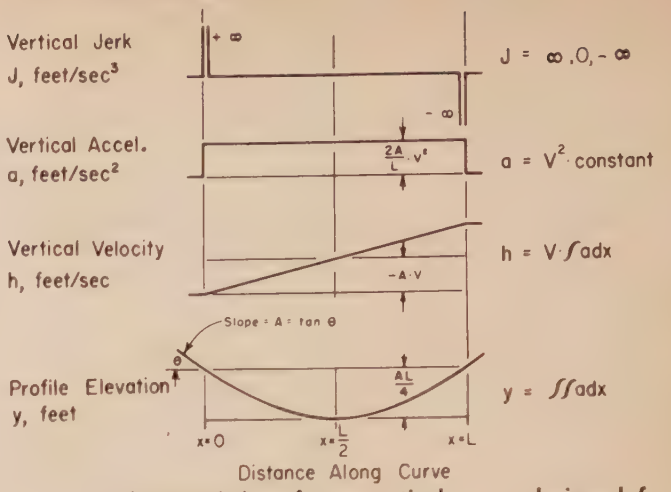


Fig. 18—Characteristics of sag vertical curve designed for minimum acceleration

further problem arises which is illustrated in Fig. 16. Should the vehicle center of gravity be kept at a constant height, or should it rise through the transition? Should the pavement be warped about the design path, the inner edge, or some other point? It is apparent that in either case, the center of gravity can be kept level only for one particular speed or path; on all other paths the vehicle will either climb or fall through the transition. The design of Ford's Michigan track has the vehicle climb through the transition. The reason for this is shown in Fig. 17. The force which the driver perceives as a vertical acceleration is made up of two components: one component is the difference between his true weight and the resultant of his weight with centrifugal force; the other component derives from the climb of the vehicle through the transition. The component from centrifugal force starts out slowly, increases rapidly near the midpoint, and then levels off. The rate of increase is above the perception threshold for vertical jerk at the midpoint. To obtain a minimum jerk requires a uniform linear increase of apparent weight. This can be accomplished, and the vertical jerk brought within theshold limits by climbing through the transition, giving the vehicle an additional vertical acceleration during the first half and decelerating it vertically during the last half of the spiral.

By integrating the required additional vertical component twice, we find the required climb of the design path to give us the desired minimum rate of change. We also find that a cubic cross-section warped about the inside edge will give the correct rise to the design path and for all other speeds as well.

As a final check on the transition design, all remaining motion characteristics were calculated. Longitudinal velocity was assumed constant with no longitudinal acceleration or jerk. Pitch motions are extremely slight and well below threshold values. In practice, they may be compensated by the slight longitudinal changes occasioned by the vehicle decelerating as it climbs through the transition. Yaw acceleration and jerk are likewise extremely small and well below threshold.

Vertical-Curve Design

Expressways which will steer the vehicles and keep them in the proper speed lane on curves by requiring constant effort to drive in any other lane are theoretically possible but quite unlikely unless roads can be built for 1-way traffic and with nothing but right turns. An immediate application for sensitivity data does exist, however, in the design of vertical curves such as sags, crests, dips, and rises.

Present practice on vertical curves is to use a quadratic, or parabolic form, with characteristics as shown in Fig. 18. With this form of curve, vertical acceleration is a constant minimum amount, but vertical jerk will be infinite at the beginning and end of the curve. While it may be possible to make such a curve sufficiently long so that the transition will be comfortable, a more efficient design would use a cubic form so that both vertical acceleration and jerk could be held to some desired finite value. Fig. 19 shows the vertical characteristics for a sag curve designed to minimize jerk. These equations can be

solved to give the curve profile, the vertical velocity, the vertical acceleration, and the vertical jerk for any station along the curve in terms of the initial slope, the design speed, and the curve length.

The expression for jerk can be equated to the jerk threshold value of 0.8 ft per sec ^{2} to find the minimum length of sag which will be imperceptible for any particular design speed and slope condition. With the length known, elevations of the curve profile can be found. The complete derivation and a sample calculation for a sag curve is given in Appendix II.

By equating both jerk and acceleration to their respective threshold values of 0.8 ft per sec ^{3} and 4 ft per sec^2, and solving the equations simultaneously with the vertical velocity expression, it is found that acceleration will not become the governing value in establishing the minimum sag length until vertical velocities exceed 10 ft per sec. The product of design speed and slope equals the vertical velocity. It is evident that design speeds much higher than are normally used on highways or steeper slopes than are suggested in the AASHO minimum design standards will be involved before acceleration rather than jerk governs the design—something like 70 mph on a 10% slope, for example.

The same shape curve described in Fig. 19 for a sag curve can be used inverted on crests or modified for highway disturbances such as are common in the Southwest for crossing dry washes. The same general dimensions of length and depth can be maintained as are now used, and the dip made imperceptible to the motorist. A sample calculation for shape which would convert an actual dip on a United States highway near our Arizona Proving Ground from an especially severe jolt to an imperceptible wave is included in Appendix III.

Acknowledgment

We wish to thank personnel of General Motors, Chrysler, and Packard proving grounds for allowing us to visit and drive their automotive test tracks, and for supplying us with much useful design data on their facilities.

APPENDIX I

Design of a Horizontal Transition Spiral

The problem is to develop a superelevated spiral transition from a straight to a curved path so that all angular and translational motions involved will be imperceptible.

Because of the extent of calculations involved, the problem will be reduced to defining the geometric shape of a particular transition used at our Romeo, Michigan, Proving Ground. The design speed and road width have been determined from projected vehicle performance, test requirements, traffic density, and economic considerations beyond the scope of this paper. Design conditions and constants are given in Table 2.

To avoid cumulative errors and since the business

Table 2—Design Conditions and Constants

Design Speed	140 mph (205.333 fps) at center of outside lane
Final Curve Radius	2500 ft at design path to keep below yaw and vertical (apparent weight increase) perception thresholds
Track Width	60 ft
Acceleration of Gravity	32.162 ft per sec^2 at Romeo
Station Interval	25 ft to avoid natural frequencies of tires and suspensions

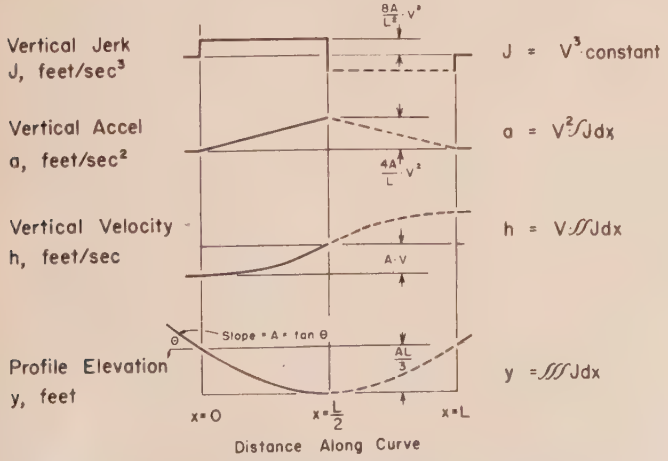


Fig. 19—Characteristics of sag vertical curve designed for minimum jerk

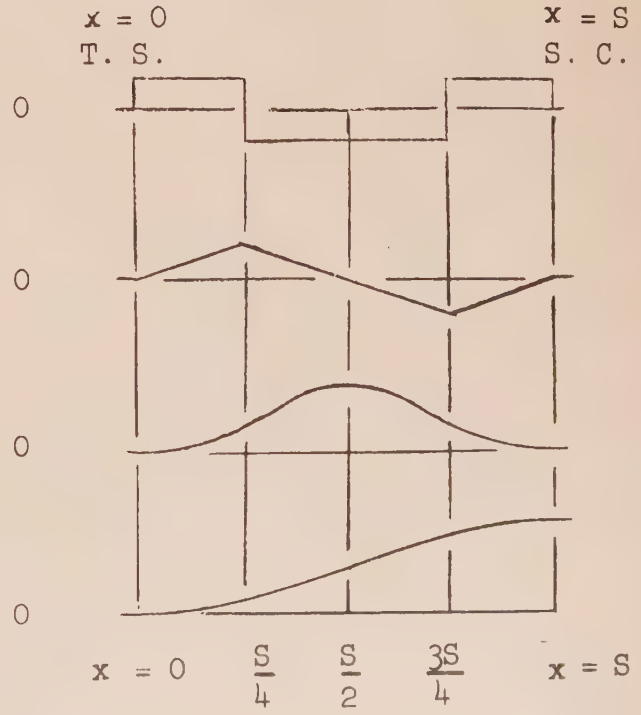


Fig. 20—Diagram of roll motion characteristics

machines don't care, calculations have in many cases been carried out to a precision which is neither significant nor warranted and rounding off left to the man with the trowel.

Step 1—Diagram roll-motion characteristics (see Fig. 20.)

$$\text{Jerk} = J = \pm C \text{ deg per ft}^3 = \pm CV^3 \text{ deg per sec}^3$$

$$\text{Acceleration} = A = \int J dx \text{ deg per ft}^2 = V^2 \int J dx \text{ deg per sec}^2$$

$$\text{Roll Velocity} = H = \iint J dx \text{ deg per ft} = V \iint J dx \text{ deg per sec}$$

$$\text{Roll Angle} = \phi = \iiint J dx \text{ deg}$$

where:

V = Design speed, fps
S = Spiral length, ft

Step 2—Develop equations of motion by integration:

From $x = 0$ to $x = \frac{S}{4}$:

$$J = C$$

$$A = Cx$$

$$H = \frac{Cx^2}{2}$$

$$\phi = \frac{Cx^3}{6} \qquad (1)$$

From $x = \frac{S}{4}$ to $x = \frac{3S}{4}$:

$$J = -C$$

$$A = -Cx + \frac{CS}{2}$$

$$H = -\frac{Cx^2}{2} + \frac{CSx}{2} - \frac{CS^2}{16}$$

$$\phi = -\frac{Cx^3}{6} + \frac{CSx^2}{4} - \frac{CS^2x}{16} + \frac{CS^3}{192} \qquad (2)$$

From $x = \frac{3S}{4}$ to $x = S$:

$$J = +C$$

$$A = Cx - CS$$

$$H = \frac{Cx^2}{2} - CSx + \frac{CS^2}{2}$$

$$\phi = \frac{Cx^3}{6} - \frac{CSx^2}{2} + \frac{CS^2x}{2} - \frac{13CS^3}{96} \qquad (3)$$

Step 3—Evaluate constant C in terms of spiral length, S:

When $x = \frac{S}{2}$, $\phi = \frac{\Delta\theta}{2}$

where $\Delta\theta$ is the total change in roll angle from straightaway to the curve.

In the curve:

$$\tan\theta = \frac{V^2}{gR} = \frac{(205.333)^2}{(32.162)(2500)} = 0.52433522$$

$$\theta = 27 \text{ deg } 40 \text{ min } 10.58 \text{ sec}$$

In the straight:

$$\tan\theta = 0.01 \text{ drainage slope}$$

$$\theta = 0 \text{ deg } 34 \text{ min } 22.58 \text{ sec}$$

$$\Delta\theta = 27 \text{ deg } 5 \text{ min } 48 \text{ sec} = 97{,}548 \text{ sec}$$

$$\phi = 48{,}774 \text{ sec} = \frac{48{,}774}{3600} \text{ deg}$$

At:

$$x = \frac{S}{2},\ \phi = C\left(\frac{-S^3}{48} + \frac{S^3}{16} - \frac{S^3}{32} + \frac{S^3}{192}\right) = \frac{48{,}774}{3600}$$

Therefore:

$$C = \frac{64}{S^3}\left(\frac{48{,}774}{3600}\right) \text{deg per ft}^3 = \frac{64}{S^3}\left(\frac{48{,}774}{3600}\right)V^3 \text{ deg per sec}^3$$

Step 4—Determine spiral length:

$$\text{Maximum jerk} = C = \frac{64}{S^3}\left(\frac{48{,}774}{3600}\right) \text{deg per ft}^3 = \frac{64}{S^3}\left(\frac{48{,}774}{3600}\right)V^3 \text{ deg per sec}^3$$

should be equal to or less than 2-deg per sec^3 threshold value:

$$2 = \frac{64}{S^3}\left(\frac{48{,}774}{3600}\right)(205.333)^3$$

$S = 1560$ ft. Use 1600 ft for spiral length.

Maximum acceleration,

$$\text{at } x = \frac{S}{4},\ = \frac{CS}{4}V^2 \text{ deg per sec}^2 = \frac{64}{(1600)^3}\left(\frac{48{,}774}{3600}\right)\left(\frac{1600}{4}\right)(205.333)^2$$

$$= 3.57 \text{ deg per sec}^2$$

This is less than the 4.0-deg per sec^2 acceleration threshold and is satisfactory.

Maximum velocity,

$$\text{at } x = \frac{S}{2},\ = C\left(\frac{-S^2}{8} + \frac{S^2}{4} - \frac{S^2}{16}\right)V \text{ deg per sec} = \frac{64}{(1600)^3}\left(\frac{48{,}774}{3600}\right)\frac{(1600)^2}{16}(205.333)$$

$$= 6.96 \text{ deg per sec}$$

This is less than the 8.0-deg per sec velocity threshold and is satisfactory.

Therefore:

Spiral length of 1600 ft will keep roll motions below perception threshold.

Step 5—Substitute values for S, V, and C in equations (1), (2), and (3) to find roll angle, ϕ, as a function of distance, x:

From $x = 0$ to $x = 400$ ft

$$\phi = 0.000127015625x^3 \text{ sec}$$

From $x = 400$ ft to $x = 1200$ ft

$$\phi = 0.3048375\left(\frac{160{,}000}{3} - 400x + x^2 - \frac{x^3}{2400}\right)\text{sec}$$

From $x = 1200$ ft to $x = 1600$ ft

$$\phi = 0.3048375\left(-\frac{4{,}160{,}000}{3} + 3200x - 2x^2 + \frac{x^3}{2400}\right)\text{sec}$$

Step 6—Calculate roll angles and slopes for each 25-ft station through transition from above equations:

Determine by trigonometric relations the horizontal coordinates of the design path at each station and the angle of the tangent at that point for field-layout use. These calculations are detailed in Table 3 for several stations.

Step 7—Develop cross-section equation:

The inner lane is to be flat, with a 1% drainage slope to the inside. The remaining cross-section is to be a cubic parabola, with the origin at the inside edge. The cubic must be adjusted for a slope and elevation match where it joins the inside lane, the first portion being discarded.

From $x = 0$ to $x = 12$

$$y = 0.01x \qquad (1)$$

Table 4—Transition Cross-Section

Station 32—800 ft Along Transition					Station 64—1600 ft Along Transition				
x	$(x-12)$	$K(x-12)^3$	$+\ 0.01x$	$= y$	x	$(x-12)$	$K(x-12)^3$	$+\ 0.01x$	$= y$
0				0.0000	0				0.0000
12	0	0.0000	0.1200	0.1200	12	0	0.0000	0.1200	0.1200
14	8	0.0004	0.1400	0.1404	14	8	0.0008	0.1400	0.1408
16	64	0.0029	0.1600	0.1629	16	64	0.0062	0.1600	0.1662
18	216	0.0099	0.1800	0.1899	18	216	0.0210	0.1800	0.2010
20	512	0.0234	0.2000	0.2234	20	512	0.0498	0.2000	0.2498
22	1000	0.0456	0.2200	0.2656	22	1000	0.0972	0.2200	0.3172
24	1728	0.0789	0.2400	0.3189	24	1728	0.1679	0.2400	0.4079
26	2744	0.1253	0.2600	0.3853	26	2744	0.2667	0.2600	0.5267
28	4096	0.1870	0.2800	0.4670	28	4096	0.3981	0.2800	0.6781
30	5832	0.2662	0.3000	0.5662	30	5832	0.5668	0.3000	0.8668
32	8000	0.3652	0.3200	0.6852	32	8000	0.7775	0.3200	1.0975
34	10648	0.4861	0.3400	0.8261	34	10648	1.0349	0.3400	1.3749
36	13824	0.6311	0.3600	0.9911	36	13824	1.3436	0.3600	1.7036
38	17576	0.8023	0.3800	1.1823	38	17576	1.7082	0.3800	2.0882
40	21952	1.0021	0.4000	1.4021	40	21952	2.1335	0.4000	2.5335
42	27000	1.2325	0.4200	1.6525	42	27000	2.6242	0.4200	3.0442
44	32768	1.4958	0.4400	1.9358	44	32768	3.1848	0.4400	3.6248
46	39304	1.7942	0.4600	2.2542	46	39304	3.8200	0.4600	4.2800
48	46656	2.1298	0.4800	2.6098	48	46656	4.5345	0.4800	5.0145
50	54872	2.5049	0.5000	3.0049	50	54872	5.3331	0.5000	5.8331
52	64000	2.9216	0.5200	3.4416	52	64000	6.2202	0.5200	6.7402
54	74088	3.3821	0.5400	3.9221	54	74088	7.2007	0.5400	7.7407
56	85184	3.8886	0.5600	4.4486	56	85184	8.2791	0.5600	8.8391
58	97336	4.4433	0.5800	5.0233	58	97336	9.4602	0.5800	10.0402
60	110592	5.0485	0.6000	5.6485	60	110592	10.7496	0.6000	11.3486

$\tan\theta_T = 0.25157737$

$\tan\theta_T - 0.01 = 0.24157737$

$\dfrac{\tan\theta_T - 0.01}{5292} = = 0.0000456495$

$\tan\theta_T = 0.52433522$

$\tan\theta_T - 0.01 = 0.51433522$

$\dfrac{\tan\theta_T - 0.01}{5292} = K = 0.0000971911$

From $x = 12$ to $x = 60$

$$y = 0.01x + K\ (x - 12)^3 \qquad (2)$$

$$\frac{dy}{dx} = 0.01 + 3K\ (x - 12)^2$$

at $x = 12$, $y = 0.12$ ft for elevation match with inner lane

$\dfrac{dy}{dx} = 0.01$ for slope match

at $x = 54$ (design path),

Table 3—Transition Spiral Calculations for Several Stations

(S = 1600 ft, ΔS = 25 ft, R = 2500 ft, V = 140 mph)

Spiral Length to Station, ft	Roll Angle, ϕ [a]	Angle Design Path, θ [b]	Slope Design Path, tan θ	Sin $\frac{\beta}{2}$ [c]	Included Angle, β	Offset Angle, α [d]	Sin α	Cos α	Δa [e]	Δb [f]	Coordinates a [g]	Coordinates b [h]	Tangent Angle [i]
0	0° 00′ 00″	0° 34′ 23″	0.010000	0.000095	0° 00′ 00″	0° 00′ 00″	0.000000	1.000000	25.0000	0.000000	0	0	0° 00′ 00″
25	0° 00′ 2″	0° 34′ 25″	0.010009	0.000095	0° 00′ 39″	0° 00′ 20″	0.000095	1.000000	25.0000	0.002387	25.000	0.002	0° 00′ 39″
50	0° 00′ 16″	0° 34′ 38″	0.010077	0.000096	0° 00′ 40″	0° 00′ 59″	0.000287	1.000000	25.0000	0.007175	50.000	0.010	0° 1′ 19″
75	0° 00′ 54″	0° 35′ 16″	0.010260	0.000098	0° 00′ 40″	0° 1′ 39″	0.000481	1.000000	25.0000	0.012024	75.000	0.022	0° 1′ 59″
100	0° 2′ 7″	0° 36′ 30″	0.010616	0.000101	0° 00′ 42″	0° 2′ 20″	0.000680	1.000000	25.0000	0.017000	100.000	0.039	0° 2′ 41″
125	0° 4′ 8″	0° 38′ 31″	0.011203	0.000107	0° 00′ 44″	0° 3′ 3″	0.000888	1.000000	25.0000	0.022201	125.000	0.061	0° 3′ 25″
150	0° 7′ 9″	0° 41′ 31″	0.012078	0.000115	0° 00′ 48″	0° 3′ 49″	0.001109	0.999999	25.0000	0.027750	150.000	0.089	0° 4′ 13″
175	0° 11′ 21″	0° 45′ 43″	0.013300	0.000127	0° 00′ 52″	0° 4′ 39″	0.001352	0.999999	25.0000	0.033799	175.000	0.122	0° 5′ 5″
200	0° 16′ 56″	0° 51′ 19″	0.014927	0.000142	0° 00′ 59″	0° 5′ 34″	0.001621	0.999999	25.0000	0.040528	200.000	0.163	0° 6′ 4″
1375	26° 41′ 41″	27° 16′ 4″	0.515425	0.004915	0° 33′ 48″	13° 5′ 53″	0.226620	0.973983	24.3496	5.665500	1369.004	86.064	13° 22′ 47″
1400	26° 48′ 52″	27° 23′ 14″	0.518071	0.004940	0° 33′ 58″	13° 39′ 46″	0.236207	0.971703	24.2926	5.905182	1393.296	91.969	13° 56′ 45″
1425	26° 54′ 27″	27° 28′ 50″	0.520135	0.004960	0° 34′ 6″	14° 13′ 48″	0.245815	0.969317	24.2329	6.145378	1417.529	98.114	14° 30′ 51″
1450	26° 58′ 39″	27° 33′ 2″	0.521688	0.004975	0° 34′ 12″	14° 47′ 57″	0.255432	0.966827	24.1707	6.385808	1441.700	104.500	15° 5′ 3″
1475	27° 1′ 40″	27° 36′ 3″	0.522803	0.004985	0° 34′ 17″	15° 22′ 11″	0.265049	0.964235	24.1059	6.626219	1465.806	111.126	15° 39′ 20″
1500	27° 3′ 41″	27° 38′ 4″	0.523550	0.004992	0° 34′ 19″	15° 56′ 29″	0.274656	0.961543	24.0386	6.866400	1489.844	117.993	16° 13′ 39″
1525	27° 4′ 54″	27° 39′ 17″	0.524004	0.004997	0° 34′ 21″	16° 30′ 50″	0.284247	0.958751	23.9688	7.106171	1513.813	125.099	16° 48′ 00″
1550	27° 5′ 32″	27° 39′ 55″	0.524237	0.004999	0° 34′ 22″	17° 5′ 12″	0.293816	0.955862	23.8966	7.345390	1537.710	132.444	17° 22′ 23″
1575	27° 5′ 46″	27° 40′ 9″	0.524323	0.005000	0° 34′ 23″	17° 39′ 34″	0.303358	0.952877	23.8219	7.583947	1561.532	140.028	17° 56′ 45″
1600	27° 5′ 48″	27° 40′ 11″	0.524335	0.005000	0° 34′ 23″	18° 13′ 56″	0.312871	0.949796	23.7449	7.821770	1585.277	147.850	18° 31′ 8″

[a] ϕ obtained from roll equations

[b] $\theta = \phi$ plus drainage angle of $1\% = \phi + 0° 34′ 23″$

[c] $\text{Sin}\ \frac{\beta}{2} = \frac{1}{2}\ \text{chord} \div \text{radius} = \frac{12.5}{r}$; $\text{radius} = \frac{V^2}{8 \tan\theta}$ $\therefore \sin\frac{\beta}{2} = \frac{12.5\ 8 \tan\theta}{V^2}$

$= 0.0095354 \tan\theta$

[d] α determined from progression; $\alpha_1 = \frac{\beta_1}{2}$; $\alpha_2 = \beta_1 + \frac{\beta_2}{2}$; $\alpha_n = \beta_1 + \beta_2 + \ldots + \beta_{n-1} + \frac{\beta_n}{2}$

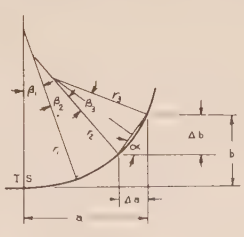
[e] $\Delta a = 25 \cos\alpha$

[f] $\Delta b = 25 \sin\alpha$

[g] $a = \Sigma\ \Delta a$

[h] $b = \Sigma\ \Delta b$

[i] Tangent Angle $= \Sigma\ \beta$



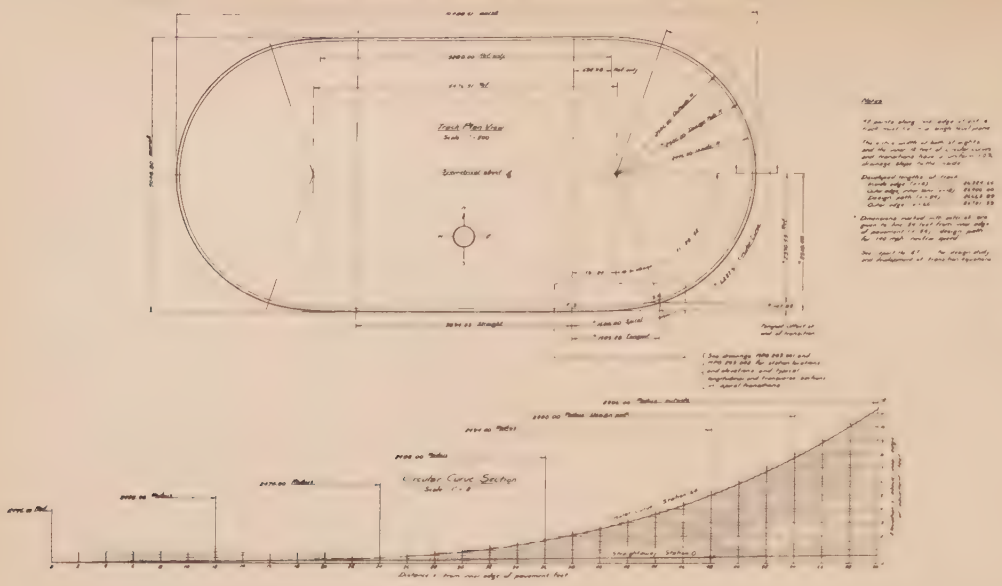


Fig. 21—Geometric design of high-speed track

$$\frac{dy}{dx} = \tan\theta \text{ for proper roll motion}$$
$$= 0.01 + 3K\ (42)^2$$

Therefore:

$$K = \frac{\tan\theta - 0.01}{5292} \qquad (3)$$

where:

x = Horizontal distance from inner edge, ft
y = Elevation above inner edge, ft
$\tan\theta$ = Slope of cross-section at design path

Table 4 and Figs. 21 and 22 show cross-section calculations for two typical stations. These calculations can be performed rapidly by machine; previous calculations could not, as trig functions were not available on IBM cards to the required accuracy. Fig. 23 shows plan of Michigan track based on calculations.

APPENDIX II

Sag Vertical Curve Design

The problem is to develop an equation for a sag vertical curve with imperceptible vertical acceleration and jerk.

Step 1—Diagram motion characteristics and write their equations (see Fig. 24):

Jerk $= J =$ constant $= C$ ft per ft^3 $= CV^3$ ft per sec^3

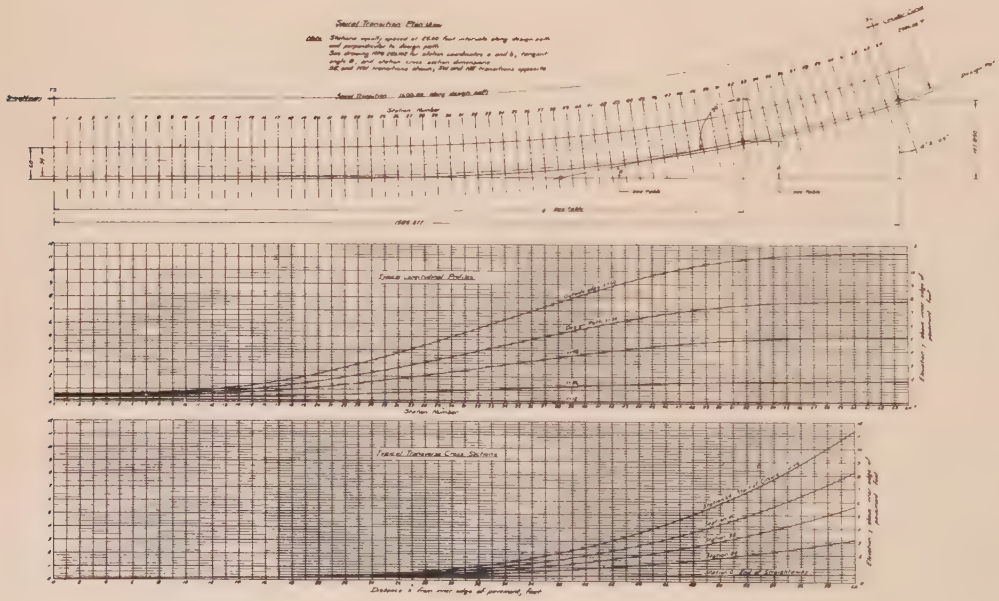


Fig. 22—Spiral transition station locations, showing typical profiles and cross-sections

Acceleration $= a = \int J dx = Cx$ ft per ft^2

$= CxV^2$ ft per sec^2

Vertical Velocity $= H = \int\int J dx$

$$= \left(\frac{Cx^2}{2} - \frac{CL^2}{8}\right) \text{ft per ft} = \left(\frac{Cx^2}{2} - \frac{CL^2}{8}\right) V \text{ fps}$$

Elevation $= y = \int\int\int J dx = \left(\frac{Cx^3}{6} - \frac{CL^2x}{8} + \frac{CL^3}{24}\right)$ ft

where:

L = Sag length, ft

V = Horizontal velocity or design speed, fps

Step 2—Evaluate constant C:

When $x = 0$, slope $= \tan \theta = A$

$$\frac{dy}{dx} = H = A = \frac{Cx^2}{2} - \frac{CL^2}{8}$$

Therefore:

$$C = \frac{8A}{L^2}$$

Step 3—Substitute constant in equations:

$$\text{Jerk} = \left(\frac{8A}{L^2}\right) V^3 \text{ ft per sec}^3 \qquad (1)$$

Should be equal to or less than 0.8-ft per sec^3 perception threshold.

Equate jerk to threshold value, substitute desired values for slope A and design speed V to determine minimum sag length.

$$\text{Acceleration} = \frac{8A}{L^2} V^2 \text{ ft per sec}^2 \qquad (2)$$

Table 5—Profile of Sag Curve

	x, ft	y, ft
Start of Sag	0	16.0
	100	13.0
	200	10.1
	300	7.4
	400	5.0
	500	3.0
	600	1.4
	700	0.4
Bottom of Sag	800	0

Sag is symmetrical about bottom point, so calculations are necessary for first half only.

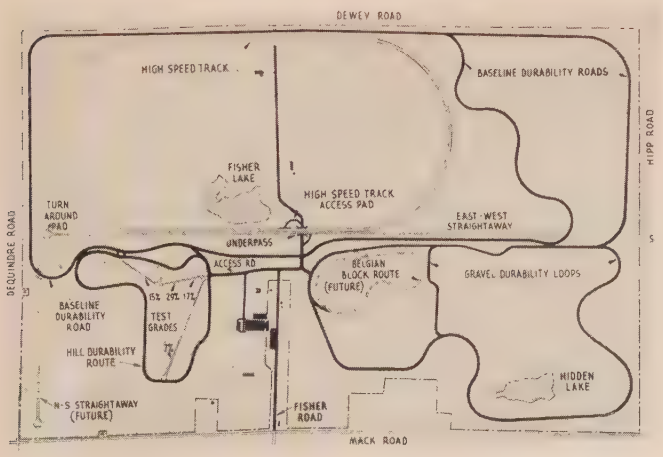


Fig. 23—Plan of Ford's Michigan proving ground

Should be equal to or less than 4.0-ft per sec^2 perception threshold.

$$\text{Elevation} = \left(\frac{8A}{L^2}\right)\left(\frac{x^3}{6} - \frac{L^2x}{8} + \frac{L^3}{24}\right) \text{ft} \qquad (3)$$

Substitute value of L from (1) above, and desired slope A in this equation to determine elevation at any station x.

Example—Determine profile of sag curve between two 3% slopes for minimum length with no perceptible change. Design speed 140 mph (205 fps). See Table 5 for dimensions of sag curve.

$$\text{Jerk} = \left(\frac{8A}{L^2}\right) V^3 = 0.8\text{-ft per sec}^3 \text{ threshold.}$$

$$L^2 = \frac{(8)\ (0.3)\ (205)^3}{0.8} = 2{,}560{,}000$$

$L = 1600$-ft sag length.

$$\text{Acceleration} = \left(\frac{8A}{L^2}\right) V^2 x$$

where maximum value at $x = \frac{L}{2}$

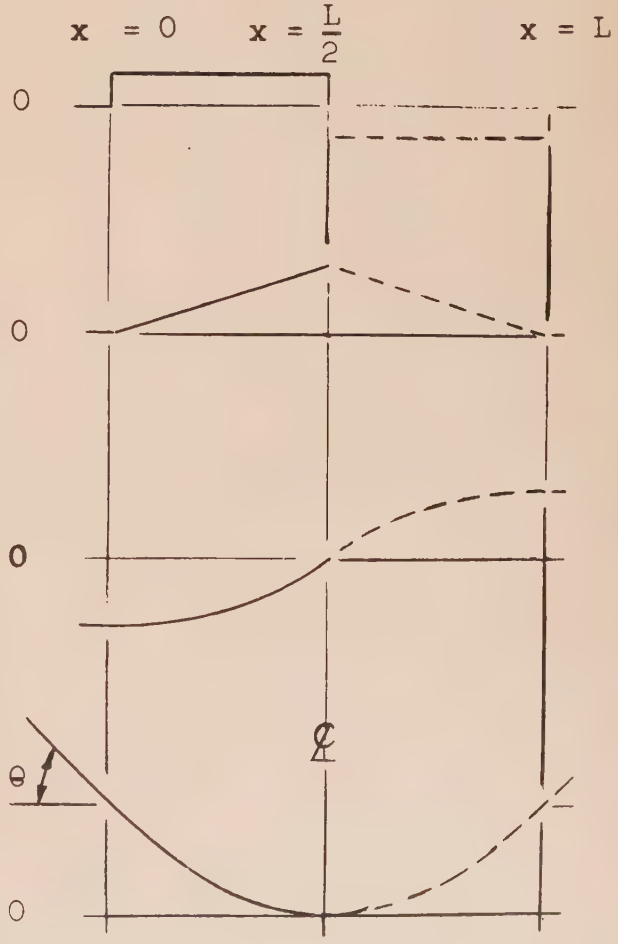


Fig. 24—Diagram of motion characteristics for sag vertical curve

Table 6—Design of Highway Dip

	Station, x, ft	Depth, y, ft
Level	0	0
	50	0
	100	0.3
	150	1.0
	200	2.3
	250	4.0
L/2	300	6.0
	350	8.0
	400	9.7
	450	11.0
	500	11.7
	550	12.0
Bottom	600	12.0

Therefore:

$$A_{max} = \frac{4A}{L} V^2 = 3.15 \text{ ft per sec}^2$$

Maximum value is below 4.0-ft per sec^2 threshold.

$$\text{Elevation} = y = \left(\frac{8A}{L^2}\right)\left(\frac{x^3}{6} - \frac{L^2x}{8} + \frac{L^3}{24}\right) \text{ ft}$$

$$= \frac{x^3}{64{,}000{,}000} - 0.03x + 16$$

APPENDIX III

Highway Dip and Rise Design

The problem is to develop an equation for a highway dip with imperceptible vertical motions. Rises such as a grade separation will use the same equations and have the same form but with all characteristics inverted.

Step 1—Diagram motion characteristics (see Fig. 25):

$$\text{Jerk} = J = \pm C \text{ ft per ft}^3 = \pm CV^3 \text{ ft per sec}^3$$

$$\text{Acceleration} = a = \int J dx \text{ ft per ft}^2$$

$$= V^2 \int J dx \text{ ft per sec}^2$$

$$\text{Velocity} = H = \iint J dx \text{ ft per ft} = V \iint J dx \text{ fps}$$

$$\text{Depth} = y = \iiint J dx \text{ ft}$$

Dip is symmetrical, so derivation is confined to one side only.

where:

L = Length of vertical curve, ft
V = Horizontal velocity, or design speed, fps

Step 2—Develop equations of motion by integration:

From $x = 0$ to $x = \frac{L}{4}$

$$J = -C$$

$$a = -Cx$$

$$H = \frac{-Cx^2}{2}$$

$$y = \frac{-Cx^3}{6}$$

From $x = \frac{L}{4}$ to $x = \frac{3L}{4}$

$$J = +C$$

$$a = Cx - \frac{CL}{2}$$

$$H = \frac{Cx^2}{2} - \frac{CLx}{2} + \frac{CL^2}{16}$$

$$y = \frac{Cx^3}{6} - \frac{CLx^2}{4} + \frac{CL^2x}{16} - \frac{CL^3}{192}$$

Step 3—Evaluate Constant C:

When $x = \frac{L}{2}$, $y = \frac{d}{2}$

where d is the desired depth of the dip

Therefore:

$$y = \frac{d}{2} = \frac{CL^3}{48} - \frac{CL^3}{16} + \frac{CL^3}{32} - \frac{CL^3}{192}$$

$$C = -\frac{32d}{L^3}$$

Step 4—Substitute constants in equations:

$$\text{Jerk} = J = \pm \frac{32d}{L^3} V^3 \text{ ft per sec}^3 \qquad (1)$$

J should be equal to or less than 0.8-ft per sec^3 threshold limit.

Substitute 0.8 value for J, and desired values for depth, d and design speed, V and solve for minimum length L.

$$\text{Acceleration} = a = \frac{32dx}{L^3} V^2 \text{ ft per sec}^2,$$

and a is maximum at $x = \frac{L}{4}$

$$a_{max} = \frac{8d}{L^2} V^2 \text{ ft per sec}^2 \qquad (2)$$

Check to make sure maximum acceleration is below 4.0-ft per sec^2 threshold limit.

From $x = 0$ to $x = \frac{L}{4}$

$$\text{Depth} = y = \frac{16dx^3}{3L^3} \text{ ft} \qquad (3)$$

From $x = \frac{L}{4}$ to $x = \frac{3L}{4}$

$$y = \left(-\frac{16dx^3}{3L^3} + \frac{8dx^2}{L^2} - \frac{2dx}{L} + \frac{d}{6}\right) \text{ ft} \qquad (4)$$

Substitute value of L from (1) above, and desired value for d. Solve for y at various stations, x.

It is not necessary to use equations beyond $x = \frac{L}{2}$, as elevations between $x = \frac{L}{2}$ and $x = L$ can be found from symmetry of curve.

Example—Design of highway dip with a maximum depth of 12 ft, for a design speed of 50 mph (approximately 75 fps) with no perceptible vertical motions. Table 6 gives calculated design for highway dip.

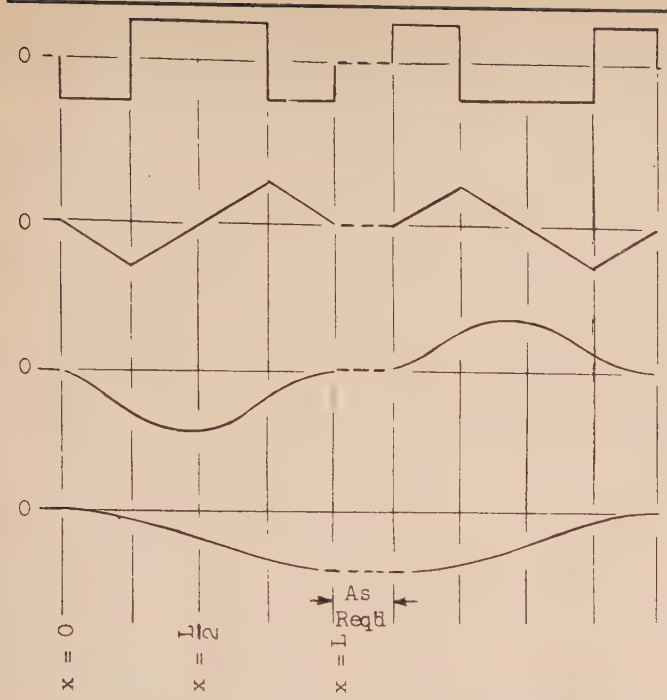


Fig. 25—Diagram of motion characteristics of highway dip

$$J = \frac{32d}{L^3} \quad V^3 = 0.8\text{-ft per sec}^3 \text{ threshold}$$

$$L^3 = \frac{(32)\ (12)\ (75)^3}{0.8}$$

$$L = \ 600 \text{ ft (approximately)}$$

$$a_{max} = \frac{8d}{L^2} V^2 = 1.5 \text{ ft per sec}^2 = <4.0 \text{ threshold}$$

From $x = 0$ to $x = 150$

$$y = \frac{16dx^3}{3L^3} = \frac{x^3}{3{,}375{,}000}$$

From $x = 150$ to $x = 450$

$$y = -\frac{x^3}{3{,}375{,}000} + \frac{x^2}{3750} - \frac{x}{25} + 2$$

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ORAL DISCUSSION

Reported by R. E. Owen

General Motors Corp.

K. A. Stonex, General Motors Corp.: What smoothness tolerance of the surface is required to produce a high-speed track?

Mr. McConnell: Specifications for our new tracks called for a surface within $\pm\frac{1}{8}$ in. of design geometry. The shape and spacing of deviations is important, and much larger variations can be unnoticed. At our Arizona track, some regularly spaced $\frac{5}{16}$-in. depressions at 200-ft intervals proved very disturbing because of suspension resonance at 140 mph. Corrections were made by driving nails into the surface with their heads at the design elevation, in a 2-ft grid, and blading in additional asphaltic concrete material. Some evidence remains that a car running 140 mph is the most sensitive instrument for evaluating the smoothness of the finished surface. The precision possible appears to be limited by surveying accuracy rather than construction technique. Conventional highway finish specifications require impossible surveying tolerances, are unnecessary in some respects, and do not control the types of variations which may be significant.

D. Whitcomb, Cornell Aeronautical Laboratory, Inc.: To what extent is Cornu's spiral used to design highway curves—that is, how many highway curves that one meets in ordinary driving are designed this way?

Mr. McConnell: Most turnpike curves are designed using tables for spirals which are approximations of the Cornu spiral. Such tables show the degree of curvature for each of 10 chords in a transition or, in other cases, for 50-ft increments of arc length with resulting deflection angles and coordinates for field-layout use. Some form of transition spiral is standard practice on all modern roads.

W. F. Milliken, Jr., Cornell Aeronautical Laboratory, Inc.: Do you have any information on the AVVS track in Berlin where such high speeds have been reached?

Mr. McConnell: I do not have information about this track.

Observations on 25,000 Hours of

DURING the past few years there has been in the United States an ever increasing interest in free-piston engines. This has been evidenced by the publications from technical societies and in periodicals of many papers and articles which extoll the virtues of the free-piston gas-turbine cycle and point out the many advantages of such powerplants. Since the literature is fairly complete along these lines it was thought that more specific information on the free-piston engine would be timely and of general interest.

THIS paper presents observations of 25,000 hr of testing on five free-piston engines. Endurance test investigations were made especially on the operation of free-piston-engine air compressors and gasifiers.

As a result of the data obtained, basic parts of these engines were redesigned for better performance, power, economy, and reliability.

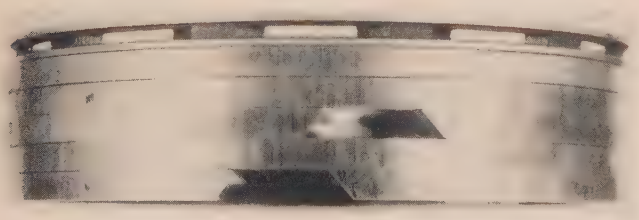

Fig. 1—P-42 piston-rings after 600 hr of operation

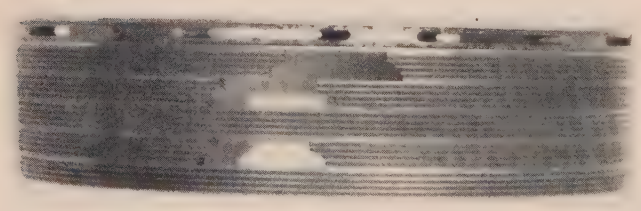

Fig. 2—New P-42 piston-rings resulting from further test work and giving improved performance

Therefore, it is the purpose of this paper to present some observations after 25,000 hr of free-piston-engine operation. It should be pointed out that these hours of operation were not accumulated on one engine, but rather represent the total hours accumulated on the following free-piston engines:

1. The Muntz P-42 air compressor.
2. The SIGMA P-13 air compressor.
3. The SIGMA GS-34 gasifier.
4. The GM-14 gasifier.
5. The GMR 4-4 (Hyprex) gasifier.

When we first became interested in free-piston engines we entered into a working agreement with two foreign firms then engaged in the development of this type of engine. There was, at this time, very little commercial application of free-piston engines. Although some experimental gasifiers had been installed in minesweepers of the French Navy and free-piston air compressors of the types mentioned above were available for sale in Europe, very little was known about their performance, life, or desirability over more conventional diesel engines. As has been pointed out, in the last few years there has been a considerable amount of literature published regarding the advantages and working principles of free-piston engines, but in order to obtain complete information it was necessary to purchase some representative engines and run evaluation tests. This, of course, is what was done, and more recently General Motors has been running two new gasifiers, namely the GM-14 and the GMR 4-4 (Hyprex) engine.

During the past few years we have learned most of the facts and foibles of free-piston engines. We are convinced of their many real advantages, and we are cognizant of their problems and limitations.

As is the case with any high-output engine, and a free-piston engine is merely a highly supercharged diesel engine, there are many problems to overcome. Early in our investigations there were several serious problems. During the subsequent development work

Free-Piston-Engine Operation

Gregory Flynn, Jr. Research Staff, GMC

This paper was presented at the SAE National West Coast Meeting, San Francisco, Aug. 8, 1956.

and the design and development programs of the last two engines, we have been able to eliminate or alleviate all of the major troubles encountered during the preliminary investigations.

Air Compressors

Both the P-42 and P-13 air compressors, as originally received, were fairly satisfactory machines. They would start readily and run at various loads for several hundred hours. The main difficulties were piston-ring scuffing and breakage, cylinder scoring, and lubricating oil deposits. Fig. 1 shows the piston-rings removed from a P-42 compressor after 600 hr of operation at various loads. The lubricant used was straight mineral oil, as specified by the manufacturer. The first tests run immediately pointed to the use of a modern heavy-duty lubricating oil to reduce deposits and ring scuffiing. The rings themselves seemed to need more than just a change in oil to be satisfactory.

Further test work was done using cast-iron-grooved and ferrox-filled piston-rings of American manufacture and a popular heavy-duty diesel-engine lubricating oil. These two basic changes resulted in a marked improvement in performance, as can be seen in Fig. 2, a photograph of the rings removed after 586 hr of running. Scuffing and scoring were completely eliminated, and the ring and cylinder wear was greatly reduced.

Since these and various other modifications, performance tests and several thousand hours of endurance tests have been run with completely satisfactory results. Free-piston air compressors are currently successful commercial machines. However, most of the interest in free-piston engines lies in the field of prime movers, which requires the use of a free-piston gas generator or "gasifier" and a gas turbine.

Gasifiers

A cross-section of a typical free-piston gasifier is shown in Fig. 3. This is the model GS-34 of SIGMA

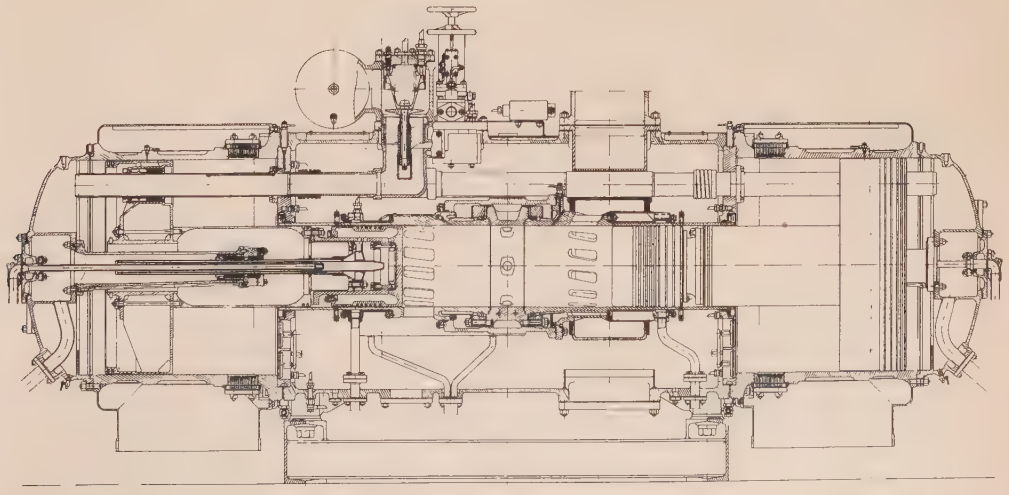

Fig. 3—Cross-section of 34 gasifier

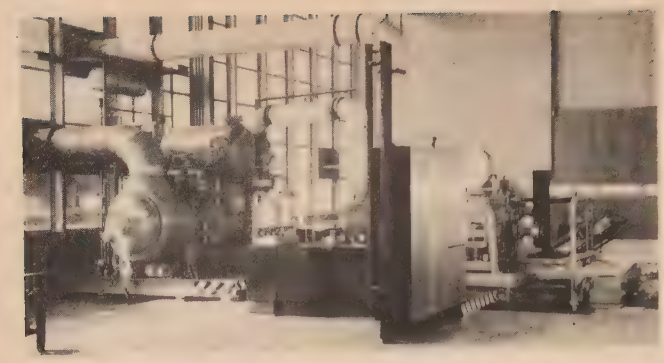

Fig. 4—Test setup of GS-34 gasifier—high bay

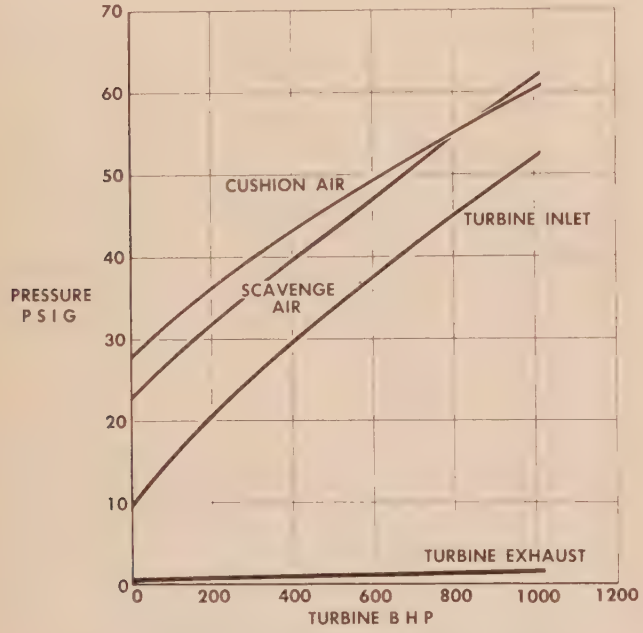

Fig. 5—Pressure versus power for GS-34 gasifier

as it was three years ago. The engine was rated at 1000 gas hp. This figure, of course, is reduced by the efficiency of the turbine, so that the brake horsepower on the shaft depends upon the turbine efficiency. For the sake of simplicity, however, we usually refer to free-piston gasifier output in terms of gas horsepower.

The engine is essentially a simple machine. It does have, of course, all the attendant oil pumps and injection equipment that any diesel engine requires. Other than that, however, the main moving parts are merely the two piston assemblies and the connection linkage. The pistons operate in a water-cooled cast-iron cylinder. Each of the cylinders, namely, the scavenge and exhaust cylinders, are bolted to a forged steel center section which carries most of the stress at the high-pressure part of the firing cycle. The pistons are steel, and oil-cooled as is our practice in high-output diesel engines. The pistons operate on the cylinder surface with cast copper-lead bronze wear bands, which actually are the bearing surface. The power pistons are directly bolted to the large steel compressor pistons—here again the piston bearing surfaces are cast bronze wear bands imbedded in the outside diameter of the piston. The compressor cylinders are cast iron.

The engine operates on the well-known opposed-piston uniflow 2-stroke-cycle principle. On the outward stroke of the pistons, air is drawn in through the inlet valves into the compressor cylinder. On the inward stroke of the pistons, air is compressed through the delivery valves into the airbox around the power cylinder. On the next outward stroke, this is the air that is used for scavenging the exhaust gases which are collected in a collector around the exhaust ports and conducted through a pipe to the turbine where shaft horsepower is extracted. The pistons are returned to the inner dead point—note that we refer to dead points rather than dead-centers, since they are constantly variable in a free-piston engine—by the compressed air in the dead spaces; these spaces are usually referred to as the "cushion" or "bounce" chambers at the extremities of the machine.

As the pistons move outward they compress air in the dead spaces. This is the air that is used to return the pistons to the inner dead point. All the energy in the firing charge is absorbed by compressing air in the bounce cylinders.

The speed of the engine depends upon the masses of the reciprocating pistons, as well as the stiffness of the airsprings upon which they act. The two springs are the bounce cylinder pressure and the compression pressure in the power cylinder. Varying any of these three variables will change the speed of the engine. Since the piston mass is of necessity constant, in order to control the engine and power output it is necessary to vary the stiffness of the airsprings in the two spaces mentioned. This is the function of the stabilizer or the governor. At times it can take air from the engine case and feed it into the bounce cylinders, thus, stiffening the spring in the bounce space and increasing the stroke and speed of the engine. At other times, if desired, the stabilizer will bleed air from the bounce cylinders into the airbox where it is consumed in the combustion process, thus, softening the spring or return force on the pistons, shortening the stroke, and slowing down the speed of the engine.

Our initial test work revealed that free-piston engines are essentially efficient machines. The thermal efficiency of the GS-34 gasifier, for instance is 43½%. Here again, reference is made to gas efficiency. Our test work further revealed that the free-piston engine is essentially insensitive to fuel quality. It seems that these engines do not care whether they get fuel with octane or cetane numbers. We have successfully run one free-piston engine on a range of fuels varying from 100-octane gasoline, through the gasoline range, kerosene, diesel fuel, bunker "C," and crude oil. In addition, we have run it on vegetable and animal oils, such as whale oil, cottonseed oil, or peanut oil. The only noticeable effect on engine output under these test conditions is the power output which

varies directly as the heat content of the fuel being used.

Early test work showed that the fuel consumption of the engine was quite acceptable, the exhaust was absolutely clean, and the engine would run satisfactorily on either diesel fuel, crude oil, or bunker "C." There were, however, several inherent mechanical problems existing in the engine itself. These are the problems that are associated with any high-output engine development and are probably common to most diesel engines, namely, cylinder-head or in the case of the free-piston engine "center section" stresses, piston-ring and cylinder conditions, injection problems, valve problems, lubrication problems, oil consumption, airbox fires, and many other minor difficulties.

The gas turbine, of course, is an essential part of the free-piston engine. One of the main advantages of this cycle is that the exhaust gas from the gasifier is only 850 F as it goes into the turbine, which eliminates the necessity for using critical or strategic materials in the turbine blading. It also prevents any damage from vanadium in the fuels which has occurred in high-temperature turbines. Since no trouble of any kind or any servicing has ever been required on the turbines we have tested, no further remarks will be made regarding turbine performance or maintenance.

The test installation of the GS-34 gasifier in our engine test building is shown in Fig. 4. Since a free-piston engine is inherently perfectly balanced, the engine was simply mounted on a framework of I-beams set on a wooden floor. It has been running here for three years and so far has not yet moved. The turbine was mounted on a dynamometer bedplate immediately adjacent to the gasifier, as shown. The turbine has an integral reduction gear and is connected to a dynamometer.

The set, as originally delivered, was rated at 600 kw on the turbine shaft. This is equivalent to 803 bhp or approximately 1000 gas hp from the gasifier. Our first tests were, of course, a run-in for the piston rings and check runs to determine that the turbine and gasifier controls were working properly. Fig. 5 shows the power output of the turbine plotted against gas pressure.

Fig. 6 is a plot of fuel consumption versus load. As can be seen, the part-load fuel economy of an inward compressing engine is not very attractive at low loads. This is due to the fact that at low horsepower the turbine only requires a small amount of gas, and the gasifier's output is restricted as shown in Fig. 7. As fuel to the gasifier is reduced, the pistons slow down and reduce their stroke. The minimum stroke is limited by a cam on the injector control so that the piston must always have sufficient stroke to uncover the ports to prevent stalling the engine. Thus, if it is desired to run the turbine at lower output, the excess gas must be thrown away through the bypass valve. This, of course, represents a direct loss of energy and an unfavorable low-load fuel consumption. Such a powerplant though satisfactory for powerhouse service would be quite undesirable for locomotives, for instance, where as much as 50% of the operating time is at low-load or idle throttle position.

Shortly after we began running our test, modifications showed great promise of alleviating the part-load fuel consumption. These were a selected pressure pickup unit for the stabilizer, and the use of a recirculation valve from the gasifier airbox. Addition of these two devices reduced the fuel consumption at idle to a level comparable to that of a good diesel engine. Thus, it was possible to consider such a power source for a locomotive or other vehicular application where a great deal of low-load operating must be done.

An explanation of the selected pressure pickup is perhaps too involved for the purposes of this paper; however, the recirculation valve is quite simple. It

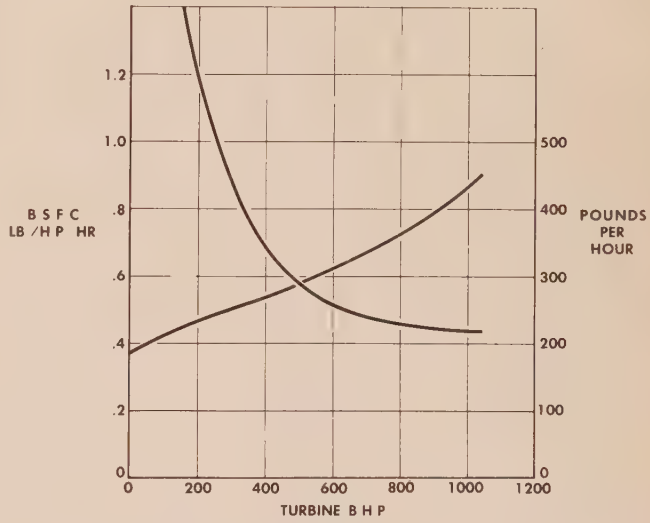

Fig. 6—Original performance of fuel consumption versus load for GS-34 gasifier

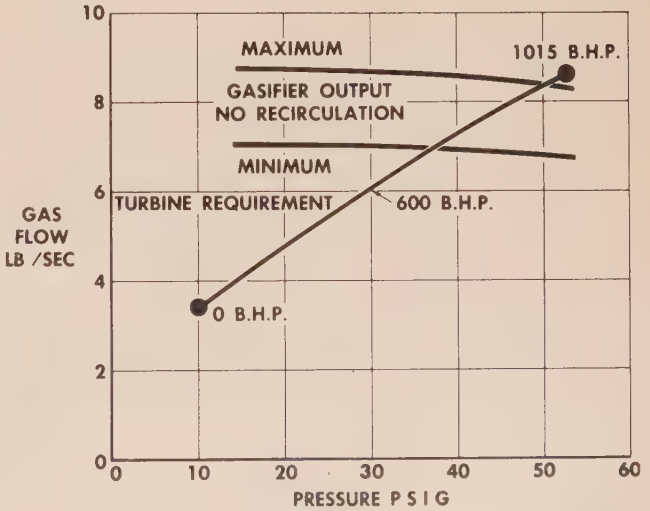

Fig. 7—Gas flow versus turbine demand for GS-34 gasifier

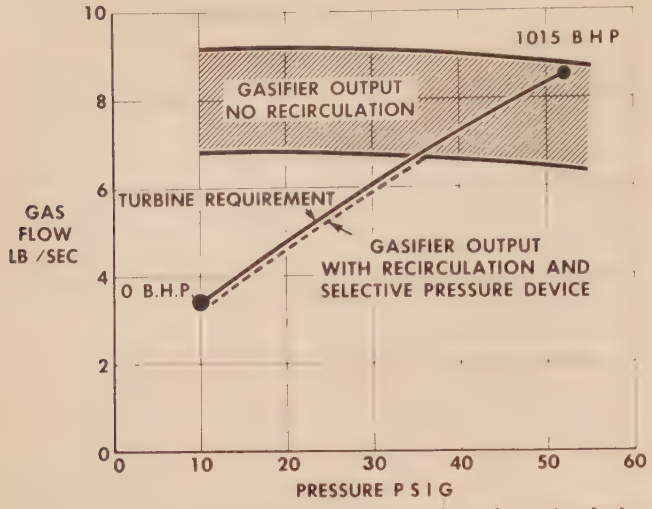

Fig. 8—Gas flow versus turbine demand with recirculation added for GS-34 gasifier

merely bypasses the air from the airbox around the cylinder back to the air inlet. As the air goes around and around the loop it gets heated with each pumping stroke of the compressor pistons until its temperature is raised to a rather high level. This reduces the amount of air pumped by reducing the density and allowing the gasifier speed and compression ratio to be reduced while still maintaining sufficient heat in the combustion chamber to ignite the fuel when it is injected. Thus, the compression ratio which at full load is approximately 50/1 can be reduced to 10/1 at idle with the recirculation valve open. With these two devices it is then possible to match perfectly the turbine gas demand curve all the way down to 0 hp or idle on the gasifier. This, of course, would make it quite an attractive powerplant for vehicular service.

As seen in Fig. 8 the addition of a recirculation valve permits the gasifier output to be matched perfectly to the turbine requirement right down to idle. In numbers the idle fuel rate was reduced from 180 to 40 lb per hr.

After the aforementioned modifications, the gasifier and turbine performance was eminently satisfactory throughout the load and speed range. Rapid changes in load could be made with no difficulty, and the fuel consumption was considered satisfactory with a turbine of this efficiency. In addition, the engine would run equally well on either light diesel fuel or bunker "C."

The first complete overhaul and inspection revealed the major problems in free-piston engines were essentially those of any high-output diesel and perhaps more so because of the high temperatures and pressures in the combustion chamber. The main problems were as follows:

1. Cylinder wear.
2. Diesel piston-ring wear, scuffing, and breakage.
3. Compressor piston-ring wear, scuffing, and breakage.
4. Intake valve blade failures.
5. Oil leakage past seals resulting in valve deposits and airbox fires.
6. Piston-ring groove deposits.
7. High piston cooling oil consumption.

Many repairs and replacements of engine parts were required during the course of the performance tests. These difficulties were tolerated in order to complete the performance measurements. After engine output, fuel consumption, and control were found to be satisfactory, it then became necessary to establish the life of the engine by running an endurance test at full load on bunker "C" fuel.

Test work on the smaller free-piston air compressors had demonstrated that better piston-rings and modern heavy-duty lubricants made a marked improvement in ring life. Therefore, prior to running an endurance test, the lubricating oil was changed to a medium V.I. base oil with a Series 2 additive-level plus zinc dithiophosphate. In order to promote ring seating, new piston-rings were given a ½-deg taper and the cylinders were "lubrited."

It was also felt that with these and other improvements the gasifier rating could be increased 25% from 1000 to 1250 gas hp. A test goal was set up to run 1000 hr at full load on bunker "C" fuel with a minimum availabiilty of 85%.

An endurance test was started using light diesel fuel. The engine was run on a 24 hr day schedule six days a week. Each Monday morning before starting, the conditions of the piston-rings were observed through the intake ports. After 600 hr at full load the fuel was changed to bunker "C." This fuel was the same as that being supplied to the boiler house for making steam to heat our buildings. Each truck load of fuel was slightly different; however, a fairly typical analysis is shown in Fig. 9. The fuel used averaged 3½% sulfur, with some deliveries running as high as 4.2%.

The endurance test was continued at full load for 1100 hr to achieve the original goal of 1000 hr on heavy fuel. At the end of 1728 total hr the test was stopped and the engine dismantled for inspection. The availability of the engine at the end of the test was 91.2% at full load and 95% overall. That is, 95% of the available working hours the engine was running, and 91.2% of the available hours the engine was producing 1250 gas hp.

The 5% lost time was caused by casualties that necessitated shutting down the engine for repairs. The main cause of engine shutdowns was failure of intake valve blades. As shown in Fig. 10, the tips of the blades would fatigue and chip out. When the failures progressed below the sealing surface the result was back-flow and loss of power. The other two principal troubles were injector seal leakage and precombustion-chamber failures from cracks in the welds.

Inspection of the piston-rings, pistons, and cylin-

ders revealed excessive wear and scuffing. The copper lead of the wear bands had corroded and worn sufficiently to allow the steel piston-ring lands to contact the cylinders with consequent scuffing of the piston itself and damage to the cylinder.

Previous experience in Europe had indicated that airbox fires would occur at regular intervals. These fires do no damage other than stopping the engine by consuming the oxygen in the airbox. They are caused by lubricating oil deposits in the airbox, which come from leaky seals on the balance tube and connection linkage rods. Only two fires occurred during the entire test, neither of which stopped the engine. This remarkable performance, as compared to previous history, was attributed to the use of a modern heavy-duty lubricant. Oil leakage past the seals was appreciable, but the deposit formation was much less than that observed in European engines.

Results of this first long-term test were so encouraging, it was decided that redesigning many basic parts could result in much superior engine performance, and a second test goal was set at 5000 hr at a minimum of 85% availability.

Several hundred engineering changes were incorporated, and the new engine was designated the GM-14. The most notable of the changes were as follows:

1. Intake valves fatigue tested on a bench test rig

OIL SAMPLE REPORT

CHEMISTRY DEPARTMENT
RESEARCH LABORATORIES DIVISION
GENERAL MOTORS CORPORATION
DETROIT 2, MICHIGAN

C#377194

File No. 51.3375 (3M)
Cert. No. 377194
Work Order
Purchase Order

Sample No. 1215
Date Sub. 2-14-56
Date Reported 3-7-56
Priority: rush
Charge No: 19

Sample Of: Bunker C

Submitted By: W. C. Smith
Department: 61
Container No: 3200 gals.
Res. P.O. No:
Ordered By: 61
Refiner: Fuel Oil Corp.
New (X) Used ( )
Test No:

Analysis:

( ) Specific Gravity at 60°F	0.994	( ) Ash (%)	0.02
( ) API Gravity at 60°F	10.9	( ) Saponification No.	
( ) Viscosity at 100°F (CS) (SUS)		H_2O (Dist.) %	Nil
( ) Viscosity at 210°F (CS) (SUS)		H_2O Sediment %	0.05
( ) Viscosity at 122 °F (CS) (SFV)	149	Carbon, %	16.25
( ) Viscosity Index		Initial Boiling Point	345
( ) Neutralization No. (two phase)			
( ) TAN			
( ) TBN			
( ) Flash Point (°F)	240 ✓	Metals: (Spectrographic)	
( ) Fire Point (°F)	275	( ) Ba (%)	
( ) Pour Point (°F)	35	( ) Ca (%)	
( ) Conradson Carbon (%)		( ) P (%)	
( ) Ramsbottom Carbon Residue (%)		( ) Zn (%)	
( ) Sulfated Residue (%)		( ) Mg (%)	
( ) Chlorine (%)		( ) Al (%)	
( ) Sulfur (%)	3.71		
( ) Pentane Insoluble (%)			
( ) Benzol Insoluble (%)			
( ) Resins (%)			

Comments:

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Fig. 9—Typical heavy fuel analysis sheet

Fig. 10—Failed intake valve blade

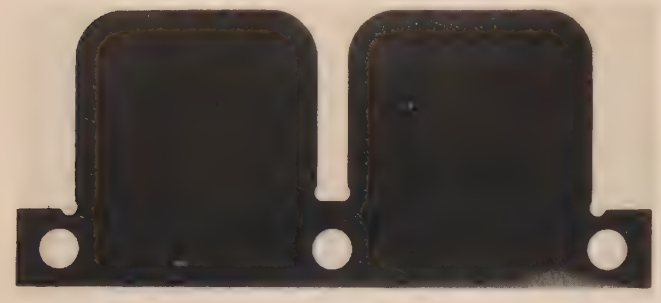

Fig. 11—Intake valve blade after 4500 hr on GM-14 engine

indicated that considerable improvement in valve life was possible. New valves were made up of aluminum segments instead of steel and valve blades made of a suitable spring stock correctly processed for their ultimate use.

2. Cylinders were given a surface finish specification in accordance with our diesel-engine practice and were "lubrited."

3. Piston rings processed in accordance with our diesel-engine experience were taper-faced, grooved, and ferrox-filled.

4. Bronze wear bands on the pistons were discarded because of the severe sulfur attack, and a special steel-backed aluminum alloy was substituted. The wear bands were grooved similarly to the piston-rings to reduce land width.

5. Precombustion chambers were redesigned and made from a more suitable material with better cooling provided.

6. Injector seals were made from a more suitable material.

7. Balance tubes, connecting links, and all sliding seal surfaces were hardened and lapped to a specified surface finish to reduce oil leakage.

8. Delivery valve plates were redesigned for water-cooling to reduce valve deposits and valve leakage.

A second endurance test was started in the fall of 1955 and was run as before at full load on bunker "C" fuel, 24 hr a day, six days a week. In May, 1956, the test was terminated at 4500 hr.

The GM-14 gasifier represents a marked improvement in engine durability over earlier free-piston engines. The three basic problems encountered in the first 1700-hr test were completely eliminated. No shutdowns were caused by injector seal leakage, precombustion-chamber failures, or intake valve failures.

Fig. 11 shows one of the valve blades removed after 4500 hr at full load. There is absolutely no evidence of fatigue failure.

The proper hardness and surface finish of the sliding seal surfaces had eliminated seal scoring and had reduced the oil consumption from 0.4 to 0.08 gal per hr. This greatly reduced the deposits in the airbox and prevented the occurrence of airbox fires.

Aluminum wear bands resulted in a marked reduction in wear band wear (or corrosion) from 0.010 to 0.0028 per 1000 hr.

Grooved and ferrox-filled piston-rings eliminated scuffing and greatly reduced the ring wear.

A close-up of the exhaust piston-rings is shown in Fig. 12. The most critical part of the engine is still the top ring; however, the latest changes have decidedly improved ring life. The No. 2 ring on both pistons, for instance, is in excellent condition with absolutely no signs of scuffing and has a very acceptable wear rate. It is expected that further changes now being tested will result in much improved top ring performance.

Perhaps it should be pointed out that piston-ring service in a free-piston engine is not quite as serious a problem as in more conventional diesels. The GM-14 engine can be shut down and pistons removed for a ring change in less than 1 hr.

Fig. 13 shows the power cylinder after the 4500-hr

Fig. 12—Close-up of exhaust piston rings

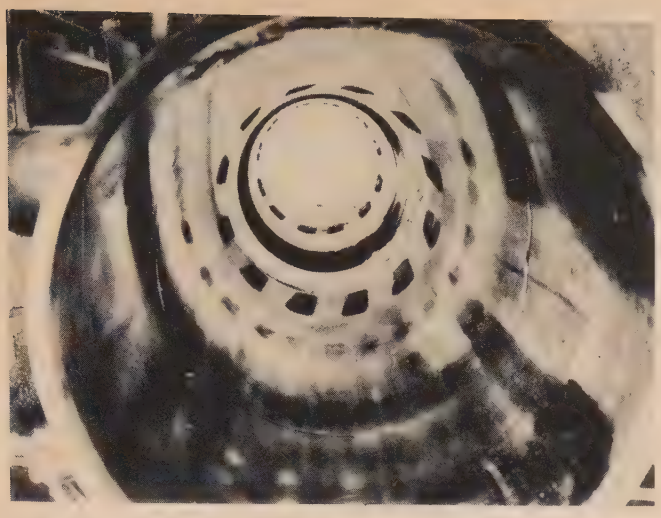

Fig. 13—Power cylinder after 4500-hr testing

test. One of the remarkable things about free-piston engines is the lack of port-clogging problems. As can be seen, there are no clogged ports, and the only deposits noted were on three intake ports located in a region of restricted airflow in the engine case. No deposits of any kind have ever been found in the exhaust ports.

Since development work on this free-piston engine began, we have increased the rated power output by 25%. The life of all of the critical parts has been extended until we can run a 4500-hr test with over 90% availability. The engine runs satisfactorily on heavy fuel with absolutely clean exhaust.

We are now confident that the GM-14 engine will be a contender for a place in commercial power generation in the United States and that continued development work will result in further increases in power, economy, and reliability.

Acknowledgment

Acknowledgment should be made of the cooperative efforts of the engineers of the Electro-Motive and Cleveland Diesel Divisions of General Motors as well as SIGMA and SEME in France for the rapid advance in free-piston-engine development over the past three years.

ORAL DISCUSSION

Reported by W. L. Kent

Union Oil Co. of California

R. J. Hooker, General Electric Co.: A reciprocating compressor is capable of producing high pressure. Then why is the turbine inlet pressure limited to 42 psi?

Mr. Flynn: Resonances in the system, which cause fluttering and breakage of valves, become a problem at pressure ratios above four. Also, the piston-rings in this engine need improvement before pressures can be raised. The critical point in the system is the exhaust pressure of the power cylinders. With further development work the pressure to the turbine inlet can be raised.

Capt. W. W. Brown, USN: These same resonance and valve breakage problems were encountered in two Navy-sponsored development programs on free-piston gasifiers. The main problem encountered at higher pressures is found in the combustion chamber, the top piston-ring being the critical component. I feel that the problem can be solved.

G. French, Union Pacific Railroad Co.: What is the vanadium, sodium, and calcium content of the bunker "C" fuel used for the 5000-hr endurance test on the free-piston engine?

Mr. Flynn: I do not have the vanadium analysis of the fuel. There has been no problem with the turbine. The engine was started up and shut down on a light diesel fuel.

If bunker fuel can be burned in a free-piston engine, why can't this fuel be burned in a gas turbine?

Mr. Hooker: The injection nozzles used in the Orion engine might be used advantageously to atomize fuel for a gas turbine.

Mr. Flynn: The free-piston gasifier as used in the GM-14 can be installed in units of one or more to give a range of powerplant sizes from 1000 to 18,000 gas hp.

Mr. French: Can the free-piston engine burn pulverized coal?

Mr. Flynn: It can if this fuel could be put into the engine, since the engine is quite insensitive to fuel because of the high compression ratio.

W. F. Ford, Continental Oil Co.: Does the GM-14 free-piston engine in its present state still require a Series 2 oil?

Mr. Flynn: We have not tried an oil of lesser compounding, since we do not want to run an oil testing program at this stage in the engine development program.

W. T. Von der Nuell, Garrett Corp.: How is the free-piston engine started?

Mr. Flynn: It is started by admitting 400-lb air through a quick-opening valve in the bounce cylinders. There is also another type of free-piston engine which is started by a spring which is compressed by a hand crank.

E. J. McLaughlin, California Research Corp.: Can you tell us about the smaller version of the free-piston engine which is installed in the passenger car?

Mr. Flynn: There are advantages in styling which result from the use of the free-piston gas-turbine powerplant. The compact free-piston gasifier permits lower hood lines, and the location of the gas turbine and transmission in the rear eliminates the driveshaft tunnel. Fuel consumption in the free-piston-turbine car has not yet been measured, but it is expected to be competitive with the piston engine. The free-piston gas turbine in combination with an afterburner has interesting possibilities as a passenger-car powerplant.

R. F. Bergstrom, Shell Oil Co.: What is the exhaust odor from the free-piston engine when burning bunker "C" fuel?

Mr. Flynn: The free-piston-engine exhaust is quite dilute, containing 80% air. The odor is that of sulfur dioxide from the high sulfur bunker "C" fuel.

M. E. Russell, Ethyl Corp.: What is the noise level of the free-piston car?

Mr. Flynn: The gas turbine effectively muffled the exhaust noise, so that the principal noise is that of the turbine reduction gears. The free-piston gasifier has the characteristic diesel-engine sound.

The combustion cylinder of the free-piston engine has a bore of 4 in. The bore of the air compressor cylinders is 11 in. Starting is by compressed air obtained from a compressor in the gas-turbine section.

G. L. Neely, Standard Oil Co. of California: Some railroads, in converting to diesel power, have scrapped the black oil tanks which formerly supplied the steam locomotives. In view of the ability of the railroad diesel engine and the free-piston engine to burn residual fuels, do you think that the railroads should save their black oil tanks?

Mr. Flynn: Yes.

C. G. A. Rosen, Caterpillar Tractor Co.: Residual fuels are apt to be the "garbage can" of the refineries. Diesel engines will not burn anything, and successful burning of residual fuels will depend on a consistent source of supply. Also, residual fuels can give more trouble than the operators will tolerate.

Mr. Flynn: Railroads are not buying fuel, but horsepower. High maintenance cost can offset the price of a cheap fuel. Thus, the railroads will select the fuel in each area which gives them the cheapest horsepower.

DARK DIETS

USE of residual fuels in medium- and high-speed diesel engines is discussed here. The objective is to improve overall economy of operation by using less costly residual fuels to get satisfactory performance.

Problems resulting from use of these residual fuels are discussed in light of economic effect on short-term performance of 4-stroke and 2-stroke diesel engines.

Two types of 2-fuel systems, the diesel fuel selector system and the diesel fuel blender system, have been developed for utilizing residual fuels in medium-speed engines.

THE motivation for development work in the field of utilizing residual fuels in diesel engines is in the interest of improving overall economy of operation. The cost of the diesel fuel consumed by the engine at the present time represents over 50% of the cost of operating a diesel locomotive in heavy freight service. Therefore, a small saving in the cost of the fuel should reflect a substantial reduction in the operating costs, provided this saving is not offset by increased maintenance costs. In general, the same is true for marine operations, and this is the principal reason for the widespread use in the shipping industry of motorships capable of burning residual fuel in their diesel engines.

Further, in the event of an all-out national emergency, distillate fuels might be in critically short supply due to the abnormal demands of jet aircraft, motor transport, agriculture, and other industries employing types of engines incapable of using residual fuels. Hence, knowledge of how to utilize residual fuels further is an important military consideration.

Historically, distillate diesel fuel has cost only about two-thirds as much as gasoline, but recently there has been a narrowing in the price differential between these two fuels. This has been due to the widespread increase in cracking facilities throughout the world and to the increased demand for distillate fuels for heating and for other uses. Thus, the economic motivation which led to the development of the small high-speed diesel engines, which was the wide differential in price between gasoline and distillate diesel fuel, no longer exists. However, the historical differential in price of roughly one to two between residual fuel and distillate fuel still exists to a large degree and is the economic force responsible for efforts to expand the use of residual fuels in medium- and high-speed rail and marine engines. No one can guarantee that this price differential will persist or what it may become during the next decade, but all can review the price differentials and trends that have existed in the past as shown in Fig. 1.

The use of residual fuels in diesel engines is as old as the diesel engine itself. Throughout the years, it has been common practice to operate large slow-speed marine and stationary diesel engines (especially those of European design) on black fuels of various viscosities. In many instances, distillate fuels have been used for starting up, shutting down, and maneuvering under part load. This practice in the shipping industry has resulted in satisfactory and economical operation, and at present the majority of commercial motor vessels throughout the world are using residual fuels in their diesel engines. However, in the United States, residual fuels are used less extensively in diesel power. This is particularly true in regard to medium- and high-speed diesel engines. It is "dark diets" for these medium- to high-speed marine and rail engines that will be discussed in this paper—engines not designed, except in certain instances, to operate on residual fuels. Recent developments discussed in this paper may enable a wide range of residual fuels to be used acceptably in these types of engines.

For many years the Standard Oil Co. of California has engaged in an investigation to develop methods for the utilization of less costly residual fuels in

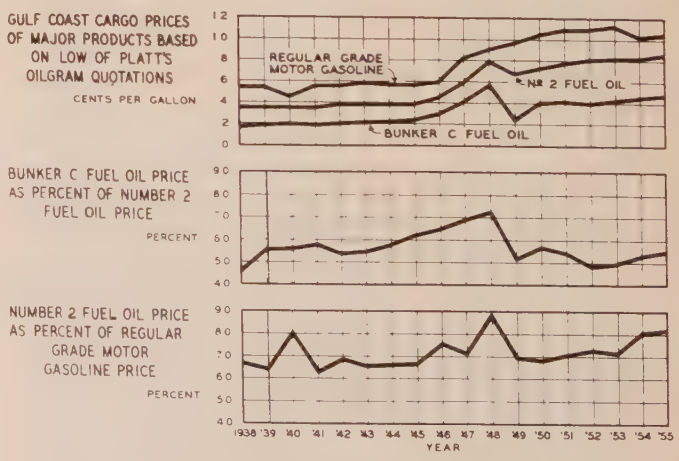

Fig. 1— United States petroleum statistics, 1938–1955

FOR DIESELS

G. L. Neely, E. F. Griep, and P. L. Pinotti,
Standard Oil Co. of California

This paper was presented at the SAE National West Coast Meeting, San Francisco, Aug. 8, 1956.

medium- and high-speed diesel engines. The problems to be solved included means to remove contaminants such as solid materials, oil-field brines, and water from the fuel. Filtration of the fuel on the engine and heating it to the proper viscosity at the injector for good atomization were found to be necessities. Increased piston-ring and cylinder-liner wear when using residual fuel occurred no matter what was done and seems to be the major unsolved problem remaining that apparently will require continuing study for its proper solution. In the meantime, the economic significance of increased wear, as a negative factor, will have to be evaluated for each instance where residual fuel is utilized in lieu of distillate fuel.

These studies also indicated that, at the present state of the art, it is inadvisable to operate medium- and high-speed diesel engines on residual fuels alone, as idle and part-load operation permits the formation of abnormal engine deposits and other attendant difficulties. This same situation may be true in larger engines as well, at least to some degree. However, with medium- to high-speed engines, as designed, there is much experimental evidence for the need for a fuel system wherein only distillate fuel is used at idle and part load, but which supplies residual fuel at the higher loads. The ideal system should, it seems, automatically blend on the engine residual fuel oil of any widely available commercial grade with distillate fuel to provide the viscosity grade of fuel at the injector demanded by the engine for each power condition. Data are presented in this paper to illustrate a diesel fuel blender system which was developed with the objective of supplying an engine with the least costly types of diesel fuel, automatically, that the engine could use, without introducing operating problems due to fuel incompatibility.

Short-Term Performance of Railroad Diesel Engines

In order to acquire certain basic information on how well certain diesel engines could digest "dark diets," short-term performance tests were conducted in both 2-cycle and 4-cycle railroad-type diesel engines, where the power output of the engine was absorbed by a loading device. The fuels used were, broadly speaking, ASTM No. 2 distillate fuel and ASTM No. 6 residual fuel and their blends. Table 1 gives comparison of fuel performance in these tests.

The purposes of these tests were to determine: (1) the most favorable engine operating conditions, (2) the change-over point based on power output from distillate to residual fuel, and (3) the ability of the engine to utilize the residual fuels of various viscosities throughout the operating load range as

Table 1—Fuels Used in Short-Term Performance Tests

	ASTM No. 2	ASTM No. 6
Flash, F	175	220
Viscosity at 100 F, SUS	37–39	6000–7400
Ash, %	None	0.04
Sediment and Water, %	Trace	0.2
Pour F	−5	+30
Carbon Residue, %	0.002	14–17
Sulfur, %	0.40–0.95	0.90–1.30
Distillation, F		
Starting Temperature	360	
10%	450	
50%	526	
90%	600	
End Point	675	

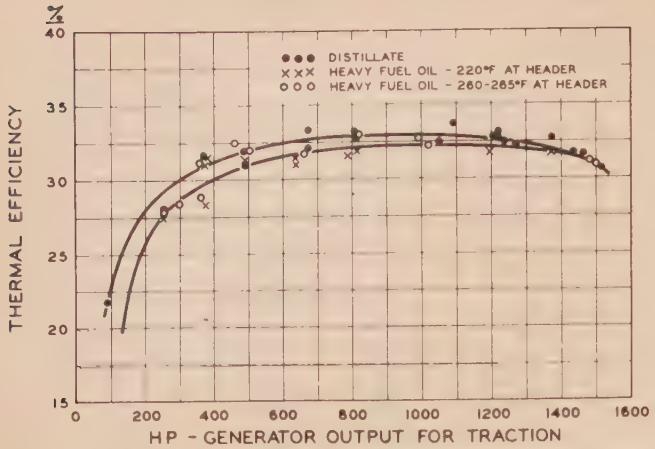

Fig. 2—Relation between thermal efficiency and generator output for traction

FOUR STROKE CYCLE ENGINE

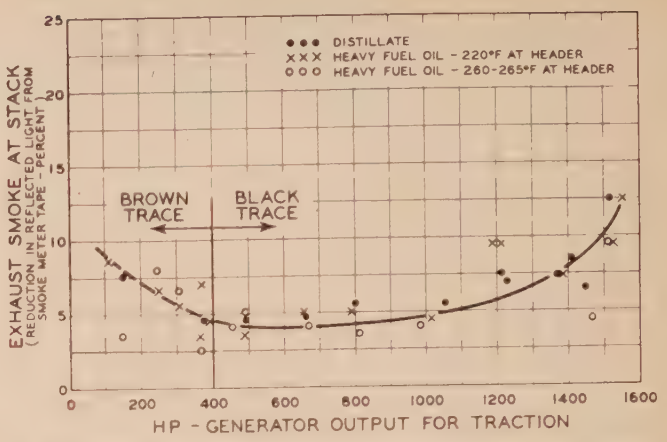

Fig. 3—Relation between exhaust smoke and generator output for traction

measured by thermal efficiency and exhaust smoke.

Four-Stroke-Cycle Engine—Short-term performance data secured in a medium-speed, 4-cycle engine indicated that:

1. Thermal efficiency with ASTM No. 6 residual fuel is comparable at high load to that observed while operating on ASTM No. 2 distillate fuel. However, the thermal efficiency becomes progressively poorer as the engine load is decreased, as shown graphically in Fig. 2.

2. Exhaust smoking with ASTM No. 6 fuel above 25% load factor was comparable to that observed with distillate fuel; however, below 25% load factor, the smokemeter trace changed from black to brown, indicating that unburned fuel was being emitted from the exhaust stacks, as shown in Fig. 3.

3. Peak cylinder pressures and exhaust temperatures while operating on ASTM No. 6 fuel compared favorably with those obtained while running on distillate diesel fuel, as shown in Figs. 4 and 5.

4. Ignition delay, measured electronically, while using the ASTM No. 6 fuel, was greater throughout the entire load range than while using ASTM No. 2 fuel as shown in Fig. 6. The maximum difference in ignition delay occurred at approximately 25% load factor.

5. Increasing the temperature of the fuel to the injectors above 220 F had no apparent effect on engine performance as measured by these tests.

Two-Stroke-Cycle Diesel Engines—The short-term performance tests in a 2-cycle railroad diesel engine indicated that:

1. The engine is more tolerant to the ASTM No. 6 fuel at high-load factor than at intermediate or light loads. Data obtained indicate that at higher loads this engine will satisfactorily deliver its power with a 1000 SUS at 100 F blend of ASTM No. 6 and ASTM No. 2 fuel; however, as the load on the engine is reduced, the tolerance of the engine for ASTM No. 6 fuel is also reduced. This is illustrated in Fig. 7 where it may be observed that the thermal efficiency observed at higher loads with the 940 SUS at 100 F blend of the ASTM No. 2 and No. 6 fuels compared favorably with that observed with the ASTM No. 2 fuel. However, as the load on the engine is reduced, the percentage of the ASTM No. 6 that can be utilized, with thermal efficiency comparable with that of the ASTM No. 2, is reduced.

This phenomenon is illustrated more clearly in Fig. 8 where the brake specific fuel consumption (bsfc) is plotted against time. It is of interest to note that in section D, Fig. 8, the bsfc remains constant with time at 65% rated load using the 940 SUS at 100 F blend of ASTM No. 2 and No. 6 fuel. However, as the load was reduced to 50%, the bsfc began to increase. The increase in bsfc was even more pronounced as the load was reduced to 35%. Similar phenomena were observed with the 379, 79, and 49 SUS at 100 F blends of the ASTM No. 6 and ASTM No. 2 fuels, as seen in sections A, B, and C of Fig. 8.

The data presented in Fig. 8 suggests that:

1. Carbon "horns or trumpets" are formed on the injector tips when operating on blends of ASTM No. 6 and ASTM No. 2 fuels at reduced loads, and in some instances these horns may break off and reform.

2. These horns or trumpets form, at least to some extent, on the injectors even with a 49 SUS at 100 F blend of ASTM No. 6 and ASTM No. 2 fuel.

3. The formation of these horns disrupts the atomization and the spray pattern resulting in increased smoking and a reduction in thermal efficiency.

4. As the engine load is decreased, the pressure and temperature inside the combustion chamber decrease, and an engine load zone is reached below which the temperature and pressure in the cylinder are too low to condition the ASTM No. 6 and ASTM No. 2 fuel-oil blends preparatory for the combustion process in the time allowed.

Fig. 9 shows the "trumpets" formed on the injec-

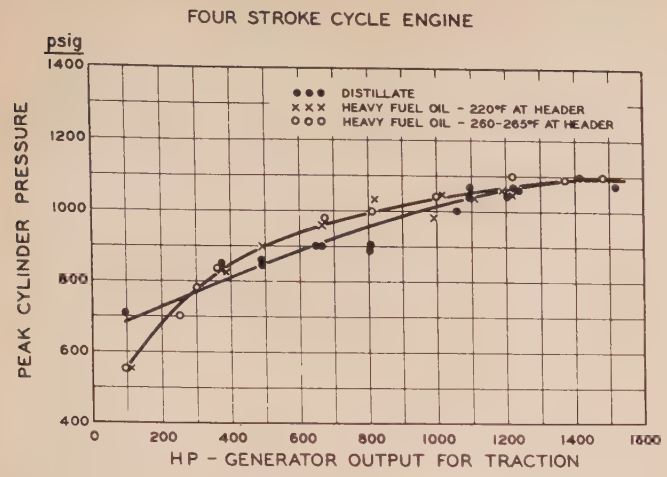

Fig. 4—Relation between peak cylinder pressure and generator output for traction

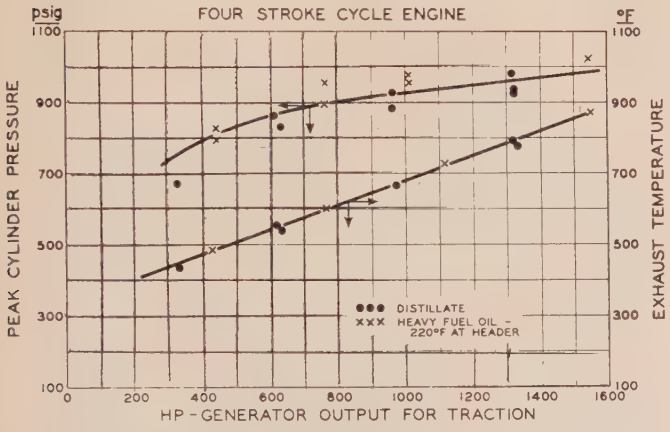

Fig. 5—Relation between peak cylinder pressure and exhaust temperature and generator output for traction

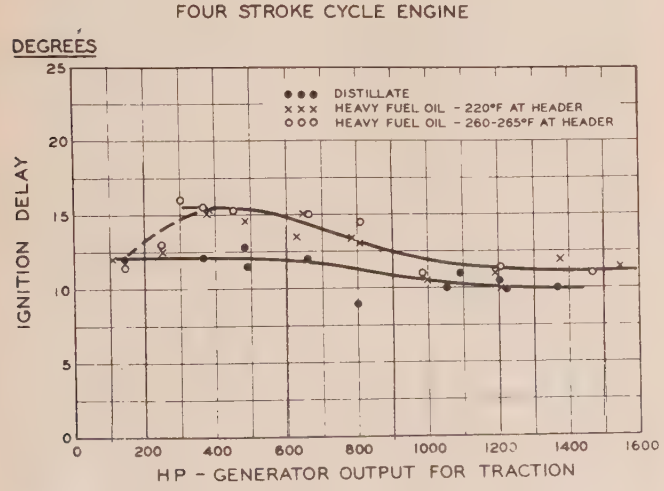

Fig. 6—Relation between ignition delay and generator output for traction

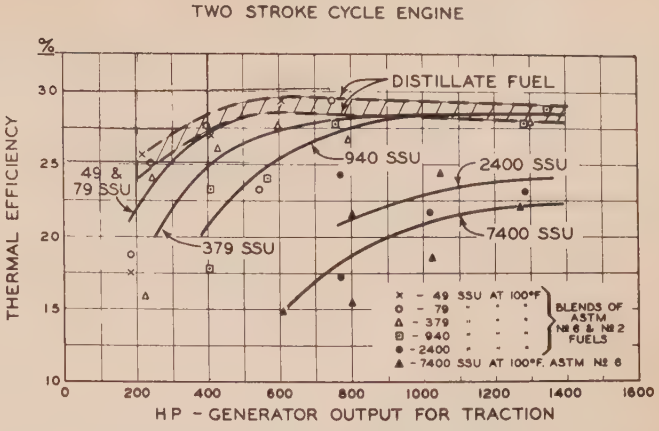

Fig. 7—Relation between thermal efficiency and generator output for traction for various blends of ASTM No. 6 and No. 2 fuels

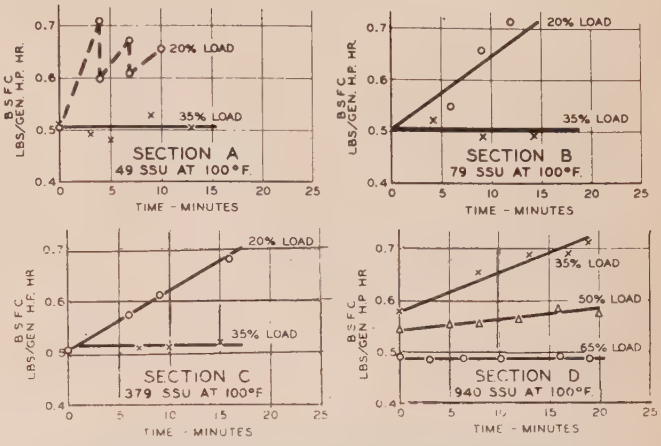

Fig. 8—Relation between engine performance and time for various engine load and residual fuel blends

Fig. 9—Fuel injector tip deposit resulting from light load operation

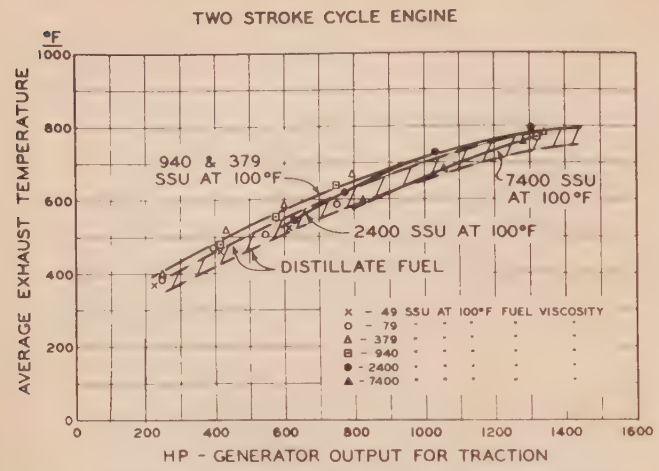

Fig. 10—Relation between exhaust temperature and generator output for traction

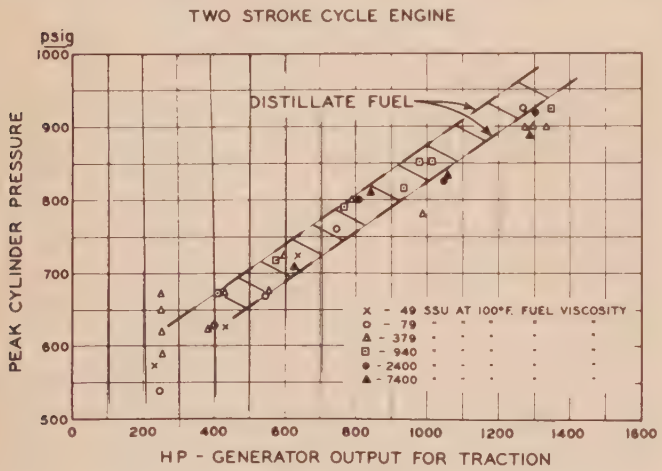

Fig. 11—Relation between peak cylinder pressure and generator output for traction

Fig. 12—Representative ring-belt deposits

tors after approximately 30 min of operation with the 940 SUS at 100 F fuel at light loads. These trumpets were removed from the combustion chamber and reconstructed.

The exhaust temperatures and peak cylinder pressures, observed in the 2-cycle railroad diesel engine when operating on blends of ASTM No. 6 and ASTM No. 2 fuels compare favorably with those observed with the ASTM No. 2 fuel alone. This is shown in Figs. 10 and 11 where generator output is plotted against exhaust temperatures and peak cylinder pressures respectively. These data were obtained in a 2-cycle railroad diesel engine.

The data presented show that with current railroad engine designs, a 2-fuel system is required. This would necessitate a duplication of storage and dispensing facilities which are costly to install and require maintenance. In an effort to utilize a selected residual fuel under all operating conditions and to avoid the necessity of using any distillate at all, the use of additives and also variations in engine operating conditions and techniques were investigated. Table 2 gives some of the schemes investigated and the results obtained.

Fig. 13—Representative piston showing deposits from blended fuels

Some improvement in the performance at light load was observed by increasing the injection pressure in that this reduced the load threshold for satisfactory operation on residual blends. However, this did not eliminate the need for distillate fuel under idle conditions; the critical load threshold with residual fuel was lowered from 25% to approximately 10% rated load before engine fouling was observed, as measured by a reduction in the thermal efficiency. Operation at loads below 10% was not satisfactory with the 379 SUS at 100 F residual fuel blend using these modified injectors. It is not known whether or not injectors can be made that will permit use of heavy residual fuel oils at idle operation and which also will be suitable for full-load operation. Even if such injectors could be built to operate an engine of current design on residual fuel of only 300 SUS at 100 F, a 2-fuel system would probably provide further economies. Also, distillate fuel is desirable for other reasons, including purging of lines when shutting down and for starting and warming-up periods.

The data obtained indicate that (1) without changes in engine design, a 2-fuel system is necessary for satisfactory operation of this class of diesel engine on residual fuels, and (2) the fuel injector is the engine component that is most critical operationally when using residual fuels.

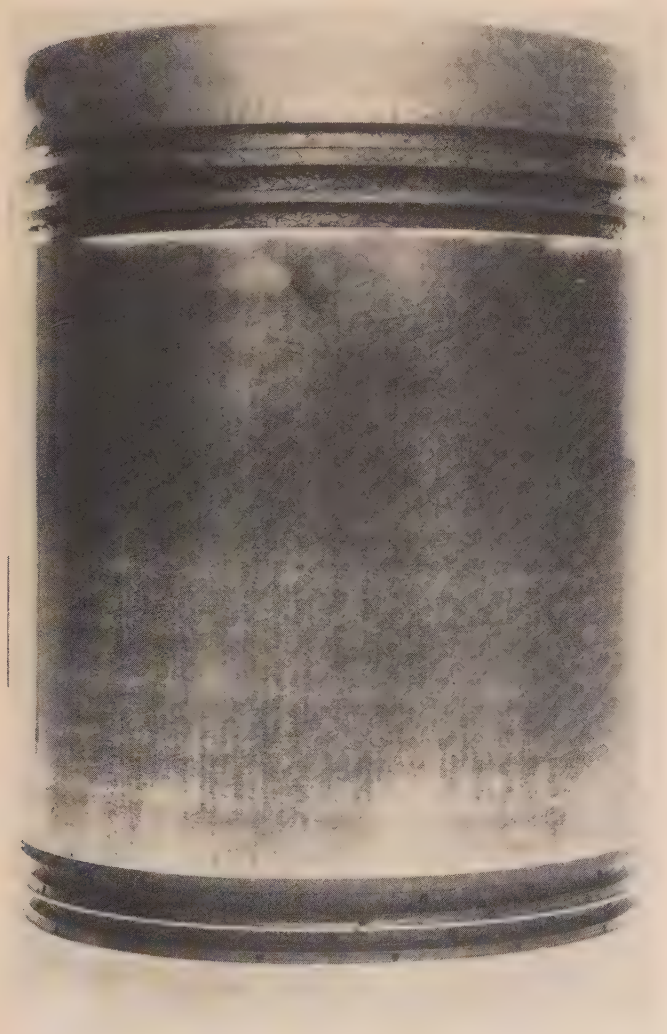

Fig. 14—Piston showing deposits resulting from use of ASTM No. 2 fuel

Table 2—Effect of Additives on Engine Operation

	Improvement in Thermal Efficiency at 25% Load
Use of Additives in Fuel Stream	
Cetane Improvers	None
Combustion Accelerators	None
Moisture	None
Volatile Hydrocarbons	None
Use of Additives in the Airstream	
Steam	None
Portion of Exhaust	None
Liquefied Petroleum Gas	None
Other Volatile Hydrocarbons	None
Engine Operation	
Split Engine (Four Cylinders Firing)	None
Heated Intake Air	None
Increased Injection Pressure (Plugging Three of the Six Holes in the Injectors)	Some Improvement

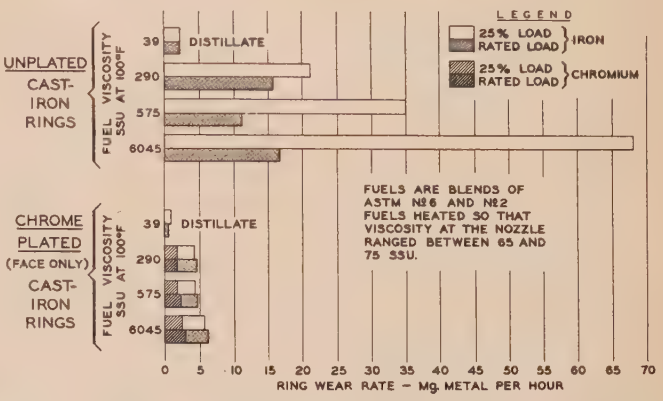

Fig. 15—Effect of chromium plating and load on wear for distillate and residual fuels

Residual Fuel Effect on Engine Operation

The engine deposits, when operating with residual fuel blends and using the 2-fuel system, compare very favorably with those observed with distillate fuel operation. The deposits on the injector tips, exhaust valves, piston skirt, ring groove, and belt area are comparable to or lower than those observed with distillate fuels in the same service. Comparative piston deposits for 1-year operation are shown in Figs 12, 13, and 14.

Engine wear while operating with residual fuels has historically been found to be higher than that observed with distillate fuel. When using residual fuels, engine wear with conventional cast-iron liners and cast-iron rings was found to be approximately two- to threefold higher than normally obtained while operating with distillate fuels in 12 months of typical railroad service.

Some interesting results were obtained showing the effects of type of fuel, operating conditions, and chromium plating on piston-ring wear when investigated in the laboratory in an engine of different design but comparable in size and speed to those used on the railroad. The data were obtained with radioactive piston rings and are shown in Fig. 15. It is

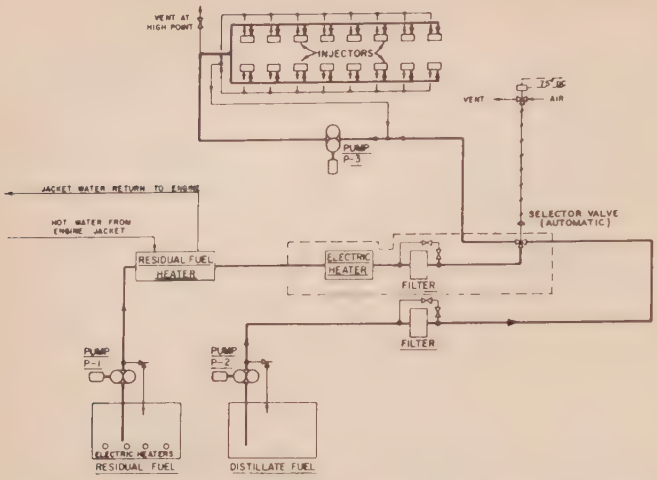

Fig. 16—Two-fuel operation for diesel fuel selector system

interesting to note that in this particular engine, increasing the ratio of ASTM No. 6 to ASTM No. 2 fuel increases engine wear at light load while using the conventional cast-iron rings and liners. Under high-output conditions, however, although the wear rate was higher with fuels containing residuum, increasing the ratio of ASTM No. 6 to ASTM No. 2 fuel had no effect on engine wear.

Chromium plating of the rings reduced the wear rate at rated load to about one-third of that obtained with unplated rings. However, engine load had no marked effect on engine wear while using the chrome-plated rings. This would indicate that changes in the metallurgy of the liners and/or rings will have an appreciable effect on the wear obtained with residual fuel.

Summary of Engine Data

From the foregoing data, it appears that medium-speed diesel engines, as presently designed, cannot be operated satisfactorily on residual fuels only, over the full-load range. Testing in this field covering a period of many years has shown no single reversal of this finding regardless of the engine or the residual fuel used. Data have been presented showing that the thermal efficiency of the engines investigated decrease with decreasing load if residual fuels are used where the viscosity of these fuels exceed certain maximum limitations. Furthermore, the use of residual fuels under idle and light-load conditions may result in markedly increased wear, at least in the case of cast-iron rings.

While it is acknowledged that much may be done mechanically and operationally, such as by the use of high injection pressures, higher jacket temperatures, and special injection systems such as multiple injectors or air injection, no completely satisfactory means of operating the medium-speed engines wholly on residual fuel, even of selected types, has been disclosed. Satisfactory operation embodies satisfactory engine performance and also reliable operation wherein the fuels are burned economically at maintenance costs comparable to those obtained with distillate fuels.

Two-Fuel Systems

In developing means for utilizing residual fuels in medium-speed engines economically, certain desirable basic requirements for a 2-fuel system were established. These are:

1. Residual fuels used should preferably be run-of-the-mill, heavy commercial grades rather than specialized formulations.
2. The heaviest residual fuel and the minimum amount of added distillate fuel consistent with satisfactory engine operation should be supplied to the engine at each particular load condition.
3. There should be no requirement that the residual fuels be fully compatible with the distillate fuel used.
4. Increased maintenance costs due to the use of residual fuels should be minimized.

With these objectives in view, two types of 2-fuel systems were developed by the Standard Oil Co. of California. These are known as the diesel fuel selector system and the diesel fuel blender system.

Diesel Fuel Selector System

The diesel fuel selector is an "on-and-off" system wherein the fuel supplied to the engine is changed automatically at a predetermined load level from all distillate fuel at low loads to a preselected residual fuel at higher loads, and back to distillate again when the load is reduced to the predetermined fuel change-over load level. This system gave satisfactory operation and complied acceptably with requirement 3

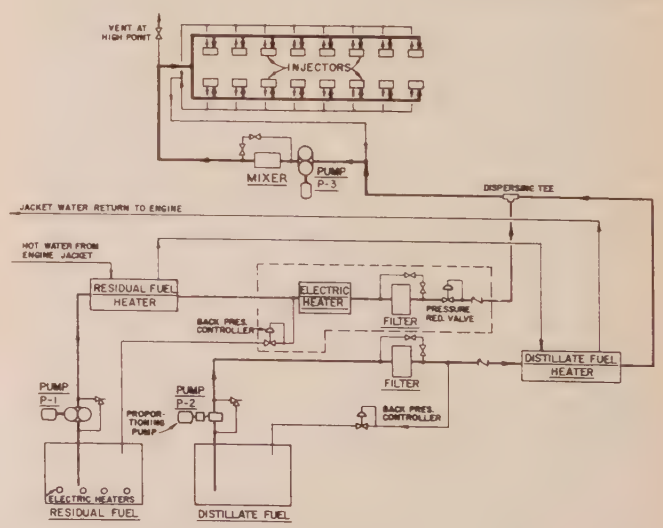

Fig. 17—Two-fuel operation for diesel fuel blender system

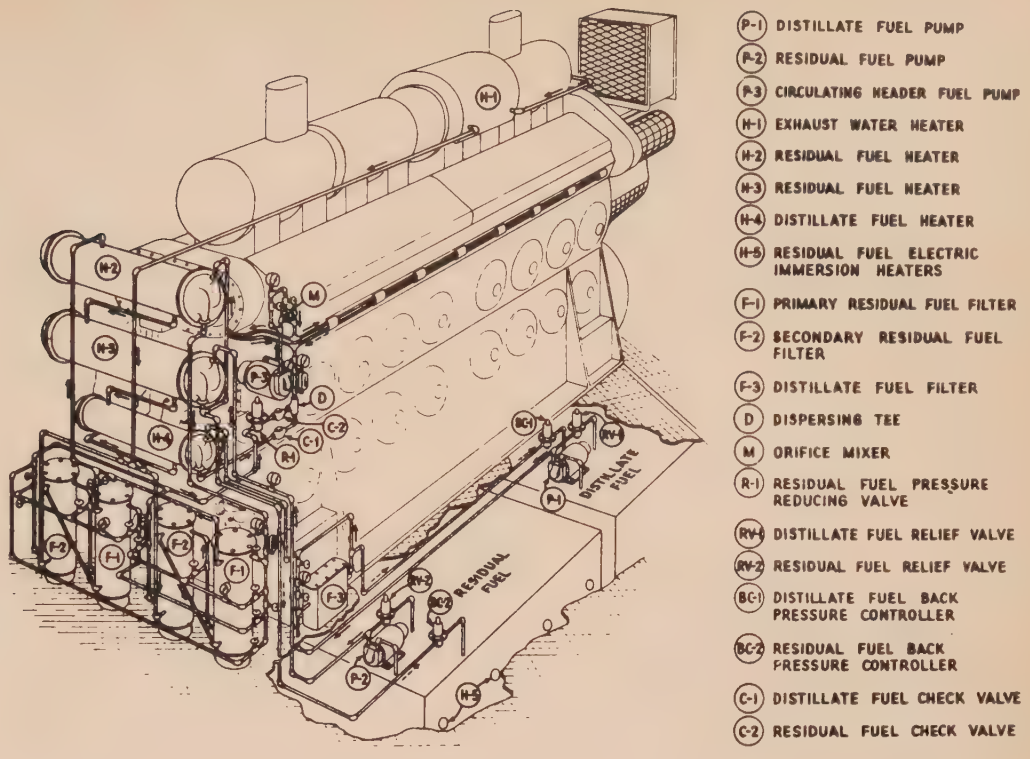

Fig. 18—View of diesel fuel blender system

listed above, but did not meet requirements 1 and 2. Therefore, it was felt that a better system should be developed. The diesel fuel selector system is shown schematically in Fig. 16.

Diesel Fuel Blender System

The diesel fuel blender is a variable blending system in which distillate fuel is supplied continuously at a preselected limited volume for idle and low-load operation. When the fuel demand of the engine exceeds this quantity, the fuel supply is supplemented automatically as required by admitting residual fuel along with the distillate in the fuel line going to the engine. Thus, when the engine is operating in the higher load range, the fuel is predominantly residual. At medium loads it may be composed of about equal proportions of residual and distillate, and at low loads and idle, only distillate fuel is supplied.

A schematic flow diagram for the diesel fuel blender is shown in Fig. 17. It includes two fuel tanks, one for the distillate and one for the residual fuel. Means are provided for heating these fuels, principally by using cooling water or waste heat available on the engine, and in certain installations this is supplemented by electrical heat.

This system employs two primary fuel pumps, one a distillate fuel pump of a predetermined fixed capacity sufficient to supply the engine with its full requirement of fuel at idle and low loads. The other pump supplies residual fuel to the system, and as fuel requirements rise with increase in load, the pressure in the fuel system is reduced. The supplementary amount of fuel required by the engine is then supplied by the residual fuel pump. This is accomplished automatically by differential pressures in the fuel system with the two fuel pumps, together with the necessary pressure relief valves as shown in Fig. 17. At full load approximately 75% of the fuel consumed by the engine may be supplied by the residual fuel pump. This system permits the engine to select the fuel it can utilize best depending on the load at which it is operating. A perspective view of one arrangement of the diesel fuel blender is shown in Fig. 18. As actually installed, however, this system is much more compact than illustrated.

Compatibility of Residual and Distillate Components

It is well known that compatibility of blends of residual and distillate fuels becomes more critical as the viscosity of the blend is reduced. With both the diesel fuel selector and the diesel fuel blender systems, incompatibility problems for mixtures of residual and distillate fuels appear to be met by utilizing a circulating header. In this design, a circulating pump is located in the fuel header, and the fuel return lines from the injectors are returned to this pump. In addition to alleviating sedimentation as a result of incompatible fuels, the rapid circulation of the fuel mixture in the header prevents any accumulation of gaseous bleed back from the combustion chamber through the injection system.

A possible disadvantage of the circulating header

is the volume of fuel remaining within the header at all times. This may represent about 2 qt. At idle, this quantity of fuel is sufficient to last approximately 10 min. Thus, if the engine has been operating under full load and then is returned immediately to idle operation, the viscosity of the fuel would be initially—say 750 SUS at 100 F—but would be rapidly diluted by the distillate. There is no apparent means of entirely overcoming this possible disadvantage when using the circulating header, but in service this does not appear to introduce operational problems provided the lowest possible circulating header volume is utilized.

In spite of the fact that the diesel fuel selector system includes the circulating header principle, it fails to comply with all of the basic requirements for a 2-fuel system set forth, because with the selector system the viscosity of the residual fuel as purchased cannot exceed certain maximum limits. This limitation is very important in the economic utilization of residual fuels. The diesel fuel blender system, however, complies more fully with the basic requirements listed above for the use of residual fuels, and, therefore, this discussion hereinafter will be limited to the diesel fuel blender system.

Practical Operation of Diesel Fuel Blender

The service tests on a railroad diesel locomotive described in Fig. 7 wherein thermal efficiency versus viscosity of the fuels for various horsepowers is depicted, shows that as the load increases a more viscous fuel can be fed to the engine. The diesel fuel blender's actual performance in this regard is shown in Fig. 19 wherein the fuel viscosity is plotted against throttle position. It will be noted that the viscosity rises sharply, though not vertically, after going through throttle position No. 4, representing approximately 400 hp. A curve is shown which indicates the maximum fuel viscosity that should be used for the various throttle positions. This curve is based on data such as those shown in Figs. 7 and 8. What this relationship means is that when the engine is operated for a considerable period of time at a viscosity above and to the left of the safe-operating-range curve, the engine performance will eventually become unsatisfactory due to drop in thermal efficiency and other factors. Thus, operating with fuels whose viscosities exceed those of the safe-operating-range line will negate the purpose of utilizing residual fuels, which is to reduce the overall operating costs.

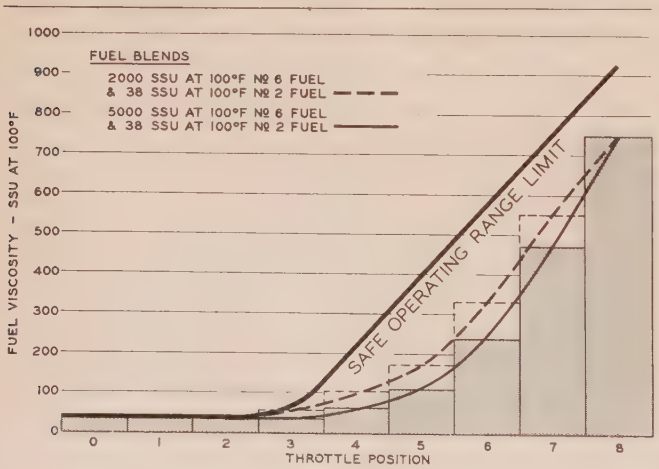

Fig. 19—Viscosity of fuel blends at various throttle positions

The most significant part of Fig. 19 is that the actual viscosities provided by the diesel fuel blender for various throttle positions closely parallels the safe-operating-range curve, and thus, throughout its entire zone of operation on residual fuel, provides a consistent factor of safety, and this is done by a simple pressure differential arrangement as previously described. Actually, the diesel fuel blender could be operated in such a way that the viscosities of the fuel for various throttle positions would coincide almost exactly with the safe-operating-range curve, eliminating the factor of safety provided in Fig. 19. However, this is felt to be an unnecessary attempt to derive the last portion of fuel economy when using residual fuels. Furthermore, variations in injection hole sizes will affect the atomization of the residual blends and the location of the safe-operating-range curve. Also, different types and sources of both distillate and residual fuels and the use of fuel additives may modify the location and shape of the safe-operating-range curve.

With the diesel fuel selector system, it is obvious from Fig. 19 that at the medium throttle positions, preselected light-viscosity residual fuels which are low enough in viscosity to give continued safe operation in this power output zone do not appear, at the same time, to be heavy enough at maximum load in that they fail to take advantage of the heaviest, cheapest fuel which the engine can utilize, and this maximum load operation embraces the greater volume of the fuel consumed by the engine. Thus, any 2-fuel "on-and-off" system such as the diesel fuel selector has the inherent disadvantages described above, and any attempt to obtain greater operating economies at maximum loads by using more viscous residual fuels increases the possibility of adverse performance at medium loads.

Economics of Railroad Residual Fuel Use

Other problems relating to the satisfactory performance, without incurring undue maintenance costs of engines utilizing residual fuels, are being further evaluated under actual railroad operating conditions. These problems include:

1. Two-fuel storage systems at the railroad terminals.
2. Two-fuel tankage on the locomotives.
3. Additional heat for the residual fuels as compared to distillate fuels to facilitate pumping.
4. Additional filtration requirements and costs.

5. Additional engine wear, particularly of rings and cylinder liners.

The problems seem to be of relatively minor economic importance in relation to potential savings in fuel cost, with the exception of increased cylinder-liner and ring wear. There is no question but that a portion of the savings attainable are lost in increased engine maintenance.

No data have been obtained in regard to these additional operating costs that may be described in terms of actual costs; however, data have been computed comparing the fuel costs applicable to actual road engine operation with the diesel fuel selector system, the diesel fuel blender system, and the conventional distillate fuel system. Fig. 20 shows the results obtained. This figure shows fuel cost reduction in per cent for the diesel fuel selector system when using a 300 SUS at 100 F residual fuel (and a second case where a 500 SUS at 100 F residual fuel is used) in the No. 5 throttle position and above. Fig. 20 also shows the savings obtainable with the diesel fuel blender using either a 300, 500, or 750 SUS at 100 F residual fuel blend at the No. 8 throttle position and lower viscosity blends, respectively, at lower power outputs. It is interesting to note that the potential savings with the diesel fuel blender system are consistently greater than those obtainable with the deisel fuel selector system, and this is done under operating conditions which provide a substantial factor of safety for all load conditions. The greater the percentage of residuum in the fuel, the greater the saving. Actually this system, with these particular fuels, would permit operating this engine with up to 1000 SUS at 100 F fuel at the No. 8 throttle position, and thus provide greater fuel economies than those illustrated. The net savings attained will depend on the cost and properties of the distillate and residual fuels available in the various localities, less increased maintenance and other costs, including that represented by the fuel required to heat the residual fuel itself, particularly in cold weather.

The primary problem of engine wear when using residual fuels and its effect on net operating economies is being actively investigated. Even at present levels of wear and at present residual fuel prices as compared to distillate fuel, the economies appear to favor the diesel fuel blender system over conventional distillate diesel-fuel-burning engines. Other additional operating costs encountered with the 2-fuel systems, such as heating and filtration, probably will not critically disturb the favorable savings indicated.

A patent has been granted on features of both the diesel fuel selector and diesel fuel blender systems. Additional 2-fuel developments are covered by pending patent applications.

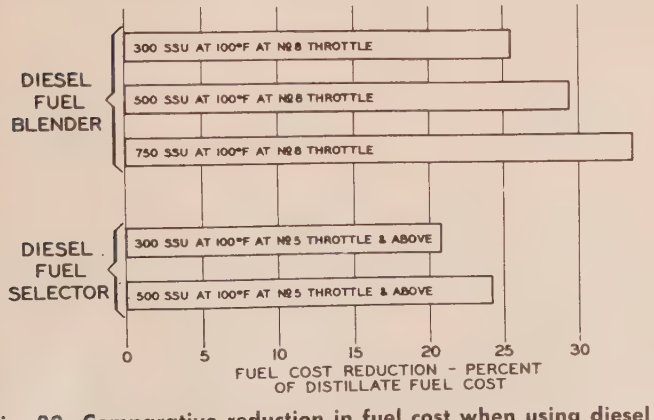

Fig. 20—Comparative reduction in fuel cost when using diesel fuel blender or diesel fuel selector

DISCUSSION

Wants Further Work on Using Heavy Residual Fuels

—Capt. W. W. Brown
U. S. Naval Engineering Experiment Station

I WOULD like to compliment Mr. Neely and his colleagues of the Standard Oil Co. of California on the very creditable job they have done in serving a "dark diet for diesels" in such fashion that the engines themselves appear to enjoy the diet. They are to be commended for the unique concept of the fuel blender system and the manner in which they have adapted it to optimum operation of the engine throughout its load range. I am sure that no one would insist that this is the only answer to burning residual fuels in medium-speed engines, but I feel certain that all will agree that it is a great advance in that direction.

Admittedly, the burning of heavy fuels in diesel engines is not new; in fact, it was an essential ingredient of the original work done by Rudolf Diesel himself and, indeed, has been a major point of attractiveness in selling the diesel engine to the customer. Certainly, for the many years that diesels have been in use, heavy fuels have been used to a considerable extent, particularly in the air injection type and in the large slow-speed marine and stationary engines. There have even been isolated and successful efforts in the burning of residual fuels in medium-speed engines. From the combustion standpoint, such success has generally been attained with the precombustion type of chamber.

It is rather recent that added impetus and interest has arisen in the burning of residual fuels in medium-speed diesels. That this interest has blossomed out on such a wide scale is, in the main, due to the large contribution fuel makes toward the operating costs of railways. The economics of the situation have encouraged the railways to take steps to make the most of the price differential between premium distillate fuel and the heavy residual-type fuel.

The technique described by the authors is recognized as one tailored to adapt a residual refinery product to existing engines which were basically designed for a diesel fuel having premium characteristics. That they have done an excellent job in this respect is quite evident. It is likewise evident that they have followed that highly desirable and rather rare approach of, in effect, listening to what the engine tried to tell them and concentrating their line of endeavor accordingly, rather than determinedly trying to force preconceived ideas on the engine which the engine could neither readily nor comfortably accommodate.

This paper, along with a search of other literature on this subject, prompts several observations that I would like to make, which might be helpful, and which appear quite justified. First, I would like to point out that there is a great lack of fundamental knowledge on combustion and the burning of heavy fuel and the relative effects of the many factors influencing such actions. We, therefore, have a very complex picture to deal with, one which requires clear and careful documentation of data, along with a

scientific approach toward solving the problems encountered, or we will never attain concrete answers of real value to the petroleum engineer, the engine designer, or the operator. Likewise, we must appreciate the fact that every type and model of engine is as different in character as individual people and that what is good for one engine may not be so for another. As a word of caution, I think it's well to point out that we should not use with great exactness data and experience gained in the burning of heavy fuels in very large slow-speed engines and attempt to apply it directly toward the use of heavy fuel in medium-speed engines. Rather, such data should be used to indicate likely trends or problems that we might encounter in the medium-speed engine field.

I believe that the main problem confronting us from here on is that of high rate of cylinder wear. The fact that we have it to a greater degree than one should economically tolerate has already been shown, but how we can combat it or offset it economically apparently has not been fully demonstrated. This problem will require much thought and effort on the part of all of us, including the petroleum industry, before the best solution is found.

There appears to be considerable confusion on this matter of burning heavy fuel; as an example, I would like to cite the age-old argument as to what causes cylinder wear. Some authorities consistently maintain that it is abrasive in character, while other equally competent authorities insist that it is corrosive. Surely we can do a quicker, better, and more economical job in solving this problem if we definitely determine what actually causes cylinder wear. Another example of the lack of scientific approach on this subject is the apparent discrepancy in findings regarding a change in timing. Some people indicate that timing should be advanced while others insist that it should be retarded. Possibly they are both right in that one is basing his argument on one set of conditions, while the others apply to a different set. Here again, differences in fuels, engines, and operating conditions have important bearings on results attained.

I would like to ask Mr. Neely the following questions:

1. Was your research work over the years done with only two engines, or were several makes and models of both 2-cycle and 4-cycle engines used?
2. What was the vanadium content of the fuel used in such work?
3. How well does the fuel blender use residuals having several hundred seconds difference in viscosity?
4. What change in timing, if any, has been made in the railroad engines using the fuel blender?
5. How would you recommend combatting the high wear rate resulting from burning of residual fuels?
6. Can we count on the oil industry's actively tackling such a problem by the treatment of fuel and lube oil?

We at the Naval Engineering Experiment Station have done some work on the burning of residual fuels in medium-speed engines and, as far as we've gone, generally concur with the findings given in this paper. We are convinced that it is highly essential in the interest of reducing cylinder wear to: (1) thoroughly centrifuge and filter the fuel and (2) maintain elevated coolant temperature. Furthermore, we feel that the inevitable increase in liner wear rate can be offset largely by the use of chromium-plated liners and possibly by advances in the metallurgy of cast iron. During World War II, we made extensive use of chromium-plated liners in bores of 5 in. and larger and have accumulated experience with more than 100,000 such liners in service. The average wear rate that we have experienced with chromium-plated liners has been about 25% of that encountered in cast-iron liners using premium straight-run distillate fuel. With residual fuels, it is possible that an even greater improvement in wear rate might be expected in using chromium-plated liners in lieu of conventional cast-iron liners. Our experience with rings has been comparable; namely, that cast-iron rings used in conjunction with chromium-plated liners show a lower wear rate than when used with cast-iron liners. It is possible that, when burning residual fuel, the wear rate of cast-iron rings used with chromium-plated liners may be considerable and even more than is experienced in burning distillate fuel with cast-iron liners. I would like to emphasize, however, that our experience on wear has been only with distillate fuel.

As another thought, I would like to point out that in this field of endeavor different customers will have different problems. For instance, the ocean-going marine trade must be concerned with wide variations in fuel quality occasioned by a multiplicity of sources and suppliers throughout the world such that consistency in technique in burning heavy fuels may be quite impossible. Likewise, the compatibility of various residuals becomes an important factor. Apparently these problems are minimized in such service as the railroads, where a minimum number of petroleum sources and suppliers can be enjoyed. As an additional question for Mr. Neely, I would like to ask him what answers does the petroleum industry have for marine users in this respect?

In closing, I would like to mention two developments that aid the internal-combustion process toward better digestion of heavy fuel. One is turbocharging, which makes an engine less sensitive to fuel quality. The other is a comparatively new internal-combustion engine, the free-piston gasifier, which even in the extremely small bore appears to be very insensitive to fuel characteristics.

Note: The opinion and assertions contained herein are the private ones of the writer and are not to be construed as official or reflecting the views of the Navy Department or the naval service at large.

Authors' Closure To Discussion

WE would like to thank Capt. W. W. Brown, USN, for his preparation of comments on our paper. We would now like to answer his questions.

The research work on the utilization of residual fuels was conducted in both 2-cycle and 4-cycle engines. The bore of these engines ranged from 4.25 to 8.5 in. for the 2-cycle, and 4.25 to 12.25 in. for the 4-cycle engines.

The vanadium content of the fuel used in our work with residual-type fuels was low. We do not have actual determinations on vanadium content on all of the fuels; however, the several samples which were analyzed contain less than 15 ppm of vanadium.

One of the features of the diesel fuel blender system is to supply the injectors with the desired viscosity fuel regardless of the viscosity of the light distillate or heavy residual fuel being used. All that is required to utilize fuels of different viscosities is to vary the output of the distillate fuel proportioning pump. This can be accomplished in two ways: (1) by changing the capacity of the pump on the unit, or (2) by changing the speed of the pump.

The railroads now using the diesel fuel blender system have made no changes in timing of the engine.

The increased wear rates observed when using residual-type fuels can be reduced by paying close attention to the metallurgy of the parts used in the diesel engines such as the rings, liners, and exhaust valves. Also, the proper selection of operating conditions will help to reduce engine wear. The operating conditions and the engine parts best suited for one operation may or may not be the optimum selection for another. Therefore, it is suggested that each user determine what combination of engine parts and operating conditions are best suited for his particular operation.

Work is proceeding actively in this field. We are in no position to forecast whether this work will bear fruit; however, it will increase our knowledge of the mechanisms which are involved and place us in better position to combat the problems which will arise.

One new approach in the technique in burning residual fuels in medium-speed marine diesel engines is to employ the diesel fuel blender system as described in the paper. The viscosity of the residual fuel purchased can be adjusted by the distillate blending rate for each new combination of distillate and residual fuels procured.

With the diesel fuel blender system, mixing of residual and distillate fuels is localized substantially in the fuel circulating header and the injection system on the engine. Possible incompatibility effects in this injection system for blends of various types of residuals and distillates will require extensive investigation to evaluate to what extent such incompatibility problems may exist.

RESEARCH IN THE FUNDAMENTALS OF AUTOMOBILE CONTROL AND STABILITY

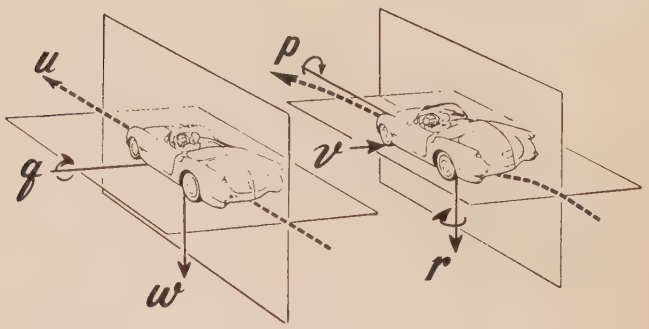

Fig. 1—Car motions

Leonard Segel, Cornell Aeronautical Laboratory, Inc.

This paper was presented at SAE National Summer Meeting, Atlantic City, June 5, 1956.

DESIGNING satisfactory "handling" into a vehicle is certainly a problem which is common to all forms of transportation. It is particularly demanding in the case of the automobile since the American motor car has become truly a mass transportation media used alike by both the expert and nonexpert. While this is the major justification for performing automotive handling research, there is also the curiosity of the engineer who wants to know how an automobile maneuvers on the road and executes those directional motions commonly grouped under the term "handling." This curiosity on automotive matters did exist among a few personnel in the Flight Research Department of the Cornell Aeronautical Laboratory in Buffalo, N. Y., and resulted in the recognition that know-how obtained from research in aircraft stability and control could be applied directly to the automotive "handling" problem.

Accordingly, a research program was formulated, and initial efforts were directed towards obtaining experimentally verified equations of motion which adequately describe the directional behavior of an automobile. In attempting to tell the story of the accomplishments achieved to date, this paper will:

1. Present a definition of the terms "handling," "control," and "stability."
2. Review briefly the state of the art which existed at the start of this program.
3. Describe the development of the vehicle lateral equations of motion and their subsequent experimental verification.
4. Use the derived theory to explain certain basic steady-state and dynamic properties of car "handling."

The investigations to be described herein were performed under a research and development agreement with the Research Staff of General Motors Corp.

Definitions

The term "handling" has been used whereas the title of this paper speaks of automobile control and stability. The question might well be asked, "What is the relationship between the two?"

In answering this question, let us first seek a definition of "handling." It is generally agreed that handling is concerned with the motions of a car outside of its plane of symmetry (Fig. 1). Thus "handling" is concerned with the yawing, rolling, and sideslipping motions of the car produced either by driver-applied steering inputs or by external disturbances in the form of road irregularities or wind gusts. Handling also has its subjective aspect. Very often it implies a driver evaluation of the following two car properties:

1. The ease and precision with which it is possible to steer the vehicle or achieve a desired path.
2. The ease with which this path or heading is maintained.

These two subjective properties are, of course, intimately associated with the objective problem of maneuvering a car and keeping it on a given course. We find that the problem of maneuvering and course keeping can be objectively examined by looking at the so-called control and stability characteristics of the car.

At this point, the words "control and stability" should be defined. On applying the laws of classical mechanics to the study of the motions of rigid bodies,

THE relationship between handling and automotive control and stability is examined and defined, and brief review of the state of the art is presented before describing current research in this field.

The automobile's response to steering control is the specific handling behavior discussed in this paper which describes (1) the development of the vehicle's lateral equations of motion, (2) their subsequent experimental verification, and (3) uses the substantiated theory to explain certain basic steady-state and dynamic properties of car handling.

whether they be automobiles, airplanes, or ships, we find that control implies:

1. A means of applying forces and moments to a vehicle in order to produce linear and angular accelerations, velocities, or displacements.

2. The existence of some specific vehicle response behavior to control force and moment inputs—this behavior being characterized by a steady state and transient response of the vehicle to the control input. Stability, on the other hand, is the ability to maintain a given state of equilibrium. A vehicle which is stable returns to its initial state of equilibrium after the disturbance has been removed or acquires a new equilibrium state if the disturbing force is held constant.

It is evident that stability and control characteristics will influence both the maneuvering and course-keeping properties of a vehicle and thereby affect its handling. The important point to note, however, is that since control and stability are capable of mathematical definition, it is possible to develop a quantitative, objective definition of a number of the many facets of "handling."

Achieving satisfactory handling or stability and control through design is a problem which is common to all types of transportation vehicles. Each vehicle has had its own distinct history of development, and examination of the literature reveals that the automobile has indeed had an interesting "past." Although the task of guiding an automobile through restricted lanes and over undulating roads, while avoiding pedestrians and other vehicles, appears to be more exacting than that of steering an airplane through the air or a ship through the ocean, we find that the science of automotive "handling" undoubtedly lags behind the stability and control theory developed to date for aircraft and water-borne vessels. Before presenting a description of the research efforts now under way, it is appropriate that the prior state of the art be briefly reviewed.

Past Automotive Handling Theory and Experiment

A literature survey reveals that, prior to 1925, engineering discussions of automotive handling were nonexistent in the various technical journals being published in the major automobile-producing countries. This does not indicate that automobile manufacturers were not concerned with the handling of their vehicles. Indeed they were. Long before an understanding was obtained of the manner in which the pneumatic tire produced a side force, the automotive engineer was concerned with the problem of designing suspensions which would keep the wheels in contact with the ground the major portion of the time. Certainly the problem of reducing unsprung mass vibrations and the design of satisfactory steering systems were the stability and control problems of that day. One did not need to know about the tire cornering-force phenomenon to realize that if he could not keep the wheels on the ground or provide the driver with a suitable "rudder" it would be difficult to maneuver the vehicle with any degree of precision. These automotive engineering problems were discussed at great length and are, without doubt, part and parcel of handling. For the purposes of this discussion, however, we have limited the concept of handling to one which is primarily concerned with the motions of the vehicle in the plane of the road. We have further restricted handling to a concept which recognizes that (1) the tire is the source of the guiding forces acting on the vehicle and (2) the vehicle turns in accordance with these guiding forces rather than in accordance with the kinematics established by a given steering geometry.

Thus a theory of handling, as we have defined it, was nonexistent before the basic concept of wheel side-slip was first advanced by Broulhiet [1] in 1925. While it was generally believed that the lateral motion behavior of an automobile was influenced by tire lateral flexibility, a so-called tire "stability" as defined by Healey on tire mechanics,[2] it remained for Broulhiet to recognize the role of side-slip in the tire flexibility mechanism. This recognition came about during attempts to explain the automobile shimmy phenomenon, a problem which was then plaguing the automobile world. Investigation of the tire as a major factor in the vibrations of steering systems continued in Germany and resulted in a classical text on the subject by Becker, Fromm, and Maruhn.[3] This work shed additional light on tire lateral mechanical behavior and perhaps set the stage for its subsequent application to the problems of automobile stability and control.

During this same year, suspension engineers at General Motors Corp. observed [4] that a car which had a steering geometry such that when it rolled to the right it steered to the left, could not be safely handled on the road at high speeds. The car was said to

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[1] Bulletin 78, Societe des Ingenieurs Civils de France, 1925, "Suspension of Automobile Steering Mechanism: Shimmy and Tramp," by G. Broulhiet.

[2] *Proceedings of Institute of Automotive Engineers*, Vol. 19, 1924–1925, p. 26: "Tire as Part of Suspension System," by A. Healey.

[3] "Vibrations of Steering Systems of Automobiles," by G. Becker, H. Fromm, and H. Maruhn. Pub. by Krayn, Berlin, 1931.

[4] General Motors Corp. Engineering Report, May, 1937, "Suspension and Handling," by M. Olley.

"oversteer." It was soon discovered that overloaded or underinflated rear tires produced the same behavior as the "oversteer" steering geometry. In an effort to obtain an explanation of this phenomenon, General Motors, in 1932, asked the Goodyear Tire and Rubber Co. to perform tests on a rolling tire. The objective, in the words of Maurice Olley, was "to find how tires pushed cars around corners." This paper [5] which published results of this investigation presumably constituted the first discussion of handling in the sense in which we have defined it here.

With the limited tire data available, the suspension group at GMC began to build up their understanding of so-called oversteer and understeer. The skid-pad test was devised as a means of measuring the oversteer or understeer properties of the car. Simultaneously the art of designing roll steer properties into the suspension was being analyzed and refined. This whole concept of vehicle steady-state stability has been set forth.[6,7,8]

Meanwhile investigators in other countries began to publish their theories of vehicle handling. While a few of the earlier works properly recognized the role of the tire, many continued to picture the vehicle turning process as one which was governed entirely by considerations of steering geometry. The development of handling theories abroad took place primarily in the countries of France, Germany, and Russia. Some of these investigations concerned themselves with the influence of (1) front and rear wheel drive, (2) the differential, (3) braking, and (4) traction on the stability of the vehicle or its inclination to skid. In France, starting in 1937, numerous papers appeared.[9-14]

Proceeding from the earlier work on tires and shimmy,[3] roadability and dynamic directional stability studies continued in Germany. The major portion of this effort was performed at the Research Institute for Vehicular Science and Motors which was established in 1930 by Dr. W. I. E. Kamm at the Stuttgart Institute of Technology. After 10 years of research in this field, a significant group of papers were published in 1940 by students and assistants of Dr. Kamm.[15-18] More recently, Moller [19] has described methods of performing wind-tunnel measurements on motor cars. In 1953, Dr. Kamm summarized the work of his colleagues at the Automotive Institute by preparing a report [20] on vehicle stability, in which emphasis was placed on the very real problem of minimizing vehicle response to wind gusts.

Russian contributions [21,22] to the study of vehicle handling began to appear in 1936. Their efforts during the Thirties and early Forties neglected the lateral flexibility of the tire but, by 1946, tire studies which had been performed both in the United States and in Germany had apparently come to their attention. Accordingly, Pevsner [23] in 1946 described various methods for testing for oversteer or understeer, and Chudakov [24] in 1947 presented a theoretical derivation of the equations of vehicle motion which did include the influence of tire lateral elasticity. Additional treatments of vehicle stability have been published by the Russians but, for the sake of briefness, they will be omitted in this account.

By and large, the language barrier proved to be sizable and, though interchange of ideas did take place between the United States, England, and France, little recognition was given to the work of the Germans, presumably because World War II was then under way and this material was not available for study. In this country General Motors Corp., as a result of their many tests, concluded that the transient behavior of the car, or turn entry and recovery, was a facet of handling as important as the steady-state behavior. The so-called checkerboard test [8] was devised in order to shed some light on the transient behavior of the car, but efforts in this direction were stopped by the demands of the war effort. Interest in transient behavior continued, however, and an analysis of a vehicle's lateral motion characteristics was performed by Schilling [25] in 1953.

To sum up, the state of the art at the beginning of this research effort was as follows:

1. The lateral mechanical properties of automotive tires had been investigated by various United States and English tire manufacturers [5,26,27] and by German researchers in this field.

2. The various design features affecting vehicle steady-state stability had been identified within the overall concept of vehicle oversteer and understeer. More recently the concept of "stability margin" had

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5 SAE *Journal*, Vol. 36, February, 1935, pp. 41–49: "Properties of Tires Affecting Riding, Steering, and Handling," by R. D. Evans.

6 Proceedings of Seventeenth Annual Meeting of Highway Research Board, December, 1937, p. 202: "Relation Between Curvature and Speed," by M. L. Fox.

7 *Proceedings of Institute of Automotive Engineers*, Vol. 30, 1938, p. 509, "National Influence on American Passenger Cars," by M. Olley.

8 SAE Transactions, Vol. 36–49, 1941, pp. 81–93: "Car Control Factors and Their Measurement," by K. A. Stonex.

9 *Journal*, Societe des Ingenieurs de l'Automobile, Vol. 10, October, 1937, p. 469: "Theoretical Study of Stability of Automobiles on Road with Front Wheel or Rear Wheel Drive," by V. G. de Seze.

10 *Journal*, Societe des Ingenieurs de l'Automobile, Vol. 15, 1942, p. 195: "Theory of Road Stability," by M. Gratzmuller.

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12 *Journal*, Societe des Ingenieurs de l'Automobile, Vol. 20, March-April, 1947, pp. 57–59: "General Dynamics of Suspended Vehicles," by M. A. Julien.

13 *Journal*, Societe des Ingenieurs de l'Automobile, Vol. 22, September, 1948, pp. 231–236: "Agreement between French Theories and Anglo-Saxon Practice Related to Road Stability," by M. A. Julien.

14 *Journal*, Societe des Ingenieurs de l'Automobile, Vol. 24, 1951: "Predetermination of Stability of Motor Vehicles," by V. Buen de Lozano.

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16 Deutsche Kraftfahrtforschung No. 44, 1940, "Tracking Position of Automobiles on Curves," by O. Dietz and R. Harling.

17 Deutsche Kraftfahrtforschung No. 44, 1940, "Directional Stability of High-Speed Automobiles," by L. Huber.

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21 Uchenyie Sapiski Dalnevostochnogo Universiteta, 1936, "On Stability of Automobile on Straight Road," by I. Kukles.

22 Isvestiya Akademii Nauk Otdeleneiye Teckhnicheskikhnauk (Bulletin of Academy of Sciences, Division of Technical Sciences), 1938, "On Stability of Automobile in Curve," by E. A. Chudakov.

23 Experiments on Stability of Automobiles," by Ya. M. Pevsner. Pub. by Moskgiz, Moscow, 1946.

24 Isvestiya Akademii Nauk Otdeleniye Teckhnicheskikhnauk (Bulletin of Academy of Sciences, Division of Technical Sciences), 1947, "Influence of Sidewise Elasticity of Wheels on Steering of Automobiles," by E. A. Chudakov.

25 "Directional Control of Automobiles," by R. Schilling. Paper presented at Industrial Mathematics Society, Detroit, February, 1953.

26 SAE *Journal*, Vol. 45, August, 1939, pp. 344–350: "Tire Behavior in Steering," by A. W. Bull.

27 Proceedings of Automotive Division, Institute of Mechanical Engineers, No. 6, 1953–1954, p. 113: "Tire Characteristics as Applicable to Vehicle Stability Problems," by T. J. P. Joy and B. C. Hartley.

Fig. 2—Simplest block diagram which can quantitatively show input-output relation

been advanced by Lind-Walker [28] as a principal measure of steady-state stability.

3. Vehicle equations of motion had been developed by French, German, and Russian engineers with varying degrees of complexity. Certain portions of this work were verified experimentally, either full-scale or by means of models. Their direct or modified adoption for use in the present study was not considered advisable since it was believed that the techniques developed for studying aircraft dynamic stability and control would not only be applicable but best suited for unraveling the intangibles of automobile dynamic behavior.

Automobile Response to Steering Control

Developing a theory of automobile directional stability and control is equivalent to developing a mathematical model of the automobile. By definition, this mathematical model would yield the control and stability characteristics of the vehicle and would quantitatively define the input-output relationship indicated in Fig. 2, where the problem is shown reduced to its simplest terms. In this figure, the car is shown as a simple black box representing a complex dynamic system. The merit of this representation is that it shows the relationship of the output, or motion of the car, to the input. The car is said to operate on the input producing a resultant output. In the process of creating a mathematical model for the car, it is very desirable that the input-output relationship be assumed invariant for a given dynamic configuration, irrespective of the size of the applied input. Stated in another way, this means that it is desirable that the automobile be treated as a linear dynamic system. The advantages of doing so are many; indeed, the application of many of the experimental techniques developed for studying dynamic systems requires the existence of linearity. Linear behavior was accordingly the first assumption made in developing a mathematical model of handling. We shall see later that this assumption is quite adequate for lateral motions of a reasonable magnitude.

Since the principle of superposition applies for linear dynamic systems, our first assumption makes it possible to break down the complicated overall problem into its simpler parts. First, it was recognized that there are three different types of inputs to the car, namely steering, road, and wind inputs. In the initial attack on the problem, study was made of the vehicle response to steering inputs while study of the response to the latter two inputs was temporarily set aside. Further simplifications were made by differentiating between steering-wheel displacement and steering-wheel torque inputs. Although the human driver steers his car by a combination of wheel displacement and feel, it is possible to study the responses to these two types of steering inputs separately. The stability and control characteristics associated with a fixed steering wheel or the response produced by a steering-wheel displacement have been called the "fixed control" characteristics, in accordance with established aeronautical practice. Conversely, the characteristics associated with a free steering wheel or the response produced by a steering torque have been designated the "free control" behavior or response. This is shown schematically in Fig. 3 where a torque or displacement is shown imposed on the steering wheel, which in turn produces an angular displacement of the front wheels, which in turn causes a side force and yawing moment to be impressed on the car, resulting in directional maneuvering of the vehicle.

The situation pictured in Fig. 3 may be simplified even further by neglecting the steering system and considering the car response to front-wheel angle inputs. Note that the essential car dynamics are contained in the "black box" at the bottom and that side force and yawing moments produced by road and wind inputs act on the same "black box." In building up the mathematical model and substantiating it with actual measurements of the car's response, we started working with the block at the bottom and then proceeded to work our way to the top of this diagram.

With the initial study restricted to the fixed control automobile, in which the front-wheel steering angle produces the only force and moment input, the assumption was then made that the lateral motions (Fig. 1) are independent of the longitudinal motions, pitch, and bounce. Thus our mathematical model would consist of vehicle equations of motion which

[28] *Automobile Engineer*, Vol. 40, August–November, 1950, pp. 281, 370: "Directional Stability, Study of Factors Involved in Private Car Design," by G. E. Lind-Walker.

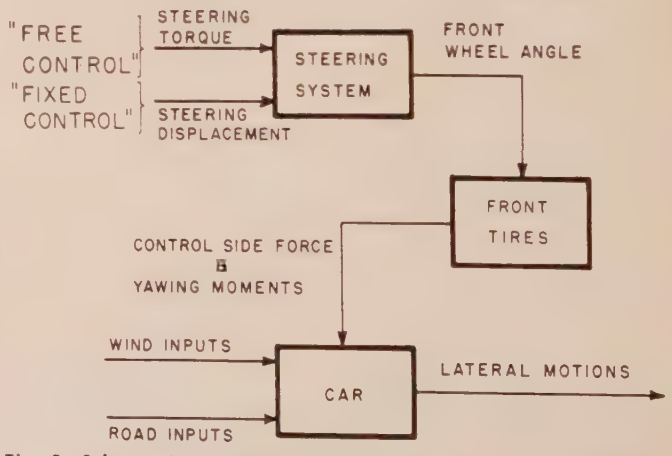

Fig. 3—Schematic representation of "fixed" and "free control"

would involve only the three lateral degrees of freedom indicated in Fig. 1—yawing, rolling, and sideslipping.

Fig. 4 shows in detail the body axes which were adopted for the automobile and the nomenclature used to describe the lateral motions of the car. Using the axis system shown in this figure, we find that the tire slip angles, defined as the angle between the center plane of the tire and its direction of motion, can be expressed in terms of the independent variables of motion, β, r, ϕ, and δ. On assuming small perturbations, the tire slip angles are as shown in Fig. 5. Note that the steer angle of the rear wheels is shown to be proportional to the roll angle of the vehicle. The indicated factor of proportionality is the so-called "rear roll steer" as determined by geometric properties of the rear suspension. In a similar fashion, the front wheels of an independent front-suspension vehicle will camber or tilt proportionally to the roll angle of the car. Accordingly, forces and moments are created in the ground plane which act on the car and are derived from the cornering forces, aligning torques, and camber thrust produced by the resultant slip and camber angles of the tires. As indicated earlier, these forces and moments are assumed to be linear functions of the tire slip and camber angles. Test data [5,26,27] acquired to date have shown that this is a good assumption, if slip and camber angles are restricted to reasonable amplitudes. Although these data have also shown that the cornering forces, aligning torques, and camber thrust are influenced by change in tire vertical loading, the variation in these properties with tire loading is neglected in order to retain linearity. Experience has shown, however, that this effect is one of second order, since the combined properties of both front or rear tires are approximately constant for lateral accelerations up to 0.3 g.

[29] "Car Stability and Transient Tire Forces," by S. A. Lippman. Paper presented at SAE National Passenger Car, Body, and Materials Meeting, Detroit, March 4, 1954.

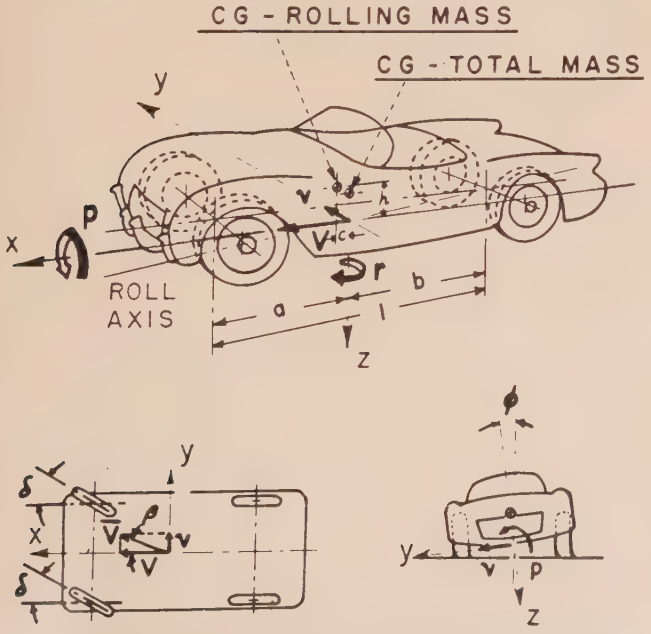

Fig. 4—Axis system for simplified automobile

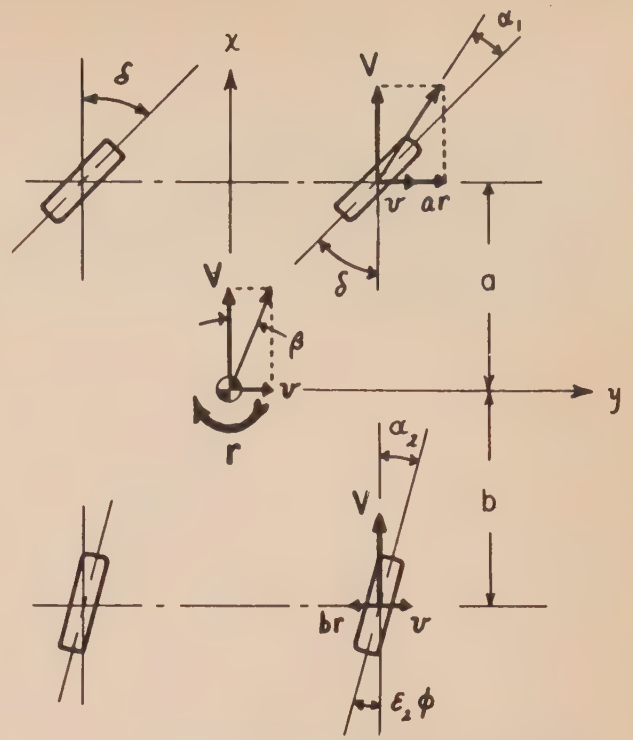

Fig. 5—Tire slip angles produced by vehicle motion

To sum up, the external forces and moments on the car derive mainly from the tire and, on assuming linear tire behavior, can be expressed as linear functions of the front and rear slip angles and the camber angle of the front wheels. Referring again to Fig. 5, we see that these side forces and yawing moments may therefore be expressed as linear functions of the variables of motion themselves. Thus there is a vehicle side force and yawing moment produced by (1) lateral velocity along the y axis (vehicle sideslip), (2) yawing velocity around the z axis, (3) roll displacement about the x axis, and (4) steering displacement of the front wheels. In addition to the forces derived from the tires, there are, of course, the rolling moments due to roll displacement and rolling velocity, produced by the suspension roll stiffness and the shock absorbers respectively. On equating all external forces and moments along and about the y, z, and x axes to the respective inertia reactions, three lateral equations of motion were obtained.

These equations are presented in the Appendix, both in their dimensional and nondimensional form. They describe the behavior of a three-degree-of-freedom, fourth-order dynamic system which represents the fixed control automobile with tire dynamics [29] neglected. (While tire dynamics, or lag in tire force buildup, does affect the car's response characteristics, the scope of this paper does not permit a full discussion of this point. We will see later that the effects of

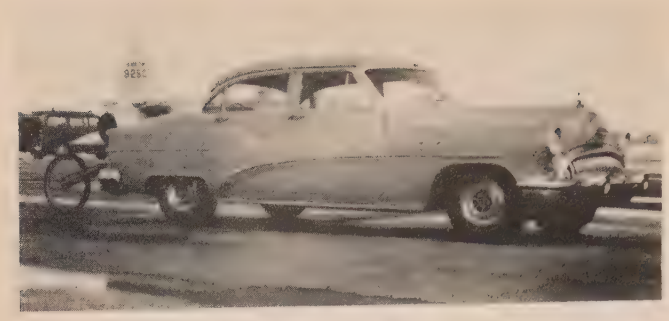

Fig. 6—Buick test car

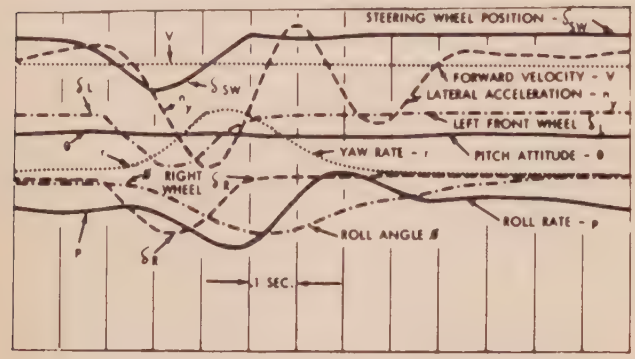

Fig. 7—Transient response of Buick to pulse input of front-wheel angle

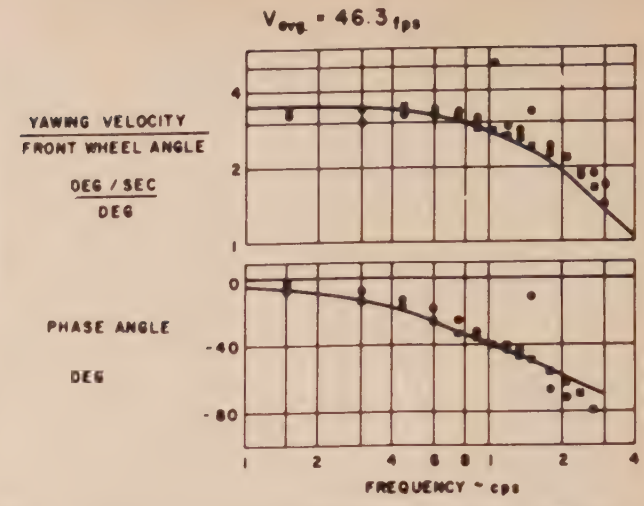

Fig. 8—Calculated and experimental frequency response of yawing velocity for 1953 Buick test car

tire dynamics are small in the particular frequency range which characterizes the rigid body motions of the automobile.)

Examination of the vehicle lateral equations of motion revealed that a total of 20 properties of the simplified automobile and its tires must be known in order to predict its response to steering inputs. Necessary measurements of the (1) mass and inertia characteristics, (2) chassis or suspension characteristics, and (3) tire characteristics of the test vehicle (the 1953 Buick, 4-door sedan, pictured in Fig. 6) were carried out jointly by the Cornell Aeronautical Laboratory, General Motors Corp., and United States Rubber Co.

Vehicle response measurements were then planned for the purpose of experimentally substantiating the response to front-wheel displacement predicted by the derived equations of motion. To measure the response to front-wheel angle, the Buick was instrumented to permit dynamic recording of the variables listed in Table 1. Both left- and right-front-wheel positions were recorded and averaged to yield the effective front-wheel angle, δ. Since it is not convenient to measure side-slip or lateral velocity, the total lateral acceleration along the y axis was measured with a lateral accelerometer. The pitch attitude, θ, as listed in Table 1, is not a lateral degree of freedom, but its measurement was made to check any coupling that may occur between the longitudinal and lateral motions. Listed also in Table 1 are the sensing instruments or transducers which were used to convert each variable to an appropriate electrical signal for recording purposes.

In order to substantiate the theory as extensively as possible, response data were obtained for as wide a range of stability conditions as was practicably possible. Accordingly, data were obtained in tests wherein the following stability parameters were varied:

1. Forward velocity.
2. Directional stability or oversteer-understeer properties.
3. Roll stiffness.
4. Total cornering coefficient.

Variations in the directional stability were achieved by moving the car's center of gravity and varying tire pressures front and rear. A roll stiffness modification was achieved by disconnecting the roll stabilizer bar, and a major change in total cornering coefficient was obtained by raising all tire pressures from 24 to 40 psi.

The initial phase of the test program was devoted to the measurement of the steady-state response of the Buick. Tests were made at nominal forward speeds of 15, 30, 45, and 60 mph on two runways of the Buffalo Municipal Airport. The primary purpose of the steady-state response tests was to obtain static or steady-state stability information for all stability configurations in which the dynamic response was to be measured. While the General Motors skid-pad test had previously indicated the range of lateral acceleration over which the automobile could be treated as a linear system, a need still existed for correlating the steady-state response in this range with predictions based upon linear tire data and linear equations of motion.

Fig. 7 shows the transient response of the Buick to a pulse input of front-wheel angle as obtained during typical transient test maneuvers. Response-time histories such as these were harmonically analyzed with the aid of a rolling-sphere-type mechanical har-

monic analyzer. The harmonic content of the output or response variables were ratioed to the harmonic content of the input yielding the frequency response of the automobile plus that of the recording circuitry. On removing the dynamics of the recording equipment, amplitude, and phase-response data were obtained, such as are illustrated in Figs. 8, 9, and 10.

Fig. 8 presents the amplitude and phase response of yawing velocity per unit front-wheel angle for the standard instrumented test configuration of the Buick. (In this configuration the center of gravity was located 51.7% of the wheelbase aft of the front-wheel centers, and the tire pressures were 24 psi front and rear.) The solid line shown in this figure is the frequency response predicted by the derived equations of motion of the linearized automobile. Note that excellent agreement exists between theory and experiment for the particular test configuration.

The calculated and experimental lateral acceleration response shown on Fig. 9 also shows remarkable agreement. The unusual shape of this response curve is explained by noting that the acceleration measured by the lateral accelerometer is composed of:

1. The centrifugal acceleration caused by yawing velocity.
2. A component of the acceleration of gravity caused by roll of the car body.
3. A linear acceleration along the lateral y axis.

The first two of these three components of lateral acceleration exist in the steady state and then decrease with increasing frequency while the third component is zero in the steady state and increases with increasing frequency. Each acceleration component is a vector quantity when considered in the frequency plane, and they add vectorially to produce the amplitude and phase characteristics observed on this figure.

Fig. 10 presents the amplitude and phase response of rolling velocity per unit front wheel angle. Again we note that there is excellent agreement between theory and experiment.

Similar comparisons were made for seven different test configurations, and on the basis of these comparisons it was concluded that the derived equations of motion adequately define the lateral dynamic behavior of an automobile. It was further concluded that the effects of lag between tire slip angle and tire force on the directional response of an automobile are very small at oscillation frequencies below 3 cps for forward speeds greater than 15 mph.

Control and Stability Characteristics

The major advantage of using linear equations of motion for the study of vehicle handling is that the factors governing vehicle stability and control can be examined analytically without resorting to actual numerical solutions. The steady-state response can be expressed as a linear function of the tire, roll steer, and roll stiffness properties of the car, and the dynamic or transient behavior in a free oscillation is completely described by the characteristic equation

Table 1—Motion Variables and Detectors

Variable of Motion	Sensing Instrument
Left Front Wheel Position, δ_L	Angular Potentiometer
Right Front Wheel Position, δ_R	Angular Potentiometer
Steering Wheel Position, δ_{SW}	Angular Potentiometer
Lateral Acceleration, n_y	Statham Lateral Accelerometer
Roll Attitude, ϕ	Minneapolis-Honeywell Attitude Gyro
Pitch Attitude, θ	Minneapolis-Honeywell Attitude Gyro
Angular Yaw Velocity, r	Doelcam Rate Gyro
Angular Roll Velocity, p	Doelcam Rate Gyro
Forward Velocity, V	Fifth Wheel—Generator Set

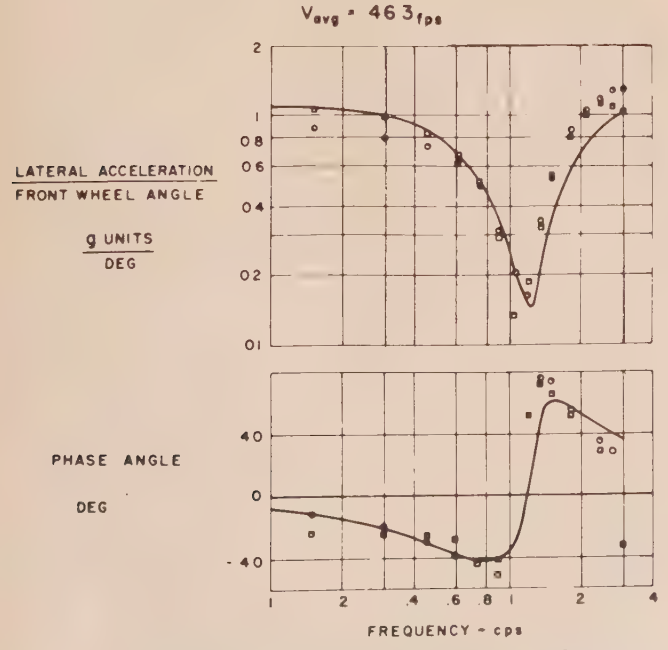

Fig. 9—Calculated and experimental lateral acceleration response for test vehicle

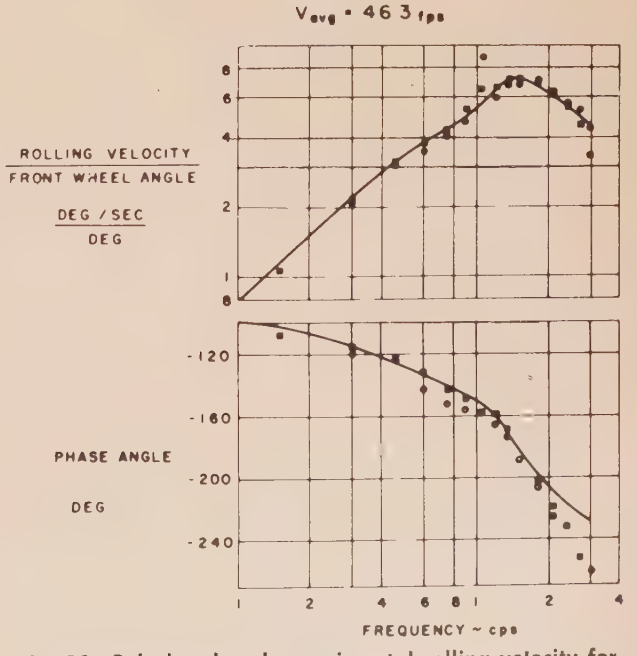

Fig. 10—Calculated and experimental rolling velocity for 1953 Buick

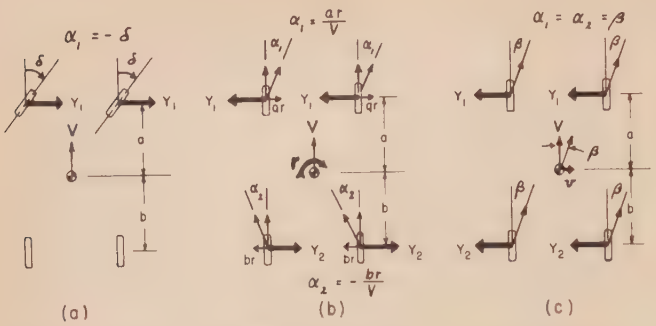

Fig. 11—Forces and moments acting on car during steady turn

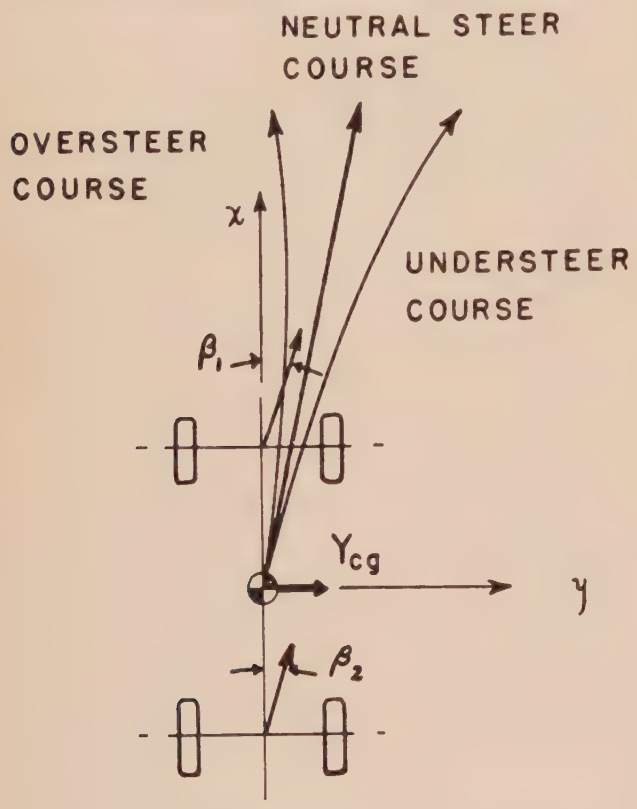

Fig. 12—Oversteer and understeer representation of vehicle stability

obtained from the three lateral equations of motion. In examining the factors which govern control and stability, let us first consider the static or steady-state yawing moment acting on an automobile during a steady turn.

Fig. 11 shows the forces and moments acting on a car which are produced by front-wheel steer angle, angular yaw velocity, and lateral velocity respectively. (For simplicity, aligning torques and forces caused by roll of the vehicle are not shown.) We see that a positive steer angle produces a positive side force at the front wheels and, thus, a clockwise or positive yawing moment on the vehicle which is equal to the total front-wheel side force times the distance between the front-wheel center and the vehicle center of gravity. The next diagram to the right shows that a positive yawing velocity induces slip angles at the front and rear tires such that the front tires produce a force to the left and the rear tires a force to the right. These tire forces produce a negative yawing moment which opposes the yawing motion of the car. A moment opposing the angular velocity is a damping moment and, therefore, the yawing moment due to yawing velocity is termed the "damping in yaw." Finally we see that a vehicle which has positive lateral velocity along its *y* axis or is side-slipping to the right is subjected to tire side force, front and rear, acting to the left. The resultant yawing moment is dependent on the center of gravity location and the relative magnitude of the front and rear cornering powers. Forward center of gravity locations and rear cornering power higher than front cornering power result in a positive yawing moment for a positive angle of side-slip while the reverse is true for rearward center of gravity locations and low values of rear cornering power relative to front cornering power. Note that a clockwise or positive yawing moment produced by positive side-slip tends to align the vehicle with its direction of motion or reduce the angle of side-slip. Such a vehicle is defined as being attitude or directionally stable. If the yawing moment is such as to increase the angle of side-slip, the vehicle is said to be attitude or directionally unstable. We see that this stability is identical to the weathercocking stability of a weathervane or the directional stability provided by the vertical tail on an airplane. In the case of a 4-wheeled vehicle, the rear tires provide a stabilizing moment while the front tires provide a destabilizing moment. Note that this attitude stability is merely a property of a vehicle which is side-slipping. It does not say anything about the overall stability of the car.

Let us now examine the moment equilibrium during a steady turn in which the steer angle is held fixed and the car possesses a constant yawing velocity and angle of side-slip. Since the moment due to side-slip is small and may be zero, it is the steering moment which primarily balances the yaw damping moment. Thus, in a right turn, the positive steering or control moment overcomes the negative "damping in yaw" moment. With the moment due to side-slip assumed zero, we find that the yawing velocity per unit steer angle is equal to the forward velocity divided by the wheelbase. This is another way of saying that the steering angle in a steady turn is equal to the wheelbase divided by the radius of the turn, the so-called Ackerman angle. Thus a vehicle, without directional stability or yawing moment due to side-slip, has a yawing velocity response which is a function of its steering geometry. A steady-state yawing velocity response equal to the forward velocity divided by the wheelbase is the basic response characteristic of the automobile and is modified only by the existence of the vehicle's directional stability properties.

The above conclusion was deduced from examining the moment balance in a steady turn. It may also

be shown that the yawing velocity response per unit steer angle is modified by moments which are a function of the roll angle of the vehicle. These moments due to roll may be positive or negative and either increase or decrease the moment tending to align the vehicle with its direction of motion. Moments due to roll angle therefore contribute to the total directional stability or instability of the vehicle. Since the moment due to side-slip can be altered by shifting the center of gravity fore and aft, it becomes convenient to describe the total directional stability in terms of an effective moment arm in per cent of the wheelbase. This effective moment arm has been termed the "static margin" of the vehicle.

It is now possible to relate "oversteer and understeer" to the static margin, which is a means of numerically defining the total directional stability of a vehicle. "Understeer and oversteer" were defined by Maurice Olley [30] in terms of the path or course pursued by a straight running vehicle when subjected to a steady side load applied at the center of gravity. On defining front and rear side-slip angles as the angle between the longitudinal axis of the car and the velocity vectors of the front and rear axle centers, it is seen that the path of the car is determined by these steady-state side-slip angles. If an applied side load produces equal front and rear side-slip angles, the vehicle moves off on a new straight-line path. A rear side-slip angle greater than the front side-slip angle will result in a curved path toward the applied load, and front side-slip angle greater than rear side-slip angle will result in a curved path away from the applied load. The terms "neutral steer," "oversteer," and "understeer" were applied to these three path configurations, in the order named, as indicated in Fig. 12.

In terms of yawing velocity Fig. 12 may be interpreted as follows:

1. When the vehicle is neutral steer, a side load applied at the center of gravity produces zero yawing velocity.
2. When the vehicle is understeer, a positive side load produces a positive yawing velocity.
3. When the vehicle is oversteer, a positive side load produces a negative yawing velocity.

A solution of the equilibrium equations of motion reveals that the steady-state yawing velocity response to a force applied along the positive y axis is positive when the static margin is positive. There is no yawing velocity response when the static margin is zero. The yawing velocity response is negative when the static margin is negative. Using the above definitions of "understeer" and "oversteer" we see that an understeer vehicle is one which possesses a positive static margin or positive directional stability; an oversteer vehicle possesses a negative static margin or negative directional stability.

To recapitulate, the automobile's yawing velocity response to steering inputs, that is, angular displacement of the front wheels is basically equivalent to the forward velocity divided by the wheelbase, and is modified by the total directional stability of the vehicle which is expressed numerically by its static margin. Fig. 13 shows the manner in which the yawing response to front-wheel angle varies with velocity for a vehicle with positive, zero, and negative static margins. If we speak in terms of curvature response rather than yawing velocity response, Fig. 13 is transformed to Fig. 14 where it is seen that a "neutral steer" vehicle

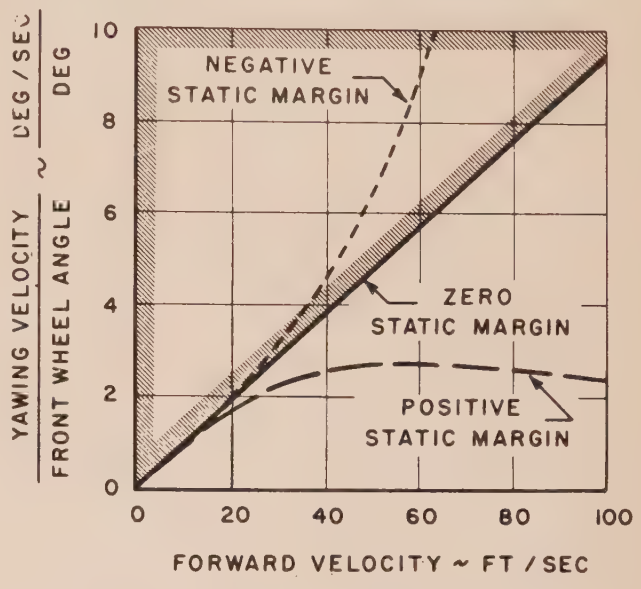

Fig. 13—Influence of static margin on steady-state yawing velocity response

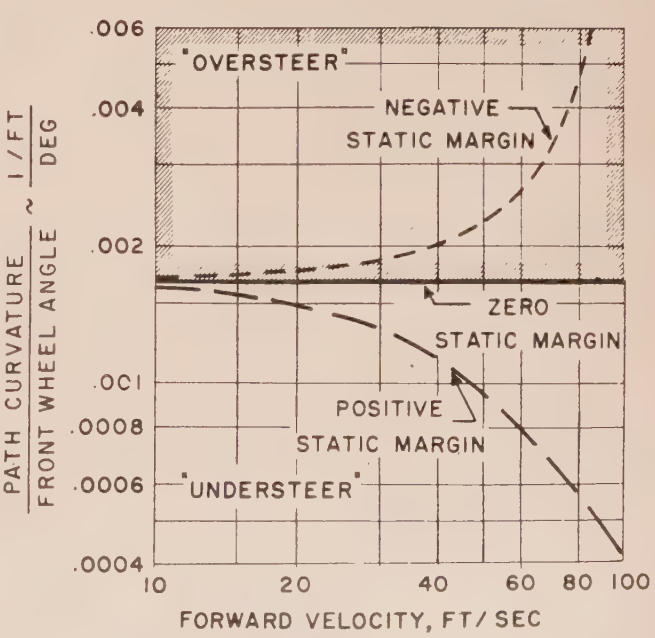

Fig. 14—Influence of static margin on steady-state curvature response

[30] *Journal*, Institute of Automobile Engineers, Vol. 15, February, 1947, p. 147: "Road Manners of Modern Car," by M. Olley.

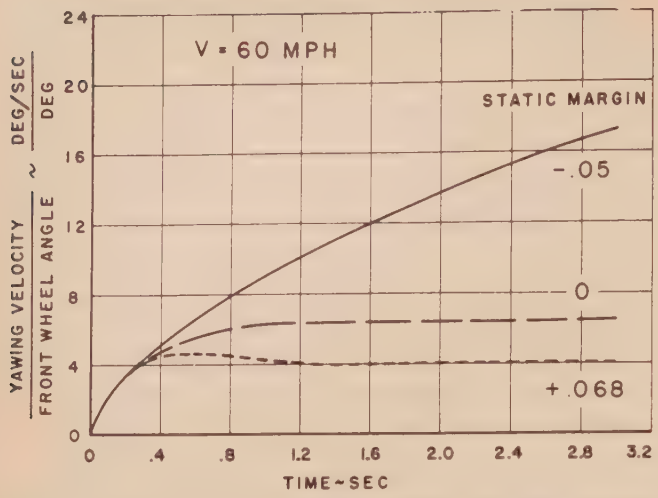

Fig. 15—Influence of static margin on yawing velocity transient response to step input of front-wheel angle

is one which has a constant curvature response with increasing forward velocity, an "understeer" vehicle has a decreasing curvature response with increasing forward velocity, and an "oversteer" vehicle has an increasing curvature response with increasing forward velocity.

Figs. 13 and 14 show that the static margin affects the steady-state response, or what is often called the control sensitivity. The static margin also has a profound influence on the transient response, and a negative static margin can produce dynamic instability above a certain critical speed. Examination of Fig. 5 shows that the slip angles induced by yawing velocity vary inversely with forward velocity. Thus the "damping in yaw" decreases in direct ratio to increases in forward speed. If the vehicle is directionally unstable, that is the yawing moment due to side-slip and roll angle is positive for negative steady-state values of side-slip and roll angle, there will be a certain speed at which the yawing moment due to side-slip and roll angle will just wash out the "damping in yaw," and a given positive control input will produce an infinite response. Mathematically speaking, dynamic instability occurs at this same combination of speed and negative static margin. At this point, a convergent mode represented by a real negative root of the characteristic equation becomes zero and then becomes positive as velocity is increased.

The static margin and the "damping in yaw" are therefore the two most important properties governing the control sensitivity and transient response characteristics of an automotive vehicle. Fig. 15 shows the yawing velocity response of the 1953 Buick to a steering control which is a step input of front-wheel angle. (While step changes in steering cannot be achieved in practice, there are significant advantages in studying the response to a mathematical step function.) The step response of the Buick is shown for an "understeer," "neutral steer," and "oversteer" configuration at 60 mph. Note that at this speed the oversteer vehicle is just on the verge of becoming unstable. The directional stability, as indicated numerically by the static margin, has a significant effect on the transient behavior. Contrary to popular belief, the understeer vehicle is found to be the faster responding vehicle, if speed of response is taken as the time required to reach steady state.

Fig. 16 presents the response of an understeer Buick at speeds of 30, 60, and 90 mph. The decrease in yaw damping with increasing speed is completely responsible for the indicated change in transient behavior. Note that this overshoot characteristic will be still larger at speeds higher than that shown and that this explains, to a large extent, the increased driver attention required to achieve precise control of very-high-speed vehicles. Fig. 17 presents the yawing velocity response of an oversteer Buick for the same three values of forward velocity. No overshoot characteristic is encountered with increasing speed in this case. Rather, the response becomes slower as the static sensitivity increases sharply with increase in speed. At 90 mph, we note that this vehicle is dynamically unstable. It is conceivable that a human may be able to stabilize such a vehicle by means of corrective steering action, but experience has shown that this is a very difficult task.

To summarize, Figs. 15, 16, and 17 indicate the manner in which static margin (oversteer or understeer) and forward speed affect the transient response of an automobile to steering control. Only by assuming linearity is it possible to obtain a basic understanding of the resultant motion behavior by means of the directional stability and damping-in-yaw concepts.

Summary

The development of a mathematical model for the fixed control automobile and its subsequent verifica-

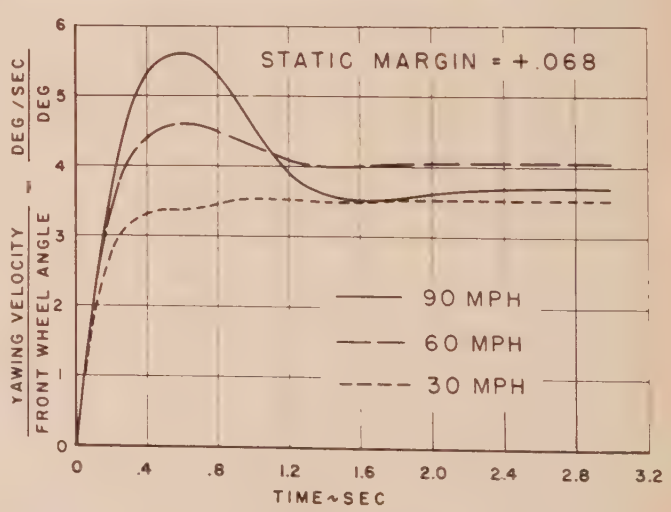

Fig. 16—Influence of forward velocity on yawing velocity transient response of understeer vehicle

tion by full-scale response tests was the objective of this research program. This work has defined vehicle dynamic behavior in response to steering inputs and will lead to a fuller understanding of the rigid body mechanics of the linearized automobile.

The performance of studies on automotive dynamic behavior implies, however, that basic information on such things as tire mechanics and vehicle aerodynamics is available. Such is not the case and, therefore, it is necessary that additional research and test programs be conducted in order to obtain the required tire and aerodynamic data. To this end, the test rig pictured in Fig. 18 is currently being used at Cornell Aeronautical Laboratory, Inc. to make six component force and moment measurements on a tire operating on a flat road surface. This equipment was designed and developed by the Cornell Aeronautical Laboratory, Inc., under contract with the USAF. It has been used in several test programs to determine the lateral force and moment properties of a variety of automotive tires.

One final point should be made. It should be noted that the investigation described in this paper treats, in actuality, only one part of the overall handling problem. The other part of the problem is the scientific determination of desirable handling qualities from the driver's point of view, that is, what does the driver want and need in the way of handling qualities? This corollary question is now being actively investigated in the aeronautical field, and application of this experience to automotive research would go a long way in helping to solve the complete problem of automobile handling.

Acknowledgment

The author acknowledges, with many thanks, the assistance and cooperation of his colleagues in the Flight Research Department of the Cornell Aeronautical Laboratory and the invaluable aid of personnel of General Motors Corp. and United States Rubber Co. In particular, acknowledgment is made to J. Bidwell for permission to publish this information and to A. Pulley who was responsible for instrumenting the test vehicle and who assisted the author throughout the test program.

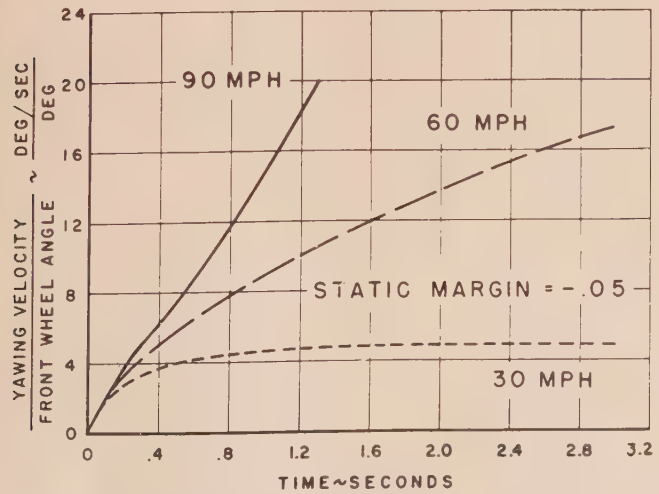

Fig. 17—Influence of forward velocity on yawing velocity transient response of oversteer vehicle

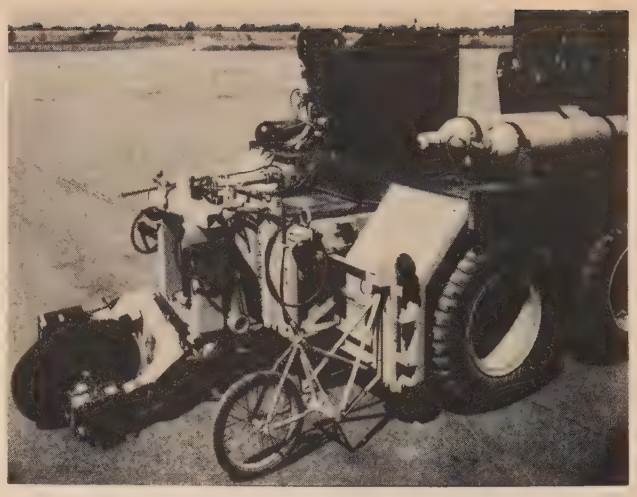

Fig. 18—Cornell-USAF tire test apparatus

APPENDIX

The level road, fixed control equations of motion used in the research described herein are presented in this Appendix. They are given below in both dimensional and nondimensional form.

List of Symbols

M = Total mass of vehicle, slugs
M_s = Rolling mass of vehicle, slugs
I_x = Rolling moment of inertia of the rolling mass about the x axis, slug-ft^2
I_z = Yawing moment of inertia of the vehicle about the z axis, slug-ft^2
h = Height of center of gravity of rolling mass above the roll axis, ft
a = Distance between vehicle center of gravity and front-wheel center, ft
b = Distance between vehicle center of gravity and rear-wheel center, ft
c = Distance between center of gravity of the rolling mass and the z axis, ft
I_{xz} = $M_s hc$, product of inertia of the vehicle about the x and z axes, slug-ft^2
x, y, z = Body axes fixed in vehicle
ϕ = Roll angle of vehicle, deg or radians
σ = Steer angle of front wheels, deg or radians
V = Forward velocity of the vehicle, fps
v = Lateral velocity along y axis, fps
p = Rolling velocity around x axis, radians per sec

APPENDIX—Continued on following page

r = Yawing velocity around z axis, radians per sec
Y = Force along y axis, lb
L = Rolling moment about the x axis, ft-lb
N = Yawing moment about the z axis, ft-lb
l = Wheelbase, ft
z_1 = Height of roll axis above ground at front axle, ft
z_2 = Height of roll axis above ground at rear axle, ft
$\epsilon_2 = \partial\sigma_2/\partial\phi$, rear roll steer, radians per radian
γ = Camber angle of front wheels, radian
$\partial\gamma/\partial\phi$ = Rate of change of front wheel camber with respect to body roll, radians per radian
k = Roll stiffness of suspension, ft-lb per radian
$\partial L/\partial p$ = Roll damping produced by the shock absorbers, ft-lb-sec per radian
α_1 = Slip angle of front tires, radian
α_2 = Slip angle of rear tires, radian
$C_1 = \partial Y/\partial\alpha_1$, cornering power of front tires, lb per radian
$C_2 = \partial Y/\partial\alpha_2$, cornering power of rear tires, lb per radian
$C = C_1 + C_2$, total cornering power of vehicle, lb per radian
AT = Aligning torque, ft-lb
$\partial AT/\partial\alpha_1$ = Slope of aligning torque versus front-wheel slip angle, ft-lb/rad
$\partial AT/\partial\alpha_2$ = Slope of aligning torque versus rear-wheel slip angle, ft-lb/rad
D = Rolling resistance of tire, lb
Z = Vertical load on tire, lb
$\partial D/\partial Z$ = Change in rolling resistance with change in vertical load, lb per lb
$\partial Y_1/\partial\gamma$ = Camber thrust per unit camber angle of front wheels lb per radian
$\beta = v/V$, side-slip angle of vehicle
g = Acceleration of gravity, ft per sec^2
$\tau = \sqrt{l/g}$, sec
$\hat{r} = \sqrt{l/g}\, r$
$\hat{p} = \sqrt{l/g}\, p$
$D = \sqrt{l/g}\,\frac{d}{dt}$, nondimensional operator
$\mu = V/\sqrt{lg}$

(Note: A dot over a variable denotes its time derivative, d/dt.)

Dimensional equations of motion of automobile:

Side force equation:

$$Y_\beta\beta + Y_r r + Y_\delta\delta + Y_\phi\phi = MV(\dot{\beta} + r) + M_s h\dot{p}$$

Yawing moment equation:

$$N_\beta\beta + N_r r + N_\delta\delta + N_\phi\phi = I_z\dot{r} + I_{xz}\dot{p}$$

Rolling moment equation:

$$L_p p + L_\phi\phi = I_x\dot{p} + M_s hV(\dot{\beta} + r) + I_{xz}\dot{r}$$

where:

$$Y_\beta = C_1 + C_2 = C$$

$$Y_r = C_1\frac{a}{V} - C_2\frac{b}{V}$$

$$Y_\delta = -C_1$$

$$Y_\phi = -\epsilon_2 C_2 + \frac{\partial Y_1}{\partial\phi}$$

$$N_\beta = aC_1 - bC_2 + \frac{\partial AT}{\partial\alpha_1} + \frac{\partial AT}{\partial\alpha_2} + \frac{\partial D}{\partial Z}(C_1 z_1 + C_2 z_2)$$

$$N_r = C_1\frac{a^2}{V} + C_2\frac{b^2}{V}$$

$$N_\delta = -aC_1 - \frac{\partial AT}{\partial\alpha_1}$$

$$N_\phi = b\epsilon_2 C_2 + a\frac{\partial Y_1}{\partial\phi} + \frac{\partial D}{\partial Z}k$$

$$L_p = \frac{\partial L}{\partial p}$$

$$L_\phi = M_s gh + k$$

Nondimensional equations of motion of automobile with input terms placed on right-hand side:

Side force equation:

$$(\mu D - y_\beta)\beta + (\mu - y_r)\hat{r} + (mD^2 - y_\phi)\phi = y_\delta\delta$$

Yawing moment equation:

$$(-n_\beta)\beta + (i_z D - n_r)\hat{r} + i_{xz}D^2 - n_\phi)\phi = n_\delta\delta$$

Rolling moment equation:

$$(m\mu D)\beta + (i_{xz}D + m\mu)\hat{r} + (i_x D^2 - l_p D - l_\phi)\phi = 0$$

where:

$$m = \frac{M_s h}{Ml}$$

$$i_{xz} = \frac{I_{xz}}{Ml^2} = \frac{M_s hc}{Ml^2}$$

$$i_x = \frac{I_x}{Ml^2}$$

$$i_z = \frac{I_z}{Ml^2}$$

$$y_\beta = \frac{Y_\beta}{Mg} = \frac{C}{Mg}$$

$$y_r = \frac{Y_r}{M\sqrt{lg}} = -\frac{C}{Mg}\frac{1}{\mu}\left(\frac{C_2}{C} - \frac{a}{l}\right)$$

$$y_\phi = \frac{Y_\phi}{Mg} = -\frac{C}{Mg}\left(\epsilon_2\frac{C_2}{C} - \frac{\partial Y_1/\partial\phi}{C}\right)$$

$$y_\delta = \frac{Y_\delta}{Mg} = -\frac{C}{Mg}\left(1 - \frac{C_2}{C}\right)$$

$$n_\beta = N_\beta/Mgl = -\frac{C}{Mg}\left[\frac{C_2}{C} - \frac{a}{l} - \frac{\partial AT/\partial\alpha_1 + \partial AT/\partial\alpha_2}{Cl} - \frac{\partial D}{\partial Z}\left(\frac{z_1}{l} + \frac{z_2 - z_1}{l}\frac{C_2}{C}\right)\right]$$

$$n_r = N_r/Ml\sqrt{lg} = \frac{1}{\mu} = -\frac{C}{Mg}\left[\frac{C_2}{C}\left(\frac{2a}{l} - 1\right) - \left(\frac{a}{l}\right)^2\right]$$

$$n_\phi = N_\phi/Mgl = -\frac{C}{Mg}\left(-\frac{C_2}{C}\epsilon_2\frac{b}{l} - \frac{a}{l}\frac{\partial Y_1/\partial\phi}{C} - \frac{\partial D}{\partial Z}\frac{k}{Cl}\right)$$

$$n_\delta = N_\delta/Mgl = -\frac{C}{Mg}\left[\frac{a}{l}\left(1-\frac{C_2}{C}\right)+\frac{\partial AT/\partial\alpha_1}{Cl}\right]$$

$$l_p = L_p/Ml\sqrt{gl}$$

$$l_\phi = L_\phi/Mgl$$

Three physically significant parameters appear again and again in the above listing of the nondimensional stability derivatives. These are:

1. the total cornering coefficient of the vehicle, $-C/Mg$.
2. the center of gravity location, a/l.
3. the ratio of rear to total cornering power, C_2/C.

While stability parameters such as rear roll steer, ϵ_2, and front-wheel camber thrust, $\partial Y_1/\partial\phi$, do have an appreciable influence on the steady state and dynamic response of an automotive vehicle, the basic stability parameters of the fixed control automobile, exclusive of its mass and inertia characteristics, are the three above-mentioned quantities.

DISCUSSION

Wants Aerodynamic Effects to be Considered

—Albert Strumpf
Stevens Institute of Technology

THREE main comments may be made with regard to the material presented by Mr. Segel.

1. Aerodynamic effects are neglected in the formulation of the general mathematical model of the automobile. These effects should not be ignored because large aerodynamic forces and moments will generally come into play at reasonably high velocities as soon as the resultant velocity vector of the car makes an angle (different from zero) with the principal plane of symmetry of the car. This will occur just after the vehicle is given any disturbance whether it be due to gusts of wind, road, or steering.

2. The impression is left with the reader that the response of an automobile to a pulse input of front-wheel angle obtained experimentally is in agreement generally with that predicted by the mathematical formulation (which omits aerodynamic terms). The experimental data presented in the comparison were obtained at the low velocity of roughly 30 mph. Actually, aerodynamic effects will assume an increasingly more important role (in the dynamic response of the car to disturbances) as the velocity of the car increases. The reason for this is that the aerodynamic forces and moments acting on the vehicle increase as the square of the velocity, whereas the tire forces and moments do not.[a,b] For this reason, I think that discrepancies between experiments and the mathematical model (which does not possess aerodynamic properties) will increase as the car velocity increases.

3. A formulation of asset of linear differential equations to represent the motion of the vehicle is given in the Appendix, but the solution of these equations with appropriate initial conditions and the numerical constants used in these equations is not included. It would be useful for the automotive engineer to have these solutions stated explicitly, together with the numerical values of the constants which were employed in the equations for the car which was tested.

The subject matter considered in this paper is of extreme importance to the designer of automobiles. It is necessary to have a clear understanding of the basic factors governing the handling qualities (stability and control) of the automobile in order to design for safety and economy. Since there has been very little fundamental research in this field, there are many gaps in the store of information on the subject. This paper fills one of these gaps, and it is hoped that its publication will stimulate interest and support for future research.

[a] See footnote 20 of paper.
[b] Experimental Towing Tank Note No. 391, Stevens Institute of Technology, April, 1946, "Notes on Automobile Stability," by Albert Strumpf.

ORAL DISCUSSION

Reported by R. E. Owen

Research Staff, GMC

A. C. Bodeau, Ford Motor Co.: I would like first to compliment Mr. Segel on a very interesting and comprehensive paper.

Mr. Segel states that in setting up the mathematical model, "the assumption was then made that the lateral motions are independent of the longitudinal motions, pitch, and bounce." I believe that most of us will contest this point on the basis that if the pitch and bounce and the motions of the unsprung masses are not adequately controlled the car will "lose its footing," so to speak, as the tires leave the road surface on rough surfaces.

I believe, however, that Mr. Segel's assumption is a valid one for moderately smooth to smooth road surfaces, where the suspension and tire engineers have been able to design the components so that the tires maintain reasonably good contact with the road surface.

Mr. Segel: For this discussion of control and stability only smooth roads were considered. Mr. Bodeau is correct in his statement made with regard to tires losing contact with the road surface. In connection with this problem, the following remark was made in the historical section of the paper: "Long before an understanding was obtained of the manner in which the pneumatic tire produced a side force, the automotive engineer was concerned with the problem of designing suspensions which would keep the wheels in contact with the ground . . ."

Mr. Bodeau: Could you please explain the measurements of the steady-state response?

Mr. Segel: Steady-state response measurements were made by measuring the change in front-wheel angle, yawing velocity, roll angle, and lateral acceleration on going from straight travel to a steady turn maneuver. The test procedure was to stabilize at the desired speed with the test car proceeding straight down a runway. Recording of the motion was started before the steering wheel was turned to a new fixed position and was maintained until the steady state was reached.

Mr. Bodeau: With reference to Fig. 15, the author indicated that the figure showed incipient instability with a static margin of -0.05, for which condition the curve is convex upward. My interpretation would be that instability would not be reached until the curve is concave upward, and that incipient instability would be indicated by a straight line directed generally radially from the origin.

Mr. Segel: Mr. Bodeau's interpretation is correct. The statement, "the oversteer vehicle is just on the verge of becoming unstable" implied that a few more miles per hour of forward velocity would cause one root of the characteristic equation to become zero, in which case the response curve would be a straight line after the stable modes were damped out. Additional speed would then cause the response curve to become concave upward.

Mr. Bodeau: In relation to the lag time between tire slip angle and tire force, it was concluded that the effects are very small below 3 cps for forward speeds greater than 15 mph. It would be appreciated if these figures could be translated into response time, or better still, steering error as a function of car speed. Also, can this be related to the driver's reaction time, and does this phenomenon lead to control problems such as oversteering?

Mr. Segel: In reply to the question about tire lags and their relation to items such as response time and/or steering error, no answers are available from the paper as presented. Investigations of the aircraft shimmy problem and by S. Lippman in an SAE paper entitled "Car Stability and Transient Tire Forces" have shown a tire develops its equilibrium deformation after application of a side force or change of slip angle as the tire rolls forward a distance on the order of a contact print length. Thus, at high forward velocities the equilibrium deformation is achieved in a very short time. At low forward velocities, the equilibrium deformation is achieved in significantly longer times. The time

constant is therefore the appropriate distance constant divided by the forward velocity. Differences between the steady-state values of cornering forces and the actual time-dependent values will, of course, be a function of the steering rate and/or the frequency of the motion. Measurement of the car response at frequencies below 3 cps shows that tire dynamics are not important in this particular frequency range. They do, however, affect the response of higher frequencies, and their influence on the motion is seen particularly at the initial portion of a response to a sudden input such as a step input.

Mr. Bodeau: Mr. Segel points out that further development of the approach employed in his paper will hinge on basic information regarding tire mechanics and vehicular aerodynamics. He also states that the objective approach must be carried into the subjective field to determine what are desirable handling qualities from the driver's point of view.

On these points I wholeheartedly concur. It has been our observation that there is insufficient material available on tire mechanics and that available material contains many contradictions. It would appear that tires are a rather inconsistent mechanism, or that testing techniques and facilities have not been sufficiently developed and standardized to give consistent results. I am also of the opinion that vehicular aerodynamics is an important consideration in the stability problem, and it will become increasingly more important as driving speeds increase.

The roll steering properties of both front and rear suspensions is another factor which will warrant much consideration.

There is also room for the development of stability and control information under unfavorable road conditions encountered during inclement weather. The problem of stability and control in relation to braking also offers opportunity for further study.

Mr. Segel's work has resulted in some remarkably simple criteria for stability and control. It has gone even further; it has reduced the systems approach to a level that shows the influence of factors such as tire pressure, center of gravity location, and roll bars, and these are things about which all automotive engineers know something. It is gratifying to read a paper on systems analysis which has been written so that it is useful and understandable to those of us who are not expert mathematicians.

Mr. Segel is to be complimented on a very interesting and provocative paper on an old subject. If I may prophesy, I would judge that his paper will stimulate the interest and activities of many ride and handling engineers and analysts.

R. N. Janeway, Janeway Engineering Co.: What is the frequency referred to in Figs. 9 and 10? Is it yaw oscillation?

Mr. Segel: The frequency referred to in Figs. 8, 9, and 10 is the oscillation frequency of the automobile as excited by a steady sinusoidal oscillation of the front wheels. The ordinate for the top curve represents the ratio of the peak amplitude of the output or response variable to the peak amplitude of the input variable (front wheel displacement). The ordinate for the lower curve is the phase angle between the sine wave representing the input and the sine wave representing the response or output.

Mr. Janeway: How were equations solved? Longhand or by computer?

Mr. Segel: Computer.

L. Zwissler, Ford Motor Co.: Will you evaluate the inaccuracies and magnitude of error inherent in the assumption of a linearized system.

Mr. Segel: Any dynamic system can be represented by a linear model if the disturbances are assumed to be infinitesimal. The problem is how well the linear model represents the system when the motions increase in size. One way to check the linearity of a system is to excite it at different amplitudes and then determine the amplitude range over which the response per unit input remains constant. When this was done for the test Buick, it was found that the steady-state response plotted versus front wheel angle input was linear up to lateral accelerations in the vicinity of 0.3 *g*. The repeatability of the experimental frequency response points, as determined both from pulse and step inputs, is also a proof of linearity for the specific range of front-wheel angle excitations used in the tests.

R. Sanders, Chevrolet Motor Division, GMC: I assume the oversteer and understeer tests were made on the same Buick. What mechanical changes were made in the front and rear suspensions to produce these results? This car has coil springs in the rear suspension.

Mr. Segel: No mechanical changes were made in the front and rear suspensions of the Buick to produce understeer and oversteer configurations. As mentioned in the paper, the directional stability (understeer-oversteer properties) were varied by moving the car's center of gravity and changing the tire pressures, front and rear. Lead ballast was used to obtain large shifts in the center of gravity location.

Albert Strumpf, Stevens Institute of Technology: Since tire damping terms are decreased as speed increases, while aerodynamics damping terms increase with speed, should not the results presented here be modified to include aerodynamic forces and moments? Also, I believe that no aerodynamic terms were included in your equations of motion, yet agreement with measured motions are obtained. Does this mean that aerodynamics are unimportant in automobile stability work? If these terms are important, how do you account for this agreement?

Mr. Segel: The first two questions are basically equivalent. On the basis of the results obtained in these tests, aerodynamic forces appear to be relatively unimportant at speeds of 60 mph and less. The point is well taken, however, that aerodynamic forces will increase with speed while pneumatic tire behavior is substantially constant with speed. While it is true that damping in yaw due to tire forces does decrease with speed and aerodynamic damping in yaw increases with speed, the pertinent question is what is the relative magintude of these two damping terms. It is probably well to point out that measurement of vehicle aerodynamic damping in yaw is relatively difficult.

Mr. Strumpf: How did you put initial conditions and forcing functions into your equations of motion?

Mr. Segel: For the frequency response calculations, the forcing function was a sinusoidal oscillation of the front wheels. In the transient response calculations, the forcing function was step input of front-wheel angle, and it was assumed that the initial value of the variables of motion were all equal to zero.

Author's Closure To Discussion

The author would like to reply to Mr. Strumpf's comments dealing with the desirability for including aerodynamic effects in the mathematical model of the automobile. In principle Mr. Strumpf is correct. However, as a practical matter, the question remains (1) as to when aerodynamic forces should be considered "large" and (2) what speeds should be considered to be "reasonably high velocities." The question can be answered only by considering some actual numbers which indicate the relative magnitude of aerodynamic forces and tire forces at a given speed. Barth [c] presents some vehicle aerodynamic data which should suffice for this purpose. At a speed of 100 mph the side force per unit aerodynamic slip angle for a car shape similar to a 1953 Buick is approximately 23 lb per deg. The tire force per unit sideslip angle of the vehicle is approximately 600 lb per deg. Thus, on an equivalent basis the aerodynamic side force is only 4% of the tire side force. The aerodynamic yawing moment per unit aerodynamic slip angle is approximately 80 ft-lb per deg in unstable direction, and although this is a small moment it can be several times greater than the tire yawing moment per unit vehicle sideslip angle.

The conclusion to be drawn from the above numbers is that a vehicle maneuvering in still air at a speed of 100 mph would have virtually no change in its damping in sideslip and damping in yaw although its over-understeer characteristics would be slightly modified. While we conclude that vehicle response to steering in still air is influenced only to a slight degree by aerodynamic forces and moments, we recognize that the aerodynamic forces and moments produced by a crosswind are quite large and constitute a significant input tending to push the vehicle off course unless corrective action is taken by the driver. If one is concerned with the problem of minimizing vehicle response to crosswind disturbances, then vehicle aerodynamics become an essential part of the problem, even at low speeds.

[c] *Engineers Digest*, Vol. 17, October, 1956, pp. 427–432: "Aerodynamics of Car-Body Shapes," by R. Barth.

DYNAMIC

TESTING OF SEAT BELTS

D. M. Finch and J. D. Palmer, University of California

This paper was presented at the SAE Summer Meeting, Atlantic City, June 5, 1956.

TESTS simulating actual dynamic loading that a seat belt might receive during a crash of a motor vehicle are discussed in this paper. Design criterion was set as 20-*g* deceleration uniformly applied for 50 millisec.

Development and design of the test sled, body block, frame, quick-release mechanism, and in particular the oil buffer are discussed.

According to the author this dynamic test machine is the most valid test for predicting failure of seat belts.

THE California Vehicle Code was amended in 1955 to include Section 677.5, "Safety Belts." [1] This section requires the approval of seat belts prior to their sale or use. Seat-belt testing was started at the University of California at the request of the California Highway Patrol as a result of this legislation and brought about the necessity of establishing and maintaining test requirements. Most test procedures in current use are of a static type and do not incorporate a dynamic loading test. The notable exceptions are the relatively few tests utilizing a car, a dummy, and electronic instrumentation, wherein the car is crashed into a barrier. The results of these dynamic crash tests did not always check the static test results, but they seemed to be much more indicative of actual performance.

We believed that dynamic testing was the proper way to evaluate seat belts, but our problem was to develop a test which could be made on large quantities of belts without the tremendous expense of crashing an automobile each time. The test should simulate the actual dynamic loading that a seat belt might receive during a crash of a motor vehicle. There are an infinite number of boundary conditions for such crashes; therefore, we had to decide upon some of the parameters. After numerous conferences and a review of the available data on crashes and deceleration rates, we decided to design the dynamic test around the conditions prevailing when a car crashed into a solid barrier at a speed of 20 mph. The deceleration varies under these conditions from a peak of approximately 40 *g* to 0 in a period of 50 to 60 millisec. There seems to be a period of 30 to 50 millisec where an average deceleration of 20 *g* is representative. We set our design criterion as 20-*g* deceleration uniformly applied for 50 millisec. To obtain this decel-

[1] Vehicle Code, State of California, as amended to 1955, p. 368. Pub. by California Department of Motor Vehicles, Sacramento, 1955.

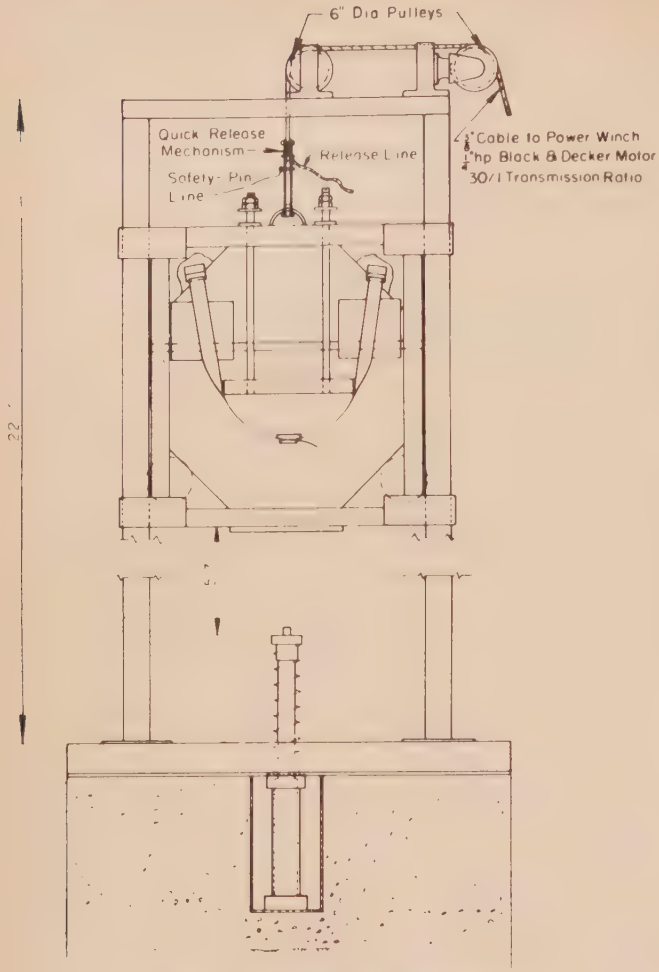

Fig. 1—Dynamic test machine

eration rate in a simulated test, several methods were available. Of these methods the use of an oil buffer was selected,[2] since it is the simplest to build, the easiest to maintain, and the best for consistent results. The oil buffer is used to decelerate a sled carrying a 150-lb body block restrained by a seat belt.

The first part of the report covers the design and development of the oil buffer, the test sled, body block, frame, and quick-release mechanism of the dynamic tester. The second part of the report covers the other tests performed on seat belts as a part of the testing procedure as established by the California Highway Patrol under authority of the 1955 California Vehicle Code.

Test Equipment Design, Construction, and Validation

The entire test machine is shown in Fig. 1 and consists of a sled mounted between two vertical columns. These columns guide the sled, which contains the body block and seat belt, during its 16-ft drop in which it is accelerated to 20 mph. The sled is then decelerated by the oil buffer at the bottom of the free fall.

The test machine was designed to have sufficient strength to withstand any force that might be applied by the dynamic loading of the seat belt and still maintain the shape of any moving part with a minimum of deformation. The body block used to apply the weight in the seat belt conforms to the CAA standard body block of 150-lb weight.[3] The general design features of the test sled and body block are shown in Figs. 2 and 3. The sled release mechanism is shown in Fig. 4.

Oil-Buffer Design—The design equations for a constant deceleration oil buffer were worked out by F. Hymans.[4] The original equations were specifically suited for elevator stops and are applicable for our purpose with a few changes.

The symbols to be used throughout the derivation of the oil-buffer design are defined as follows: [4]

S = Effective buffer stroke; also, the distance at the time $t = 0$ of the buffer piston from its terminal position

s = Distance of the piston from its terminal position at time t

y = Number of holes through which fluid escapes at time t when the displacement is $S - s$

v_e = Velocity of efflux of the fluid through the orifices

d = Diameter of the orifices

c = Coefficient of efflux of the fluid through the orifices

τ = Hoop stress in the buffer cylinder wall

D = Inside diameter of the buffer cylinder

t = Time measured from the beginning of the buffer stroke

h = Free-fall distance

Y = Total number of orifices

C = Constant defined by $\frac{cA^2}{f^2}$

P_o = Initial fluid pressure in the buffer cylinder at time $t = 0$

g = Acceleration of gravity, taken as 32.2 fps per sec

V_o = Initial speed at which the load strikes the buffer

Q = Total load to be stopped by the buffer

P_{ave} = Average fluid pressure over the effective buffer stroke S

A = Area of the buffer piston

p_t = Fluid pressure at the end of the buffer stroke

T = Time required for uniform deceleration of 20 g over the buffer stroke S

v = Speed of the load at time t

σ = Allowable stress for the metal used in the cylinder

f = Area of one orifice

a = Acceleration impressed on the piston and load at time t

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2 Personal communications with Mr. Frank B. Henry of Industrial Safety and Accident Commission.

3 Technical Standard Order C22b: "Safety Belts," Pub. by United States Civil Aeronautics Administration.

4 ASME Transactions, Vol. 48, 1926, pp. 1095–1163: "Emergency Stops of Gearless Traction Elevator," by F. Hymans.

The general equations are as follows:

1. General energy equation:

$$\begin{bmatrix}\text{work done by}\\ \text{fluid pressure}\end{bmatrix} = \begin{bmatrix}\text{kinetic energy}\\ \text{of the load}\end{bmatrix} + \begin{bmatrix}\text{work done}\\ \text{by the load}\end{bmatrix}$$

Stated symbolically

$$AP_{ave}S = \frac{Q}{2g} V_o^2 + QS$$

2. Fluid pressure:

$$P_o = C\left(\frac{V_o}{Y}\right)^2$$

3. Setting $P_o = P_{ave} = P_t$

$$C\left(\frac{V_o}{Y}\right)^2 = \frac{Q}{A}\left[\frac{V_o^2}{2gs} + 1\right]$$

Utilizing the above equations in conjunction with those governing the motion of a mass subject to free fall, the design of the oil buffer was as follows, assuming the following conditions:

1. Free-fall distance equals 16 ft.
2. Combined sled and body-block weight equals 765 lb.
3. Friction during free fall is negligible.
4. Internal diameter of the buffer cylinder equals

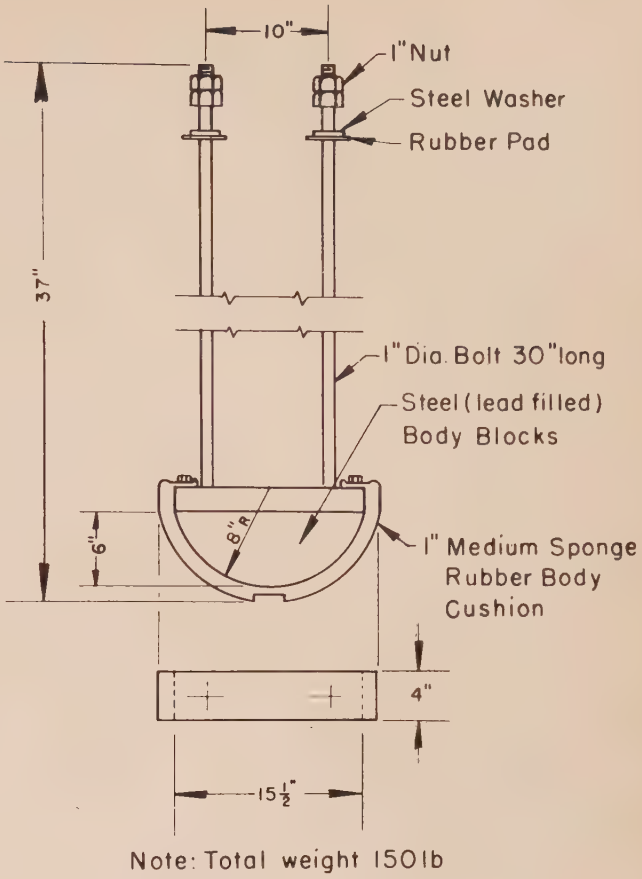


Fig. 3—Body block of dynamic test machine

Note:
1. Total sled weight including body block, 764 lb
2. Sled drop distance, 16 ft; max. velocity approx. 20 mph
3. Sled deceleration after fall, 20 G's

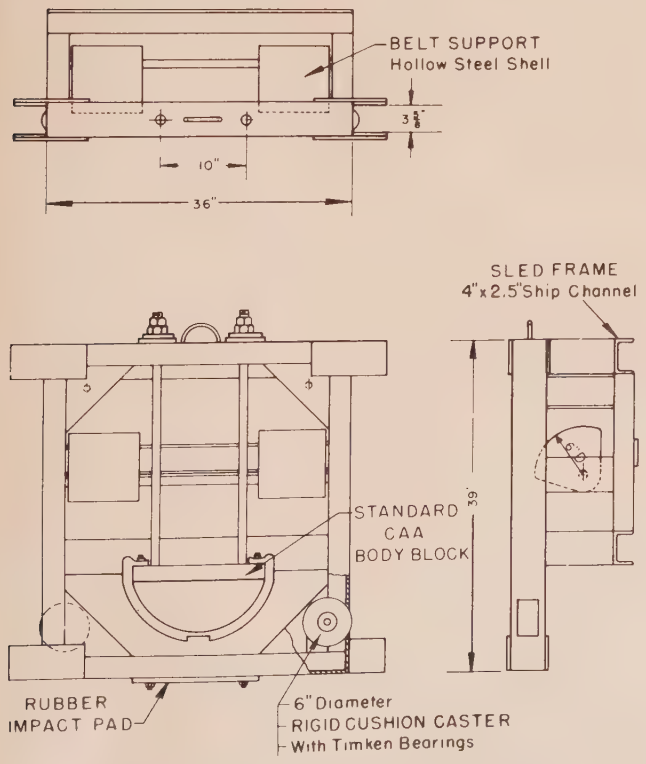


Fig. 2—Test sled of dynamic test machine

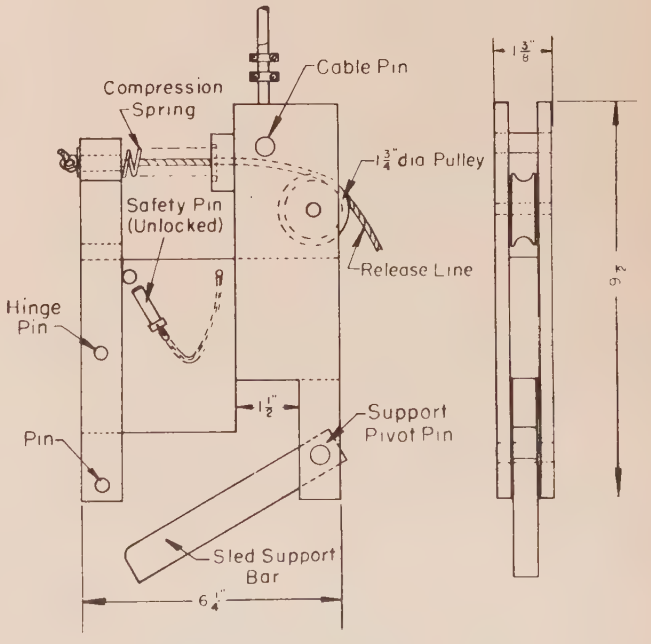


Fig. 4—Sled release mechanism

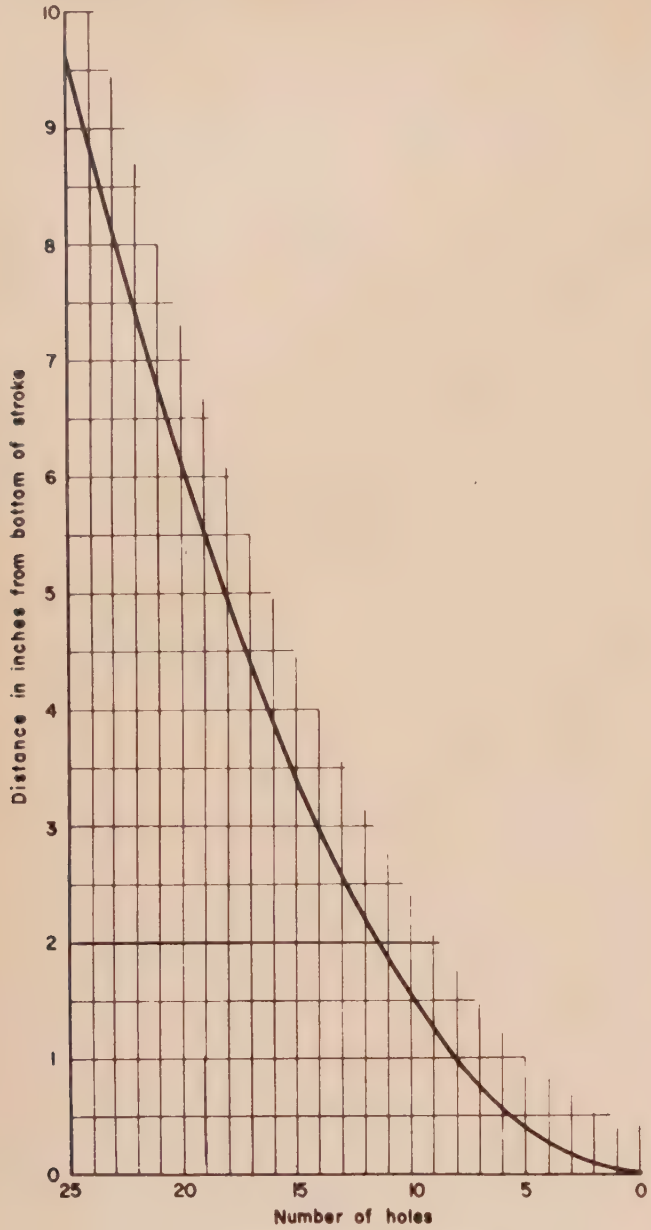


Fig. 5—Buffer orifice locations

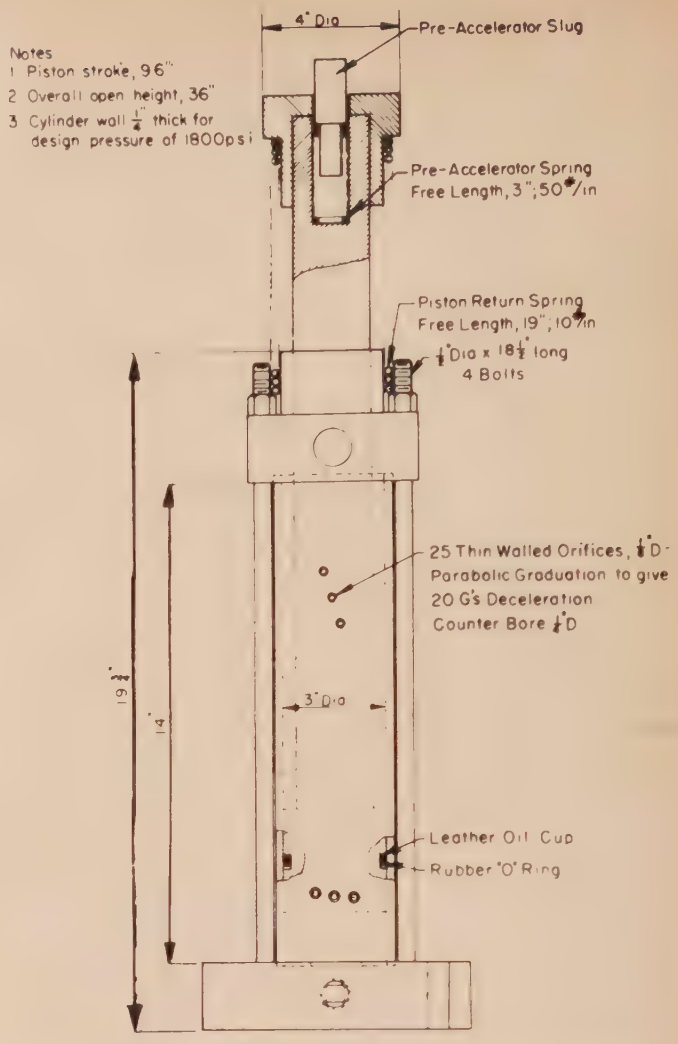


Fig. 6—Final design of oil buffer for dynamic test machine

3 in. The effective buffer stroke can be determined by first computing the velocity at the end of free fall, that is, velocity at the end of 16-ft free fall

$$V_o = \sqrt{2gh} = \sqrt{2 \times 32.2 \times 16} = 32.1 \text{ fps}$$

Distance required to decelerate a body moving with an initial velocity of 32.1 fps at the constant rate of 20 g

$$S = \frac{V_o^2}{2a} = \frac{(32.1)^2}{2(20g)} = 0.80 \text{ ft} = 9.6 \text{ in.}$$

The time required for uniform deceleration is

$$t = \frac{1}{a}\int_{v_o}^{0} v\,dt = \frac{1}{20g}\ (32.1) = 0.05 \text{ sec}$$

Using equations 2 and 3, the average fluid pressure developed in the buffer cylinder is

$$P_{ave} = \frac{Q}{A}\left[\frac{V_o^2}{2gs} + 1\right] = \frac{765}{\pi(1.5)^2}\ [20 + 1] = 2270 \text{ psi}$$

Assuming 20,000 psi as the allowable stress for steel, the cylindrical hoop stress equation gives the cylinder-wall thickness as

$$\tau = \frac{P_{ave}D}{2\sigma} = \frac{2270 \times 3}{2 \times 20{,}000} = 0.170 \text{ in.}$$

As a practical measure, the wall thickness was made 0.250 in. allowing a safety factor of approximately 1.5 for the cylinder.

Having specified the length of stroke, wall thickness of the cylinder, and pressure developed, it remains to specify the number and diameter of peripheral orifices necessary to produce a constant deceleration of 20 g.

1. Arbitarily select 25 as the desired number of holes and assume the openings act as thin-walled ori-

fices; using 0.61 as the coefficient of efflux c,[5] the velocity of efflux is

$$v_e = \sqrt{\frac{P_{ave}}{c}} = \sqrt{\frac{2270 \times 144}{0.61}} = 732 \text{ fps}$$

2. Since $v_e = \frac{AV_o}{fY}$, the diameter of the holes can be determined from

$$d = D\left(\frac{V_o}{v_e Y}\right)^{1/2} = 3.00\left(\frac{32.1}{732 \times 25}\right)^{1/2} = 0.141 \text{ in.}$$

As a practical design measure, 0.125-in. diameter holes were used initially.

3. To determine the location of the holes about the periphery the following equation is used

$$y^2 = Y^2\left(\frac{s}{S}\right)$$

The location of the holes is given in Table 1, and a plot of the hole locations is shown in Fig. 5. The drawing of the final buffer design is shown in Fig. 6.

To evaluate the deceleration of the sled and body-block test, two methods were used. The first was a photographic procedure using a Fastex high-speed camera. The developed film enabled the results of the test to be analyzed graphically and, when run at slow speed, the action could be observed. The second method used a rotating drum and a stylus actuated by the sled. The drum rotated at constant speed, and a direct plot of displacement versus time was recorded. Both methods provided usable data; however, the rotating drum was faster and simpler to analyze. The results of many tests show that the desired deceleration rate of 20 g has been very closely approximated over a period of 27 millisec and is well within the tolerances expected from the mechanical system. The results show that the reproducibility of any test is within $\pm$ 1%.

The decelerometer used to record the time displacement function of the test sled consists essentially of a rotating drum 12 in. long and 4 in. in diameter. The drive motor is a capacitor-start ⅓-hp motor with a speed of 1750 rpm. The stylus is connected directly to the piston of the oil buffer and draws the curve by means of carbon paper covered with graph paper. The coated side of the carbon paper is out so that the small pressure of the stylus is sufficient to cause a mark on the graph paper. The decelerometer is shown in Fig. 7.

A 4000 frame per sec Fastex camera was also used to record the deceleration rate of the test sled and that of the body block. The negative film contains timing marks and, together with direct displacement measurements, a time displacement curve can be drawn. The two methods mentioned above have both been used to analyze and correlate the deceleration of the test sled and body block. Deceleration curves as a function of time are obtained from the time dis-

[5] See p. 254: "Elementary Fluid Mechanics," by John K. Vennard. Pub. by John Wiley and Sons Inc., New York, 1949.

Table 1—Location of Peripheral Holes

Effective Number of Holes, y	y^2	Distance from Bottom of Stroke, s
25	625	9.6000
24	576	8.8473
23	529	8.1254
22	484	7.4342
21	441	6.7737
20	400	6.1440
19	361	5.5449
18	324	4.9766
17	289	4.4390
16	256	3.9321
15	225	3.4560
14	196	3.0105
13	169	2.5958
12	144	2.2118
11	121	1.8585
10	100	1.5360
9	81	1.2441
8	64	0.9830
7	49	0.7526
6	36	0.5529
5	25	0.3840
4	16	0.2457
3	9	0.1382
2	4	0.0614
1	1	0.0153
0	0	0

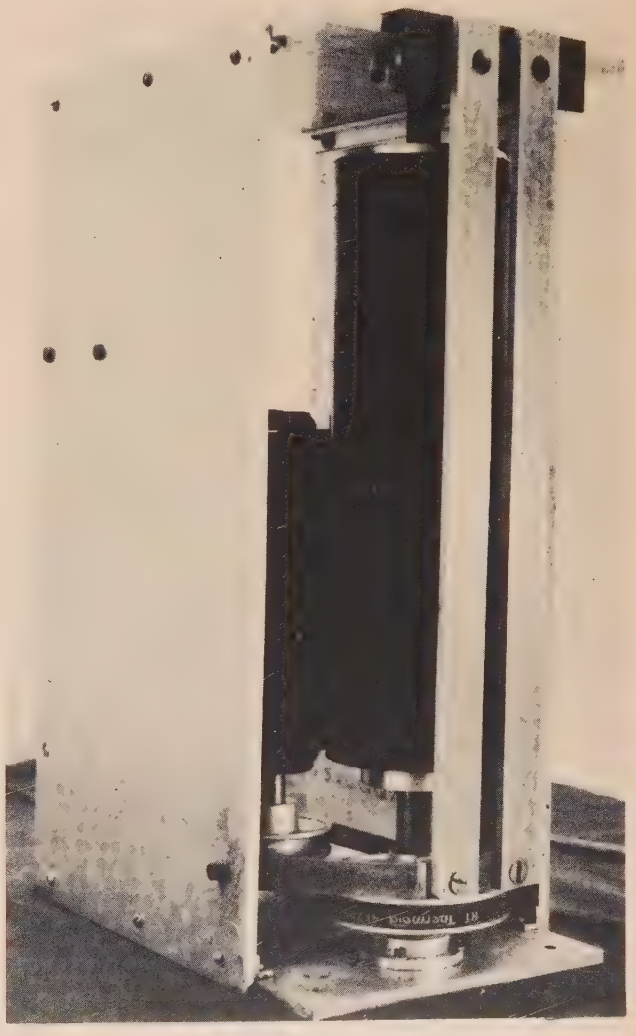


Fig. 7—Recording decelerometer

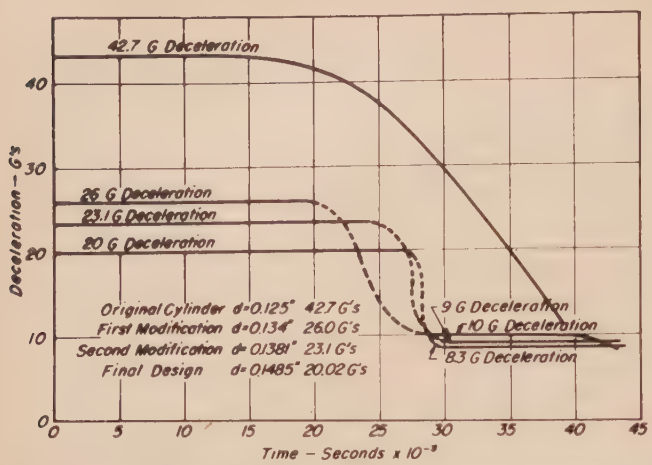


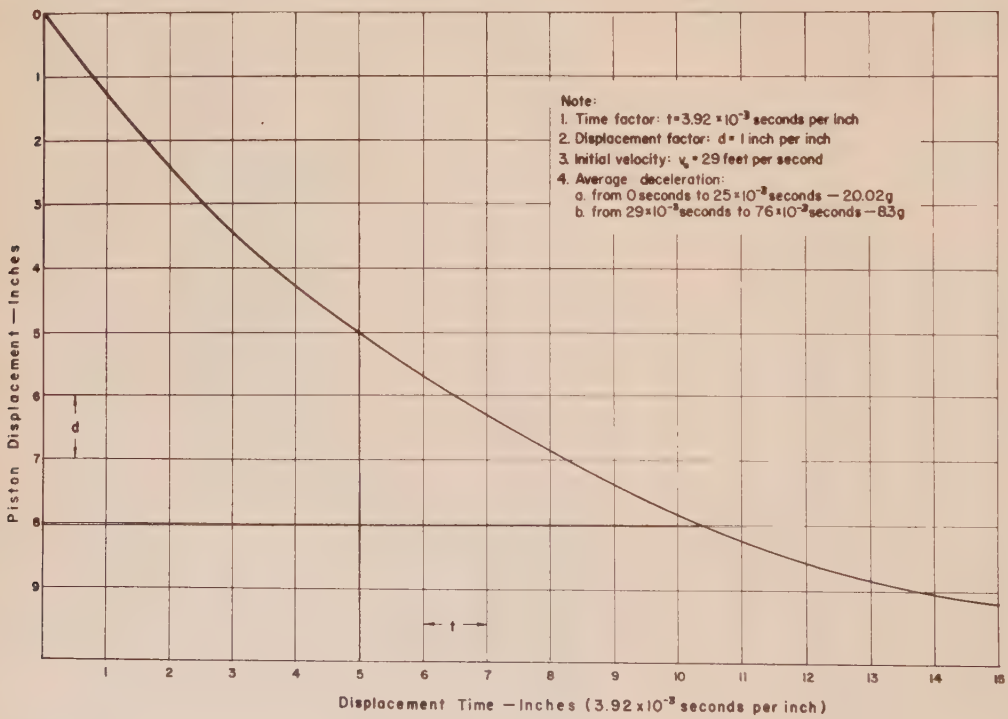
Fig. 8—Development of oil buffer to reach desired deceleration rate

placement curves by differentiating the latter curve twice.

Since differentiating a curve gives the slope at any one point, a graphical method was used. The slope of each curve was measured for several points, and the succeeding curve plotted from this calculation. A series of tests involving more than 100 actual runs have been performed in order to obtain the deceleration rate of the test sled. Several modifications have been made to the original design to obtain the desired rate of 20 *g*. Since all of the equations used in the design of the buffer involved a fluid with the viscosity of water, allowances had to be made to compensate for the higher viscosity oil used. Larger orifices were used. Water as a fluid would have presented the problem of rust and, therefore, was not used. Spindle oil (trade name of Velocite No. 3) supplied by the General Petroleum Corp. with a viscosity of 3.0 times that of water was used. Additional allowances had to be made to compensate for the friction of the runners and guide wheels of the test sled, which slowed the initial velocity from the theoretical 32.1 to 29 fps. This allowance was made by further opening the orifices. The final diameter of the orifice holes is now 0.149 in. With all of the factors taken into account, the effective stroke of the buffer is 8.3 in., and a constant deceleration rate of 20.0 *g* has been achieved over a time range of 27 millisec. The deceleration rate is an average of 8.3 *g* for the remaining 50 millisec.

After analyzing the results of many tests, it was found that belt failure occurred almost always near the end of the effective stroke of the buffer. During many of the tests the body block moved in the neighborhood of 6 to 8 in. with an average permanent set of from 1 to 3 in. for each end of webbing after the load was removed. All the belts that are tested are mounted, as closely as possible, in accordance with instructions supplied by the manufacturer.

Results of Deceleration Trials—The results of several decelerations taken from the recording decelerometer of the oil-buffer system during the first trial runs are shown in Fig. 8. The deceleration rate with



Time — Displacement Curve from Dynamic Test Machine Recorder

Fig. 9—Sample time-displacement curve from dynamic test machine recorder

the original design is seen to be 42.7 *g* over the first 15 millisec and then decreases to approximately 10 *g* over the remaining 25 millisec of the deceleration of the test sled. A change in the orifice area of approximately 15% correspondingly reduced the initial deceleration from 42.7 to 26 *g* over a time range of 20 millisec and then 10 *g* from 27.5 to 40 millisec. The last curve on this deceleration versus time plot represents the deceleration rate of the system as it is now operating. It can be seen that a total area change of approximately 30% was necessary to bring the deceleration down to the desired rate of 20 *g*. The oil-buffer system decelerates at the rate of 20 *g* for the first 27 millisec and 8.3 *g* from 30 to 70 millisec. A sample curve of the data taken by the recording decelerometer is shown in Fig. 9. The step-wise reduction of the displacement versus time data of Fig. 9 is shown in Fig. 10. The curves represent the displacement versus time function of the sled and body block, its differential curve (velocity) as a function of time, and the second order differential curve (deceleration) as a function of time.

In order to measure the deceleration rate of the body block with respect to ground, it was found necessary to film the action of the sled, body block, and oil buffer with a Fastax high-speed camera and then plot the displacement versus time curve of the body block with respect to ground from measurements taken directly from the film. The film was run at a film speed of approximately 3900 frames per sec, while a timing light, a 60-cycle discharge lamp, provided the necessary time constant for use in reduction of the data from the film.

The curves of Figs. 11 and 12 show the relationship between the displacement, velocity, and deceleration as a function of time for the body block with respect to ground. The curves of Fig. 11 are for an all-nylon webbing with metal-to-metal-type buckle, while the ones of Fig. 12 are for a nylon-cotton-nylon-type webbing with a friction-type buckle. The different deceleration rates over similar periods of time may be related directly to the spring constants of the two webbing samples. It can be seen from Fig. 11 that the deceleration rate from approximately 30 to 65 millisec is 35.8 *g*, and then there is a damped oscillation from 65 to 114 millisec. This period of damped oscillation represents an acceleration of the body block and then a corresponding deceleration. The energy of the damped oscillation expends itself entirely in the one full period. The maximum acceleration obtained during this period is 10 *g* and occurs at approximately 72 millisec from the start of the deceleration. The nylon-cotton-nylon shows a slightly

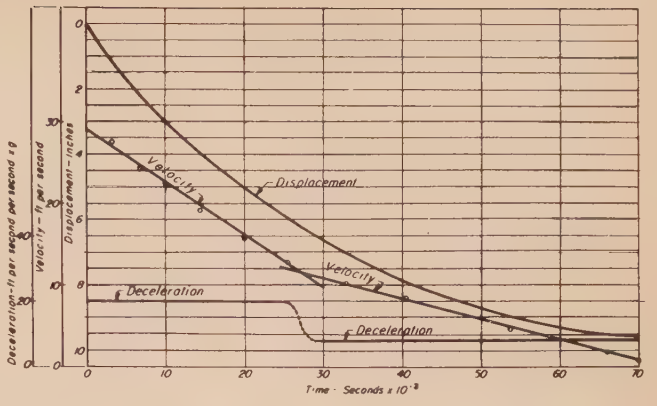


Fig. 10—Deceleration, velocity, and displacement curves for dynamic seat-belt test on test sled only

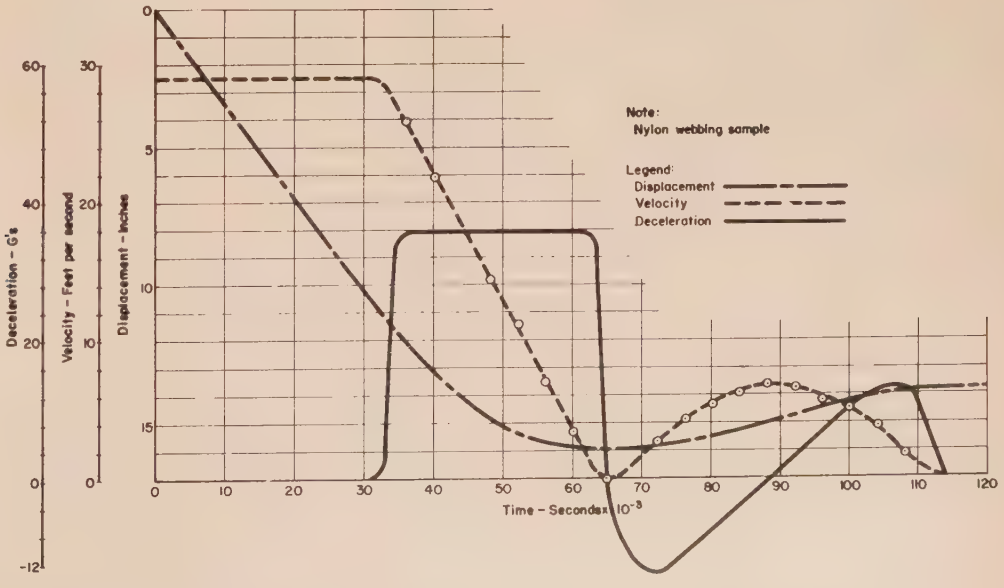


Fig. 11—Displacement, velocity, and deceleration versus time for body block with respect to ground for all-nylon webbing with metal-to-metal buckle

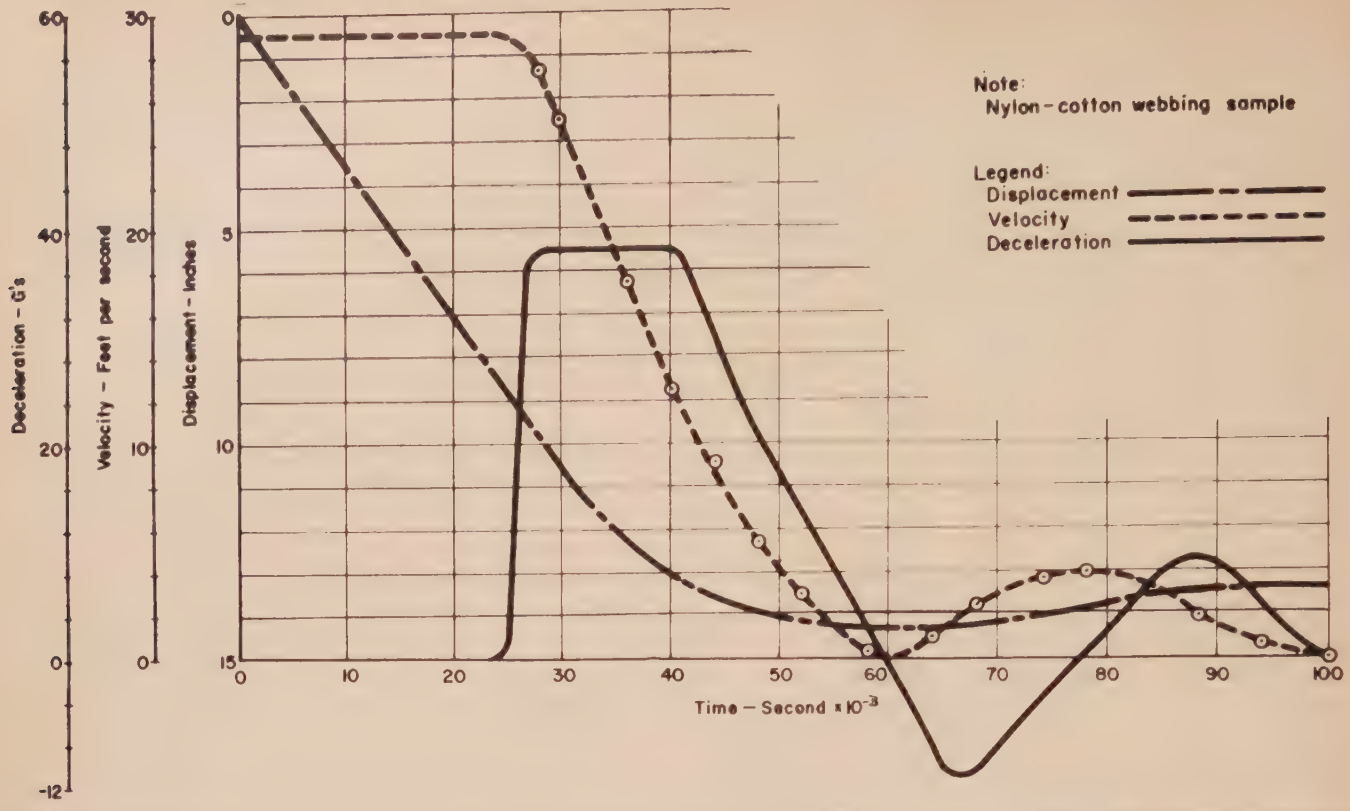


Fig. 12—Displacement, velocity, and deceleration versus time for body block with respect to ground for nylon-cotton-nylon webbing with friction buckle

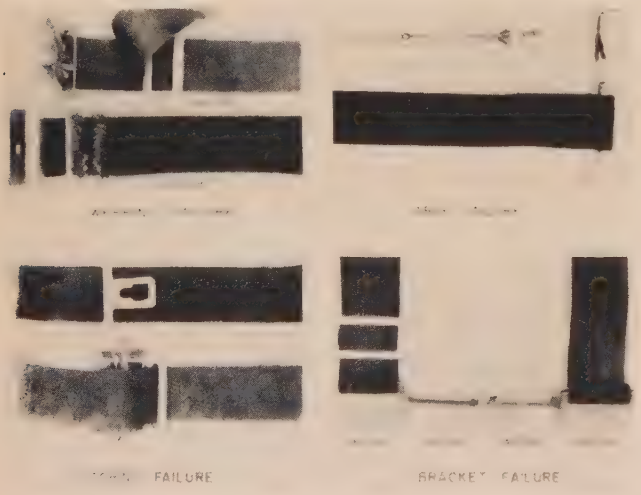


Fig. 13—Typical belt failures

higher deceleration rate over a shorter period, decreasing to zero. The total time of the period is equal to that for the all-nylon belt. The maximum deceleration rate over this period is 38 *g* and occurs between 28 and 41 millisec, then decreases to zero in the next 20 millisec. There is a damped oscillation from 60 to 100 millisec, and, again, the energy of the damped oscillation is also absorbed during the one period.

From the consideration of available energy it was noted that both the all-nylon webbing and the nylon-cotton-nylon webbing absorbed approximately 2550 ft-lb of energy. These absorbed energies are represented by the area under the deceleration curves, and the actual absorbed energy can be computed for each from the potential energy in the body block. Calculations made from the deceleration and displacement data indicate that the total available energy is absorbed during a time of approximately 36 millisec for both types of webbing used. The maximum belt load for the all-nylon webbing was 2690 lb, and the maximum load for the nylon-cotton-nylon webbing was 2750 lb. No data was compiled for other types of webbing, since the data had to be taken from film, and the labor involved is quite substantial.

No attempt has been made to date to determine the effect of oscillations upon an occupant of the seat belt when subjected to damped oscillations.

Tests Performed on Seat Belts

From the many types of seat belts that have been tested on the dynamic test machine, it has been found that failures are generally due to the following causes: (1) rough edges of the mounting brackets, (2) slippage of the webbing through the buckle or mounting assembly, (3) in some instances actual fracture of

the metal components, or (4) in stretching of the webbing. Some of the typical failures that have occurred in seat-belt assemblies are shown in Figs. 13 and 14. A critical survey of the pictures seems to indicate that in many cases the mounting hardware is actually providing a sharp edge upon which the webbing is cut when the body block is suddenly decelerated. This type of failure of the seat belt can be eliminated entirely. A number of manufacturers have done so by the use of a heavier gage metal for the mounting bracket and by a more reliable deburring method to insure removal of all sharp edges around the slots. The failure of the webbing in the buckle is due to the excessive upper jaw clamping force on the webbing as a result of the sudden deceleration. This type of failure is usually attributed to sharp edges on the friction-type buckle which cut into the webbing. Such failures can be eliminated by designing the buckle to have more surface area in contact with the webbing and by removing the sharp edges of the upper jaw. The failures caused by fracture of the metal components of the seat belts are due only to the use of insufficient cross-sectional area of the slots and bar pieces or to the use of low-tensile-strength metal not suited for the job.

The types of failures that have been observed in the seat belts are as follows:

1. Webbing failure in one or both mounting brackets.

2. Webbing failure in the jaw of friction-type buckles.

3. Webbing slipping through mounting bracket or buckle.

4. Mounting-bracket fracture in the thin sections or in sections where sharp corners are produced during the stamping process.

5. Mounting bracket failure due to inadequate design.

6. Upper jaw of the buckle opening and allowing the free end of the webbing to slip out (friction-type buckles).

7. Buckle failure due to insufficient strength or poor design of the metal components of the buckle.

8. Buckle failure in the metal-to-metal type due to the locking piece opening upon impact.

9. Stitching failure due to improperly designed stitching pattern.

10. Fracture of the mounting bolt.

11. Fracture of cables used to secure the seat belt.

Most of the types of failures mentioned above can be eliminated by a redesign of the part in question,

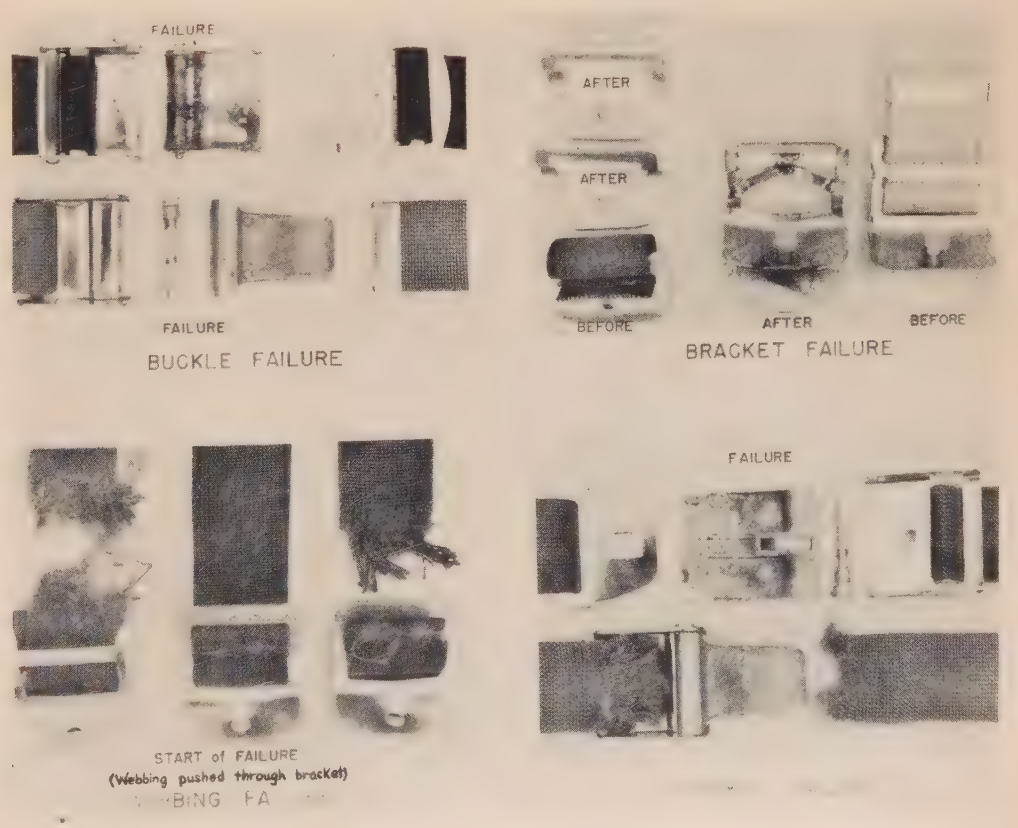


Fig. 14—Typical belt failures

usually by simply using larger cross-sections or rounding off sharp corners.

Conclusion

The testing of seat belts with the dynamic test machine has led to the conclusion that this test is the most valid of all the tests performed on seat belts. This particular test indicates the preformance of the belt when the body to which it is attached is subjected to a sudden impact-type loading with a constant deceleration of 20 *g* over a short period of time and a deceleration of 8.3 *g* over the remainder of time after impact. From figures obtained, there is an indication that the test is not too severe for crashes that involve relatively high speeds. There is no set criterion for the number of *g*'s that any given seat belt should be able to sustain in dynamic loading; however, the 20-*g* figure seems to represent a fair test for automotive seat belts. It has been found from the results of many tests performed on seat belts that approximately 35% of those originally submitted could meet the requirement of 20-*g* dynamic loading. It has also been found that, in general, belts that fail the static tests will also fail the dynamic tests. This, however, is no assurance that any belt passing the static tests will pass the dynamic test, since many of the belts that have met the requirements of the static tests have failed in dynamic loading. No direct correlation can be made at this time between the static test results and the dynamic test results. The dynamic test is more severe than the static tests that we use. It reveals some weaknesses that are not apparent by static testing, so we feel it is a more representative test, as well as simpler and quicker to perform.

Acknowledgments

Acknowledgment is made of the assistance given by F. B. Henry of the Industrial Accident and Safety Commission during the design of the oil buffer and throughout the trial runs; of the able assistance rendered by the entire testing agency crew and R. E. Cooley and C. W. LeRoy in particular during the design of the dynamic test, actual testing of seat belts, and in compiling the results of this report.

APPENDIX

Static and Corrosion Tests on Seat Belts

Tensile Test on Webbing

Samples of webbing are tested in a Universal Testing Machine using a 24-in. length of sample mounted in CAA-type grips which have a wrap-around design to eliminate localized loading in the grips. A 5-in. gage length is obtainable when a 24-in. sample is used. The tensile test is conducted at a loading rate of 2-in. per min up to 2250 lb. The elongation is checked as a function of load. The 2250-lb load is maintained for 3 sec to determine the elongation in inches per inch. One sample is tested to develop the ultimate load.

Buckle-Release Test

The complete belt assembly, including buckle and anchor brackets, is mounted in a Universal Testing Machine using a CAA-type body block and a suitable yoke to load the belt as a loop with two parallel webs in tension. Load is applied to 2850 lb, released to 250 lb, and held at 250 lb. A spring balance is connected to the end of the buckle-release mechanism to measure the release force. The release force is in a direction to simulate actual operation and is approximately normal to the axis of the buckle release lever. The buckle is required to release within a range of 5-lb minimum–45-lb maximum.

Tensile Test on Belt Assembly

The complete belt assembly, including anchor brackets and buckle, is mounted between the heads of a Universal Testing Machine as a single section of webbing (not a loop test). The tensile test machine heads are separated at the rate of 2 in. per min to develop a maximum of 1500 lb on the belt assembly and then released. The total slippage in the buckle and anchors is measured. Also, the total increase in belt length between the heads is determined (slippage plus webbing stretch). A second sample is mounted in the same manner and tested for ultimate load.

Corrosion Test

The device is suspended in a closed tank (ASTM Specification B117–44T) and subjected to a saline

and fuel flow at wide-open throttle would drop. When this occurred the vehicle was vapor locked. A run with at least one fuel having a volatility higher than the fuel which first gave vapor lock was always made.

Results and Discussion

Estimation of Pump Stroke—In previous studies on automotive fuel pumps made at the Research Staff [2] the pressure fluctuations in the pump cavity were used to estimate the amount of stroke taken by the pump. It was assumed that the pump diaphragm suction stroke began when there was an abrupt pressure drop and ended when the pressure reached a minimum value. The percentage of full stroke was estimated from the number of cam angle degrees required for this suction stroke, the geometry of the pump linkage, and the cam angle degrees required for a full stroke. There has been some doubt of the validity of this approach, and to check it simultaneous pump cavity pressure and diaphragm movement traces were taken.

With a volatile fluid being pumped through the bench setup the traces shown in Fig. 7 were obtained. The diaphragm movement trace is at the top; the pump cavity pressure fluctuation is at the bottom. It may be seen that the pressure reached a minimum *before* the diaphragm reached the bottom of the stroke. As will be shown, this is a most important observation.

From the pressure trace in the figure and using the method described previously,[2] it would be estimated that the pump was taking 48% of full stroke. However, using the calibrated diaphragm movement trace, the pump appears to be taking 92% of full stroke. At a slightly different operating condition, an even more striking example of the possible discrepancy between the two methods of estimating the stroke may be seen in Fig. 8. From the pressure trace only 7% of full stroke would be estimated, while the diaphragm movement data give 81% of full stroke.

It may be concluded that as a result of the phase shift, the pump stroke cannot be estimated from the pump cavity pressure fluctuations. Therefore, the interpretations of diaphragm movement by Caplan and Wilson [2] based on the pump cavity pressure fluctuations must be reconsidered. In fact, in all of the current series of tests it was found that at vapor lock the pump was taking a full stroke.

Phase Shift Between Pressure and Diaphragm Movement—As has been shown in Figs. 7 and 8, the pressure was considerably out of phase with the diaphragm movement. That is, when the pressure reached a minimum, the diaphragm was still moving downward. When the diaphragm reached the bottom of the stroke, the pressure already had risen above its minimum value. The reasons for this phase shift were not immediately apparent, and this prompted a detailed investigation of the relationship.

It was first thought that this phase shift might be due to flash vaporization in the pump cavity, because there was no phase shift observed with a low volatility fluid. However, the fact that a phase shift existed when air was pumped eliminated this possibility.

1. Phase Shift Studies With Air: Additional insight into the problem was obtained by observation of the phase shift with air. In Fig. 9 the effect of length of inlet line upon the phase shift may be seen. The top two simultaneous traces were taken when the entire fuel system was intact, the second two traces when only the 16-in. flexible hose was connected to the pump inlet, and the third two traces when the pump inlet was directly open to atmosphere. The amplification is identical for the three sets of traces, and as in the previous two records the pressure trace is the lower of the two traces in each set. *Measurements from the traces show that shortening the line reduced the amplitude of the pressure fluctuations in the pump cavity*. However, the phase relationship did not seem to be appreciably affected. This observation appears to eliminate an explanation of the phase shift being due to ram-inertia effects from pressure waves in the inlet line.

In Fig. 10 are shown the effects of increasing the inlet restriction on the phase shift. It may be observed that as the restriction was increased the phase shift was reduced. As would be expected, when the inlet was completely restricted, a simple pressure-volume relationship existed, and no phase shift was observed. Any increase in volume was accompanied by a de-

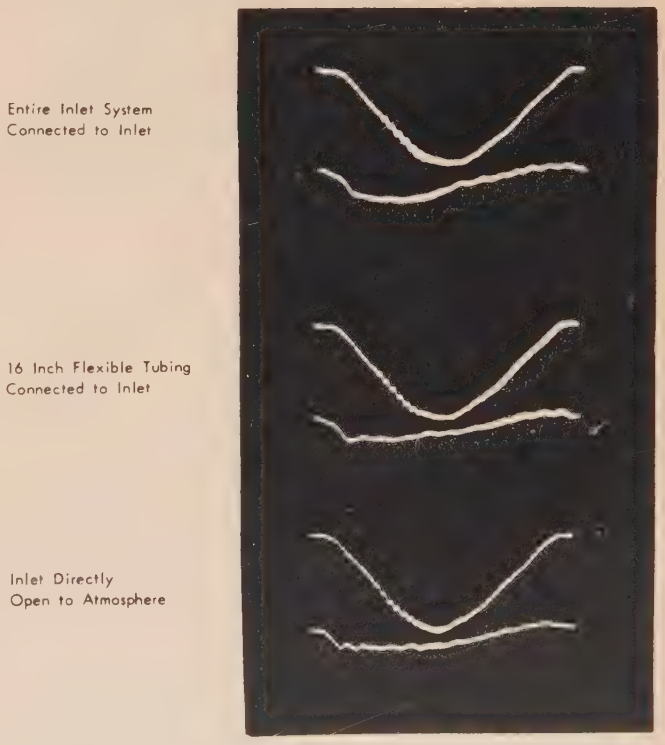


Fig. 9—Effect of inlet line length on pump diaphragm movement and cavity pressure fluctuations when air is being pumped

Table 2—Flow Characteristics with Low-Volatility Fluid

Pump Cover	Cam Speed, rpm	Ratio of Calculated to Measured Volume Flow	Measured Weight Flow, lb per min
A	300	2.5	3.2
	600	4.5	3.6
	1200	11.0	3.3
	1800	14.9	3.3
B	300	2.0	3.8
	600	5.3	3.8
	1200	10.8	3.7
	1800	18.0	3.8
C	300	2.2	3.7
	600	5.4	3.5
	1200	11.4	3.5
	1800	16.7	3.8
D	300	1.8	3.5
	600	2.5	3.9
	1200	5.4	3.3
	1800	8.9	3.4

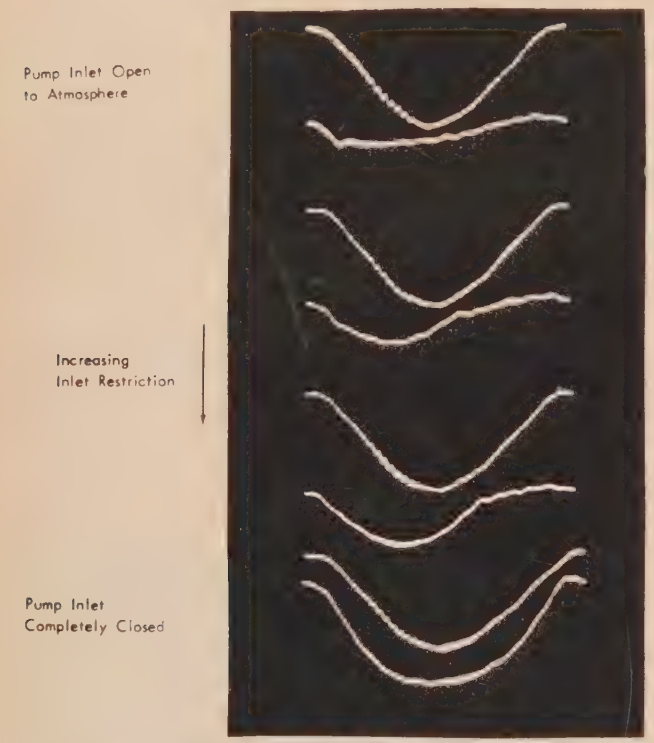


Fig. 10—Effect of inlet line restriction on pump diaphragm movement and cavity pressure fluctuations with air being pumped

crease in pressure, and the pressure minimum occurred at the bottom of the stroke.

With the pump inlet open to the atmosphere the only restriction to flow which can cause a reduction in the pump cavity pressure is due to the inlet valve. The maximum pressure drop across the valve will occur when the flow is the greatest, which should be when the diaphragm velocity is a maximum. This maximum pressure drop should determine the time of minimum cavity pressure, since the inlet pressure is constant at atmospheric pressure. It may be seen in the two figures that with the slightest inlet restriction the pressure minimum occurs very close to the point of maximum diaphragm velocity.

Another factor contributing to the phase-shift relationship becomes evident when pressure fluctuations in the inlet line and pump cavity are compared. Fig. 11 shows a series of simultaneous inlet and pump cavity pressure traces obtained with various inlet line restrictions. As before, the lower trace in each set is the cavity pressure fluctuation. As the inlet restriction was increased, the start of the decrease in the inlet line pressure occurred later and later. With more restriction in the inlet line, there is a lower pressure ahead of the inlet valve. As a result, the cavity pressure must decrease further to provide the pressure differential needed to open the inlet valve. With the inlet line completely closed, a sufficient vacuum exists in the inlet line to prevent the inlet valve from opening, and no inlet line pressure fluctuations are noted.

This variation in inlet line pressure also contributes to the phase shift in another manner. This is because the cavity pressure is determined not only by the pressure drop across the inlet valve due to flow, but also to the inlet line pressure. Inasmuch as the maximum pressure drop corresponds to maximum diaphragm velocity only with constant inlet pressure, this variable inlet pressure may shift the minimum cavity pressure from the point of maximum diaphragm velocity.

In summary, the observations with air reveal interacting effects which determine the time of minimum pressure in the pump cavity. The pressure-volume effect of a closed system tends to make the pressure reach a minimum at the bottom of the stroke. The flow-pressure drop effect with a constant inlet pressure will tend to make the minimum pressure occur at the point of maximum diaphragm velocity. A variable inlet pressure further complicates the phase relationship. With intermediate amounts of restrictions there is a combination of these effects, and the pressure minimum will lie somewhere between the point of maximum diaphragm velocity and the bottom of the stroke.

2. Phase Shift Studies With Liquids: In the preceding paragraphs the reasons for the phase shift when pumping air have been discussed. The same phase shift occurred when pumping a volatile fluid at a moderately high temperature (100–120 F). Thus, the evidence indicates that the fluid behaves as a vapor. This result is not unexpected because the V/L is sufficiently high. While flash vaporization may make a contribution to the phase shift, the same explanations of the phase shift as given for air would apply.

With a low-volatility fuel the minimum pressure occurred at the bottom of the stroke as shown in Fig. 12. Apparently, the relative incompressibility of the fluid, the increased viscosity of the fluid, and inertia effects will tend to create a situation similar to a closed system and make the pressure minimum occur at the bottom of the stroke.

A check was made to see if the phase shift could be changed by reducing the inlet line restriction. Using the low-volatility fluid the supply can was moved close to the pump inlet with only 8 in. of line between the can and the inlet. The resulting simul-

taneous diaphragm movement trace at the top and pressure fluctuation at the bottom is shown in Fig. 13.

It may be seen that the pressure minimum occurred before the bottom of the stroke. Thus, with the reduction of the inlet restrictions and inertia effects the system behaved less like a closed system, and a slight phase shift occurred.

The phase-shift studies with air and liquids show that the phase relationship between cavity pressure fluctuation and diaphragm movement may be explained on the basis of pressure drops and restrictions in the inlet system. Moreover, these studies provide no evidence which confirms or denies the theory that flash vaporization occurs in the pump cavity. A mathematical treatment of this phenomenon is beyond the scope of this paper.

Runs With Low-Volatility Fuel—Runs were made with a low-volatility fuel to determine the difference between the flow calculated from the diaphragm-movement data and the measured average flow with liquid in the system. The results of the runs are listed in Table 2 for the four pump covers at four speeds. The carburetor needle valves in these tests were wide open. The ratio of the volume flow calculated from diaphragm-displacement data to measured volume flow is listed along with the measured weight flow in pounds per minute.

In all cases the ratio of calculated to measured flow increased as the speed was increased. With the exception of pump cover D at the three higher speeds, the values are quite similar for the covers. The reasons for the lower ratios with pump cover D are not known, and are not apparent from weight flow rates or the ratio of calculated to actual flow rates with air for this pump cover. (These data will be given later.)

The diaphragm moved several times the distance needed to displace the net amount of liquid flowing. Little bubble formation was observed in a section of transparent tubing in the inlet line which would rule out vaporization in the inlet line. There is, of course, the problem of air dissolved in the fuel which could escape in the pump cavity and introduce vapor into the fuel system. It is difficult to visualize the amount of air needed to give this effect of being dissolved. The final possibility is that of appreciable back flow in the system. Possibly due to improper seating of the valves, part of the volume displaced by the diaphragm would fail to appear as net flow. Whether or not the difference between the displaced volume and the net flow is due to backflow, it is interesting to note the wide difference between these two values.

V/L Determinations—Despite the evidence of appreciable backflow, it has been possible to calculate the V/L ratio in the pump under selected conditions. From the geometry of the pump diaphragm it is possible to calculate a theoretical volume displaced for any given diaphragm movement. With the measured weight flow and the densities of the vapor and the liquid at the operating temperature, it is possible to set up two simultaneous flow equations, volume and

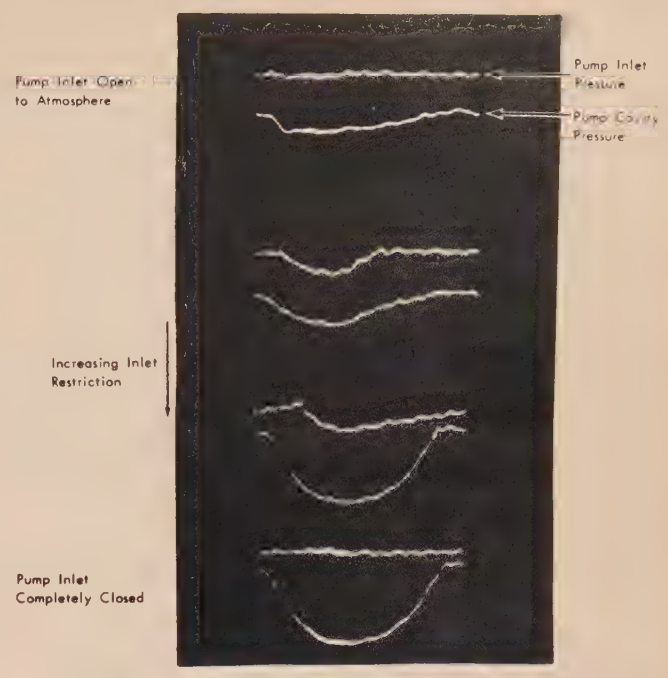


Fig. 11—Effect of inlet line restriction on pump inlet and cavity pressure fluctuations with air being pumped

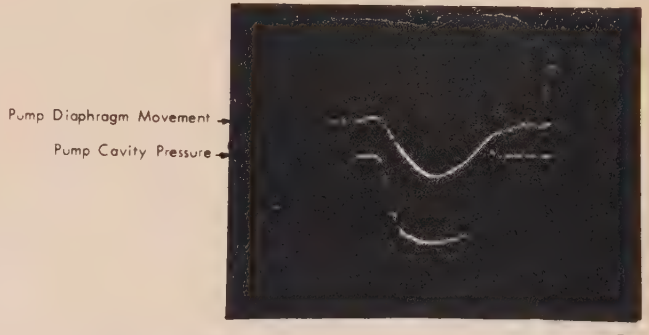


Fig. 12—Pump diaphragm movement and cavity pressure fluctuations with low-volatility fluid being pumped

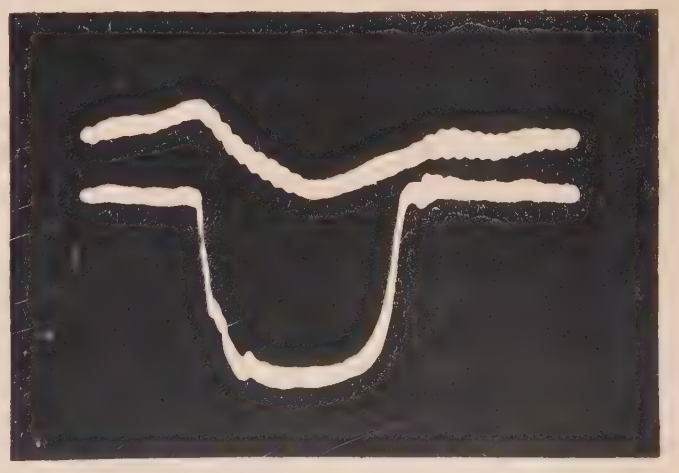


Fig. 13—Effect of reduction of inlet line length on pump diaphragm movement and cavity pressure fluctuations with low-volatility fluid being pumped

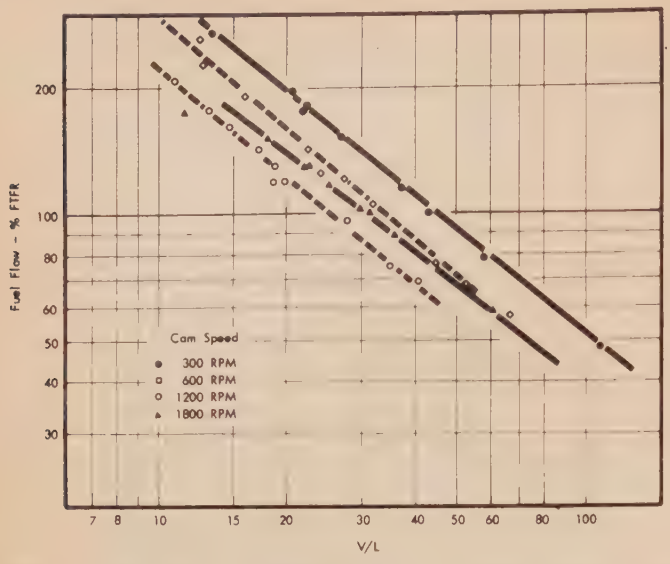


Fig. 14—Fuel flow as function of V/L for pump cover A on bench setup

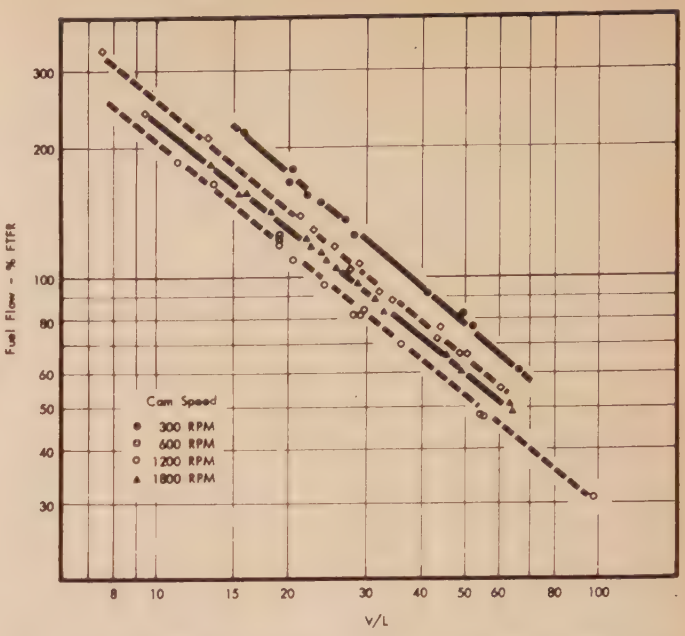


Fig. 15—Fuel flow as function of V/L for pump cover B on bench setup

Table 3—Comparison of Calculated to Measured Airflow

Pump Cover		Ratio of Calculated to Measured Airflow			
	Cam Speed, rpm	300	600	1200	1800
A		1.114	1.182	1.501	1.634
B		1.203	1.271	1.606	1.750
C		1.195	1.225	1.635	1.743
D		1.198	1.296	1.556	1.752

Table 4—V/L Values at "Vapor Lock" on Bench Setup

Pump Cover		V/L at "Vapor Lock"			
	Cam Speed, rpm	300	600	1200	1800
A		44.5	33.5	26.2	30.5
B		38	31	23.5	27
C		37.5	31	23	26
D		37.5	29	22	25

weight flow, and solve for V/L. The equations used for calculation of the V/L are:

$$V\rho_v + L\rho_L = F_w \qquad (1)$$

$$V + L = F_v \qquad (2)$$

and solved

$$L = \frac{F_w - \rho_v F_v}{\rho_L - \rho_v} \qquad (3)$$

$$V = F_v - L \qquad (4)$$

where

V = Volume of vapor
L = Volume of liquid
ρ_v = Density of the vapor
ρ_L = Density of the liquid
F_w = Weight flow through the system
F_v = Calculated volume flow through the system based on diaphragm movement

It may be shown that the selection of the absolute pressure will have little effect on the calculated V/L in the range encountered near vapor lock. The pressure will affect only the density of the vapor and a change of ± 2 psi will affect the V/L by only ±1.5%. Similarly, the selection of the molecular weight of the vapor will affect the density of the vapor only slightly. A change of ± 10 in the molecular weight at a value of 80 will affect the V/L by ± 1.5%.

An assumption which is made in the equations is that the calculated volume flow F_v is equal to the actual volume flow. As has been pointed out earlier, this is not the case with liquid flow. However, since at or near vapor lock there was 20 to 40 times as much vapor as liquid, it was assumed that corrections obtained when pumping air would be more valid.

A positive-displacement dry-test gas meter was installed instead of the 5-gal supply can. The results of these runs with air are summarized in Table 3 in which the ratios of the calculated air flows based on the pump displacement data to the actual measured airflows are listed for several speeds and the four pump covers. At a cam speed of 300 rpm the ratio is slightly greater than one which could be due to an error in the estimation of volume displaced per unit length of travel of the diaphragm. As the speed increased, however, the ratio increased. This speed effect would indicate a possible backflow due to improper seating of the valves. With pump cover A the ratios were always smaller than for the other covers. The one design feature which might affect this ratio was that the valves were considerably smaller in cover A than in the other covers.

The ratios in Table 3 were used to correct the volume calculated from displacement data. Thus, the correction is made by substituting F_{vc}, corrected calculated volume flow, for F_v.

The results of the V/L calculations for the four

pump covers tested are plotted in Figs. 14, 15, 16, and 17. The weight flow expressed as a percentage of the engine full-throttle fuel requirement at the speed corresponding to the given cam speed is plotted as a function of the V/L on log-log paper.

It may be seen that the points for a given speed with each of the covers describe a straight line on this type of plot. The lines may be expressed in the equation:

$$\% \text{ FTFR} = a\left(\frac{V}{L}\right)^{b} \qquad (5)$$

Since the lines are displaced one from the other the values of a, which is the intercept when $V/L = 1$, will vary considerably. The slopes of the lines, b, were nearly the same for all of the runs with all covers averaging -1.17 with a maximum variation of $\pm 4\%$.

Of particular interest is the value of V/L at 100% FTFR. This point, 100% FTFR, has been defined as the point of "vapor lock." The fuel flow at this point is just able to satisfy the engine full-throttle fuel requirement, and, theoretically, any less flow will give vapor lock. The values at this 100% FTFR have been read off the figures and are listed in Table 4.

Overall, the values range from 22 to 44.5. The lower speeds had the higher permissible V/L at 100% FTFR. This means that at the lower speeds the pump will tolerate a larger volume of vapor while supplying the required weight flow. A minimum appears to be reached at about 1200 rpm when the V/L at 100% FTFR ranged from 22 to 26. The spread in values from cover to cover was not large. Previous vapor lock testing with these four pump covers on the road and on the bench have shown little differences among their vapor-locking tendencies.

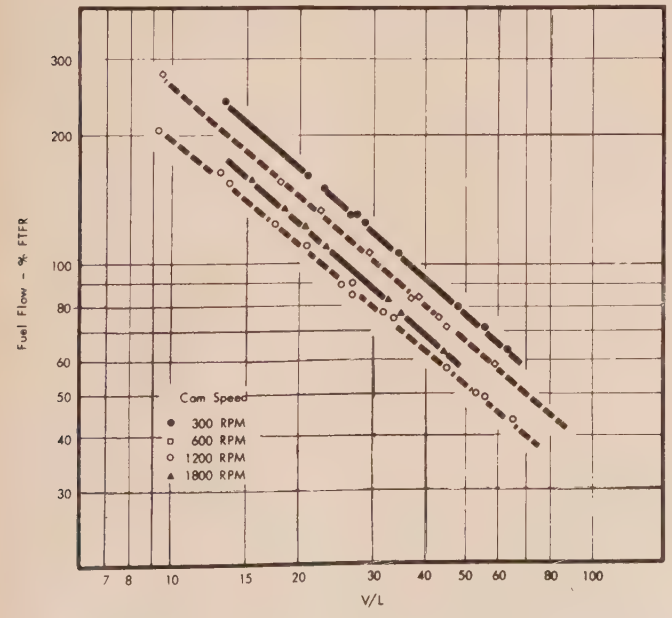


Fig. 16—Fuel flow as function of V/L for pump cover C on bench setup

Cover A gave consistently somewhat larger V/L at 100% FTFR than the other covers. One factor, while it does not explain the difference, may be related to this observation in that with cover A, the pump actually pumped an amount of air nearer the theoretically calculated amount than it did with the other covers.

Tests on Vehicle on Chassis Dynamometer—Runs were made with a vehicle on a wind-tunnel chassis dynamometer to determine the limiting V/L at vapor lock. The results of these runs are given in Figs. 18 and 19 for the two speeds along with the data at comparable cam speeds obtained on the bench setup. The plots are similar to those obtained for the different covers on the bench. However, with the vehicle, the fuel flow never could exceed 100% of the FTFR, hence, the break in the curve. The intercept of the

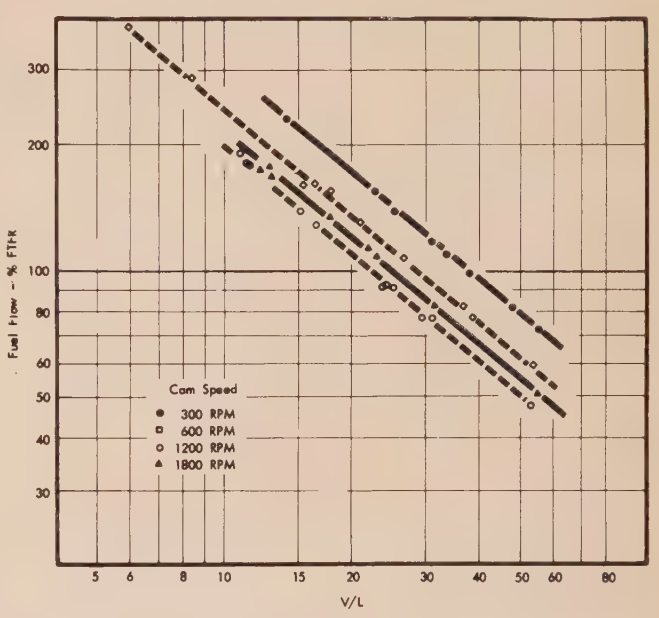


Fig. 17—Fuel flow as function of V/L for pump cover D on bench setup

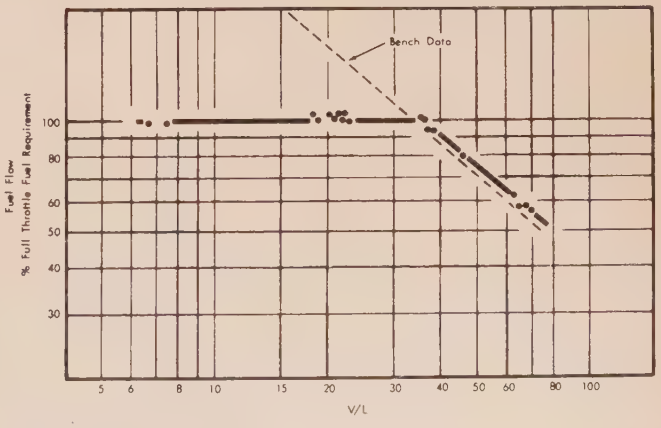


Fig. 18—Fuel flow as function of V/L for vehicle on wind tunnel chassis dynamometer with pump cover A run at 600-rpm cam speed

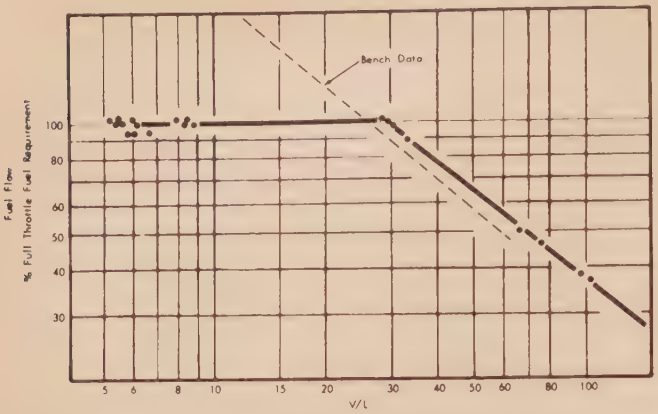


Fig. 19—Fuel flow as function of V/L for vehicle on wind tunnel chassis dynamometer with pump cover A run at 1200-rpm cam speed

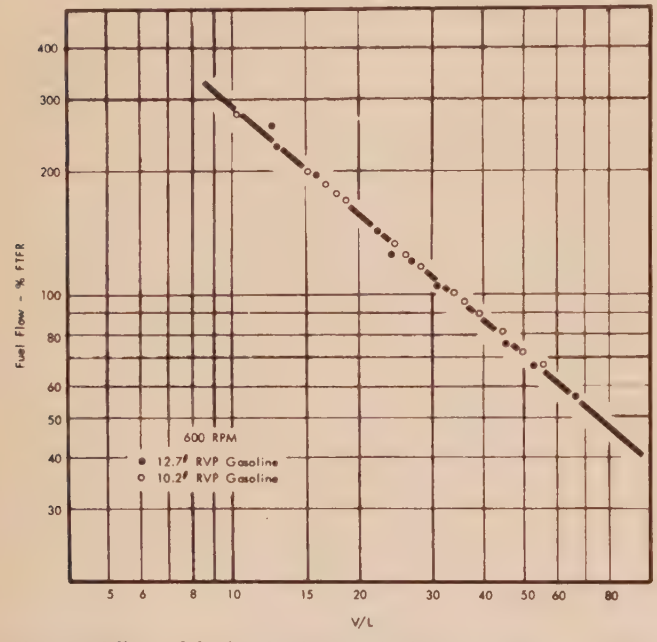


Fig. 20—Effect of fuel volatility on fuel flow and V/L relationship for pump cover A on bench setup

Table 5—Comparison of "Vapor Lock" V/L—Bench Setup to Vehicle

Cam Speed	V/L at Vapor Lock	
	Bench Setup	Vehicle
600	33.5	36
1200	28	29

curve with 100% FTFR is of particular interest.

In Table 5 is a comparison of the V/L at "vapor lock" in the bench setup and in the vehicle. There is excellent agreement between the data at comparable cam speeds.

The vehicle tests confirmed another observation of the bench tests in that the minimum pump cavity pressure occurred earlier than the bottom of the stroke.

It is interesting to compare these measured V/L values determined on a bench and on a vehicle with other reported values. Heath, Thena, and Way [3] give a "limiting V/L ratio of 26 . . . believed to be representative of a large part of the vehicle population." The measured V/L ratios on both the bench setup and the vehicle on the chassis verify their estimated values. Of course, the measured V/L ratios reported here are on only one fuel system.

Effect of Fuel Volatility on V/L—In running these tests it was assumed that over a range of fuel volatilities and operating temperatures the V/L at 100% FTFR would be independent of the fuel volatility. This assumption was verified by bench setup results obtained with two fuels of different volatilities at one speed in cover A. The two fuels run had Rvp of 12.7 and 10.2 psi and are described in Table 1. The results of the runs are shown in Fig. 20. It may be seen that there is excellent agreement between the results obtained with the two fuels.

Effect of Outlet Restrictions on V/L—Cover A was run at 600 rpm with varying amounts of outlet restrictions in order to determine the effect of these restrictions on the V/L ratio. To obtain a condition of increased restriction, the secondary needle valve in the carburetor was closed, and the primary needle was left ⅛ open. For the condition of least restriction, the ¼-in. pipe outlet was enlarged to ⅜ in., and connected with a large flexible hose directly to the condenser.

The results of the V/L determinations for these conditions are plotted in Fig. 21 along with the two runs with standard restrictions at 600 rpm. It may be seen that as the outlet restriction decreased, the V/L at 100% FTFR moved from 30 to 34 to 45. Thus, it appears that by decreasing the restriction in the outlet line the vapor-handling capacity of the system can be increased.

Vapor-Handling versus Vapor-Forming Characteristics of Fuel Systems—The results of all these V/L determinations brings up an interesting point which is connected to the basic concepts of vapor lock. In order to prevent vapor lock, the weight of fuel equivalent to the full-throttle fuel requirement of the engine must be supplied. It appears that if this weight can be supplied at a larger volume flow, or a larger V/L, the vapor-handling capacity of the system can be increased at vapor lock. One way of obtaining a larger volume flow with a given pump is to run the pump at a higher speed.

For example, with pump cover A it may be established that the FTFR for a cam speed of 600 rpm may be pumped at a V/L of 58 if the pump is run at 1200 rpm. In other words, the pump should still prevent "vapor lock" even though the V/L of 34 which was limiting at a pump speed of 600 rpm has been exceeded. Therefore, the vapor-handling capacity of the system has been increased by running the pump at engine speed rather than cam speed.

[3] "Passenger-Car Vapor Lock," by D. P. Heath, R. H. Thena, and G. Way. Presented at SAE Annual Meeting, Detroit, Jan. 11, 1955.

In the vehicle on the chassis dynamometer it was found that the FTFR at an engine speed of 1200 rpm could be pumped at a V/L of 67 if the pump was also operated at 1200 rpm. Operating at the cam speed of 600 rpm the pump could not supply the FTFR above a V/L of 36.

Of course, a possible fallacy in this reasoning is that in improving the vapor-handling capacity by going to a higher pump speed, the vapor-forming tendency of the system may have been changed. The change has probably been in the direction of increasing the vapor-forming tendency. Thus, at a given ambient condition with a given fuel a gain because of an increase in vapor-handling capacity may be partially offset by a loss due to an increase in the vapor-forming tendency. It would be interesting to determine the effect of increasing the pump speed from cam to engine speed on the vapor-locking tendency of a vehicle on the road. The customary acceleration-type vapor lock tests on the road add the complicating effects of transients and float bowl capacity which are absent from these steady-state bench and chassis tests.

One other item which must certainly be considered in the practical sense is that the carburetor may not be able to handle a V/L of, for example, 58 even though the weight flow should be sufficient to supply the engine demand. In fact, this discussion raises the question of whether or not the attack on the vapor lock problem should be directed towards reducing the vapor-forming tendency of the fuel or fuel system rather than increasing the vapor-handling capacity of the fuel system. It is encouraging that reduction of the vapor-forming tendency has received attention in the past. When the relationship between these two characteristics is fully understood and the results applied to the improvement of fuel pump designs, the full potential of the diaphragm-type fuel pump may be achieved.

Summary

Instrumentation using strain gages, with which it has been possible to follow dynamically the movement of the pump diaphragm, has been successfully designed and installed in an automotive fuel pump.

It was found that the pump stroke measured with this instrumentation did not correspond to the stroke estimated from the cavity pressure fluctuation. This was because an appreciable phase shift was found between the diaphragm movement and cavity pressure fluctuation when either a heated volatile fluid or air was pumped. The phase sift was such that the pressure minimum occurred before the diaphragm reached the bottom of the stroke. This phase shift has been explained qualitatively on the basis of pressure drop and restriction effects. The pressure drop across the inlet valve tends to make the minimum pressure occur at the maximum diaphragm velocity while restrictions in the inlet line tend to make the pump behave as a closed system, and increased restriction tends to move the minimum pressure to the bottom of the stroke.

With a low-volatility fluid, the volume displaced by the pump diaphragm was several times the net volume flow. This suggests that with no vapor in the system backflow may be large. With air also, to a lesser degree, the net volume flow was smaller than the volume displaced by the diaphragm indicating that some backflow may also occur with a vapor.

Using weight flow, volume displaced by the pump diaphragm, and liquid and vapor densities, it has been possible to calculate vapor to liquid (V/L) characteristics of the pump in a bench setup. Defining "vapor lock" as the condition when the fuel weight flow became less than the engine full throttle fuel requirement, calculated V/L's ranged from 22 up to 45 with the higher V/L at the lower speed. The minimum V/L at "vapor lock" were obtained at a cam speed of 1200 rpm and ranged from 22 to 26 for the four pump covers tested.

V/L at "vapor lock" in a vehicle on a wind tunnel chassis dynamometer were in excellent agreement with those obtained on the bench tests. It is interesting to note that these measured V/L ratios verify average vehicle V/L given in the literature.

The V/L at "vapor lock" on the bench setup were affected by restrictions in the outlet line. As the outlet line restrictions were decreased, the V/L at "vapor lock" were increased. From the data it also appears that the vapor-handling capacity may also be in-

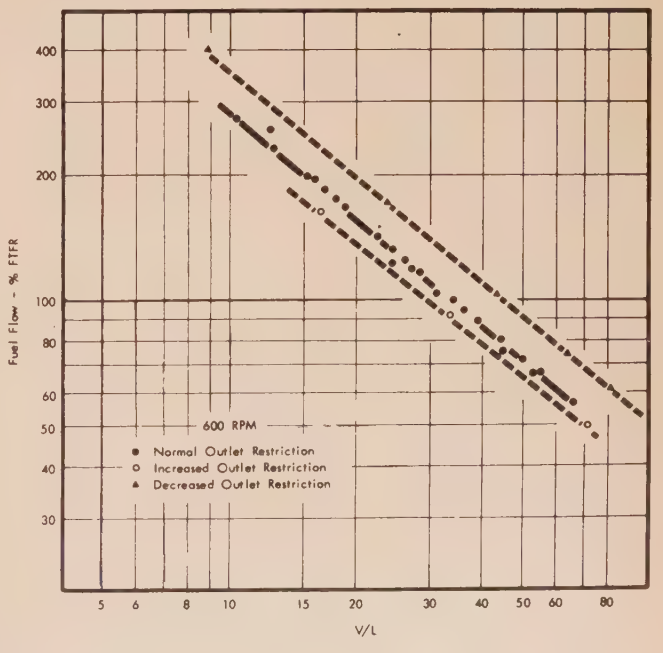


Fig. 21—Effect of outlet line restrictions on fuel flow and V/L relationship for pump cover A on bench setup

creased by operating the pump at engine rather than cam speed. However, the possible effect of pump speed on vapor-forming tendency makes it difficult to predict how the overall vapor-locking tendency of the system will be affected by pump speed.

Acknowledgment

The author is especially grateful to L. L. Withrow and J. D. Caplan of the Fuels and Lubricants Department for their help and encouragement in these studies and to J. O. Anderson and J. W. Melichar of the Special Problems Department for their assistance in the instrumentation problems.

DISCUSSION

Discusses Need for Further Study of Vapor Lock Problem

—Gordon W. Harry

AC Spark Plug Div., GMC

MR. RANDALL'S excellent paper gives us some additional information on a subject which is especially important. In reading the paper I was particularly interested in the study of the phase shift between the pressure traces registered in the diaphragm cavity as related to diaphragm movement. Some light on the subject (and a possible explanation) can be made if we consider the diaphragm design in the average pump. The piston or pumping area is essentially made up of two sections which function somewhat differently:

1. A rigid center section of pull-rod and protectors to which Mr. Randall fastened his reed.

2. A flexible annular ring or doughnut which bridges the space between the protectors and the cover and body sealing surfaces. This is the diaphragm proper.

Sufficient slack material must be provided in the annular ring to allow for full travel plus a factor of safety for variations due to mounting the pump on the engine. If we assume a position for the pump diaphragm assembly at the top of the stroke, the annular ring is formed downward from the previous pressure stroke upward. The center section, which includes the protectors, starts downward, actuated by the rocker arm and linkage connected to the engine. As it starts downward it forms a space above itself which tends to create a pressure below atmospheric, but simultaneously the surrounding diaphragm ring starts to roll upward so that, temporarily, while there is movement of the assembly, there is no pressure below atmospheric but rather a static condition registers on the pressure pickup. A somewhat similar condition, in reverse, takes place at the bottom of the stroke. This type of condition should show up as a lag of the pressure trace behind the diaphragm movement; however, in Mr. Randall's Figs. 7 and 8 this does not appear as "lag" but rather as an advance. This advance may be due to valve leakage during the time the diaphragm is practically stationary at the top or bottom of the stroke or to condensation of vapor. The no phase shift with the low-volatility liquid which would partially seal a leaky valve would add some weight to this assumption. Fig. 10, run with the pump inlet completely closed, also acts in a similar manner with little or no phase shift. Maximum diaphragm velocity with the present rocker-arm cams is approximately at the middle of the diaphragm travel. This should be the point of minimum pressure, but due to some lag the minimum pressure is at some point between the point of greatest velocity and the bottom of the stroke. This is apparent in Figs. 7 and 8. I believe some further study should be made on this relationship. It would have helped in trying to evaluate the curves if the atmospheric line had been shown.

In studying the effect of outlet restriction of V/L it would be well to review the geometry of a mechanical pump. A mechanical pump is essentially a combination of a pump and a regulating valve. It has a positive suction stroke which brings the liquid into the pump which, in turn, is discharged by a calibrated spring which actuates the discharge stroke. The amount of discharge is determined directly by the resistance to the upward travel to the diaphragm. This resistance or restriction is of two general types:

1. Internal resistance in the pump itself.

2. External due to lines, auxiliary strainers, or carburetor needle valves.

Internal resistance can be modified by larger valves or less restricted valves. External resistance can be reduced by larger lines, less restricted filters, and larger needle valves. Any reduction of either of these improves the vapor-handling capacity of the pump.

One of the points which tends to affect the vapor-handling characteristics is the size of the carburetor vents. They must be sufficiently large so that there is no appreciable back pressure to add to the outlet line restrictions.

Vapor Handling versus Vapor Forming Characteristics

Over the years our approach to the vapor lock problem has been towards improved vapor handling. This has been done in two ways:

1. By larger displacement pumps which would handle a given quantity of vapor safely.

2. By suggesting relocation of the pump on the engine and the lines and accessories so that there would be less tendency of vapor to form.

The vapor-forming tendency of the fuel system has been somewhat aggravated in recent years by increased horsepower of engines, the advent of power steering and air conditioning, the confined and restricted space under the hood, and the dual exhaust which has raised tank and intake line temperatures. This increase in vapor-forming characteristics has required us to use pumps with better vapor-handling capacities which generally means a larger displacement pump with less internal restriction. It is interesting to note that while the horsepower in some cases has been doubled on some engines in the last 10 years, the line sizes or line restrictions have remained practically the same. This is a point that should be considered on future designs.

Generally speaking, when the driver has trouble, he likes to place the blame on one unit rather than on a general condition, and when he has vapor lock his first thought is to blame the fuel pump. As Mr. Randall mentioned in his introduction the wide demand for air conditioning equipment and newer body styling has increased underhood temperatures and aggravated the vapor lock problem. In addition, the incorporation of the more volatile components of petroleum into gasoline in recent years has added to the problem. These points should be kept in mind when a new chassis is being laid out, and line locations and pump locations should be carefully considered in order that ultimate customer complaints would be minimized.

ORAL DISCUSSION

Reported by N. A. Hunstad

General Motors Corp.

C. C. Moore, Union Oil Co.: While volatile fractions are effective in raising the octane number of motor fuel, these fractions must be used in only very limited quantity due to vapor lock limitations with present-day automobile fuel systems. I would like to see a mechanical fix on the automobile fuel systems so that a larger percentage of these volatile fractions may be blended with present-day motor fuels.

SOME EFFECTS OF STROKE AND BORE ON DIESEL-ENGINE PERFORMANCE

TESTS on two series of diesel engines were run. The first group, consisting of four engines, had the stroke changed only, while the second group had the stroke/bore ratio changed and the displacement held constant.

Results of the tests indicate that the longer stroke engines had more power, higher torque, and lower fuel consumption.

Friction was high for the short-stroke engines at low speeds and for the longest stroke engine at high speeds. Theoretical analysis indicates that the optimum stroke/bore ratio for best performance may vary as the compression ratio and bore diameter are changed.

K. J. Fleck, Caterpillar Tractor Co.

This paper was presented at the SAE National Diesel Engine Meeting, Chicago, Nov. 2, 1956.

WHEN discussing the basic bore and stroke of an internal-combustion engine, many different factors, depending upon your special interests, may enter your mind. The final decision as to the size of engine manufactured will probably be based on a combination of reasons rather than on one characteristic only. For instance, cylinder arrangements influence the dimensions. In-line engines are sensitive to bore diameter because it affects overall length while a longer stroke might be had with very little difficulty. However, the longer stroke would eventually influence the length when bearing lengths and crank cheeks begin to dictate the cylinder center distance. V-type engines may present more of a width problem for installations, and the length of stroke may be a very critical dimension. Opposed-cylinder engines sometimes require short strokes to reduce width and frontal area. Radial aircraft engines also have a frontal area problem favoring short strokes. Weight efficiency expressed in pounds per brake horsepower or cubic inches may be very important and the dominating factor in an engine design. Some authorities [1] indicate that long strokes tend to increase the specific weight. This, of course, depends on the cylinder arrangement, the range of stroke/bore ratios that are being considered, and the specific engine performance obtained as a result of the change.

Engine performance and mechanical features also have their influence on the final design. This paper is primarily concerned with the effects that stroke and bore changes have on performance of the Caterpillar diesel engines.

An unusual amount of interest has been generated in "short-stroke" engines recently. Many people who normally have little concern for engine design have seen and heard advertisements comparing the length of strokes of automotive engines. Because of this renewed interest in the fundamental dimensions of engines, it seems fitting to reconsider some of the aspects of bore and stroke as they now appear to us.

Historically, the diesel has been a long-stroke engine. Obviously, that is no reason for progressive engineers to continue past practice, and it is not surprising to hear opinions expressed that a stroke/bore ratio approaching unity or less represents a modern trend of internal-combustion engine design. To support this, one need only to point to high-bhp piston-type aircraft engine design with ratios down to 1.03 along with provisions for the largest possible valves for a given displacement. Most of the cars qualifying for the Indianapolis races have engines with stroke/bore ratios of 1.07 or lower and have valves and ports similar to aircraft engines. Since every effort is made by the designers of these excellent engines to obtain good breathing, it is logical to conclude that the internal-combustion engines, fitted with a minimum of valve and port restrictions, would be the best design for obtaining maximum power. I be-

1 "High-Speed Combustion Engines," by P. M. Heldt. Pub. by P. M. Heldt, Nyack, N. Y., 1951.

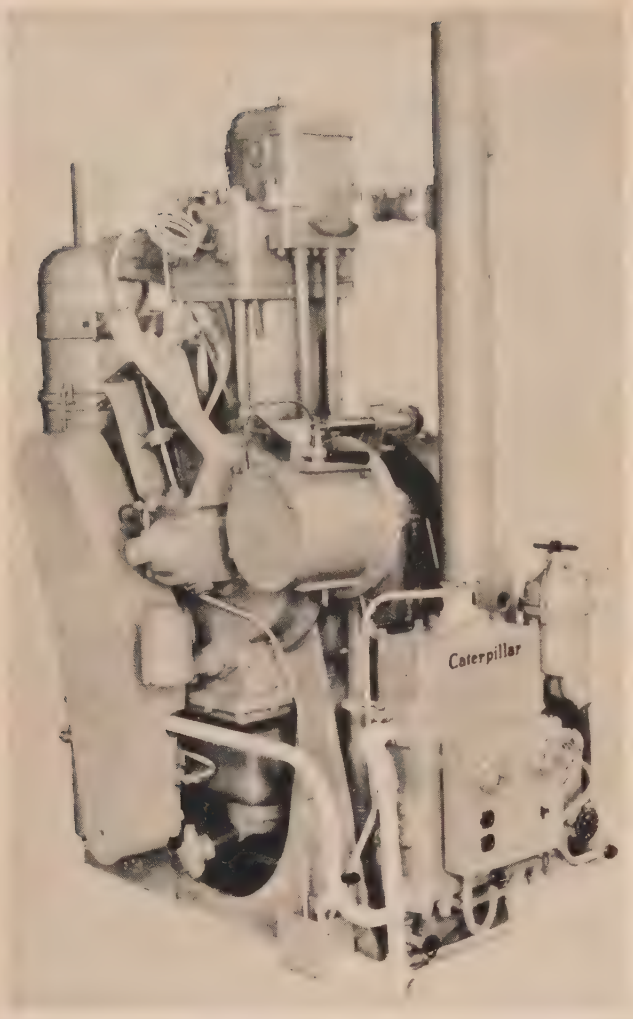


Fig. 1—One-cylinder engine used for laboratory testing

lieve it is generally agreed that the problem of obtaining adequate breathing at a given rpm is simplified as the stroke/bore ratio is reduced.

While the spark-ignition and compression-ignition engines do have very much in common, with high compression ratios rapidly being added to the list, one important difference might be mentioned. In the diesel engine, the combustible mixture is not formed until the piston approaches close to top center. The fuel to be mixed with the air is only brought into contact with the air when the combustion chamber is close to its minimum volume. In the diesel combustion system, we hope to retain a high temperature of the compressed air and to obtain almost immediate auto-ignition of the fuel so that it is burned as it is injected, thus avoiding any uncontrolled detonation. The time factor in achieving effective combustible mixtures is so short and the process so complicated because of the superimposed influences of the ensuing combustion, that the phenomena are difficult to analyze. However, these complexities can influence the design toward a stroke/bore ratio other than that considered the most ideal, judged only from the standpoint of valve design and air-charging efficiencies. Indications from our past engine experience have shown the larger stroke/bore ratio engines to give the best performance. This was observed by comparing the different size engines we manufacture or have tested. The comparisons were not always valid, because many changes other than bore and stroke would often be included. One author [2] of a diesel-engine design textbook states that, "For normal engines, the longer stroke (length/bore = 1.48 to 1.57) is to be preferred unless external factors exert an overriding influence." Another author [3] states that, "There is no important advantage in any stroke/bore ratio as such. It is usually chosen on the basis of other considerations." This illustrates the differences of opinion found in the limited amount of information published on this subject.

Because of these and many other factors involved, we decided to ask the engine what stroke and stroke/bore ratio it liked best.

A total of seven different engines were designed and tested, with as much similarity as possible, so that the changes in cylinder dimensions could be pointed to as the major difference. The parts which had to be changed will be mentioned for consideration. A 1-cyl laboratory test engine similar to the one shown in Fig. 1 was used for all the different investigations. The accessories such as the water, oil, and fuel pumps, as well as the governor and gears, were the same for all test engines. All of the engines were of the precombustion-chamber type as illustrated in Fig. 2. When it was necessary to change the volume of the main chamber or prechamber, they were all proportioned according to our standard design practice. Identical test conditions were held throughout. Intake air pressure was controlled to 29.6-in. Hg. Intake air temperature was maintained at 100 F. Oil temperature was controlled at 165 F and the cooling water at 180 F.

Two groups of single-cylinder engines were made to study two different conditions.

The first group represented a situation where the cylinder block, head, and bore diameter were fixed, and the stroke was changed. These engine sizes are

[2] "Diesel-Engine Design," by T. D. Walshaw. Pub. by Neunes, London, 1953.
[3] "Diesel Engines," by J. W. Anderson. Pub. by McGraw-Hill Book Co., New York, 1949.

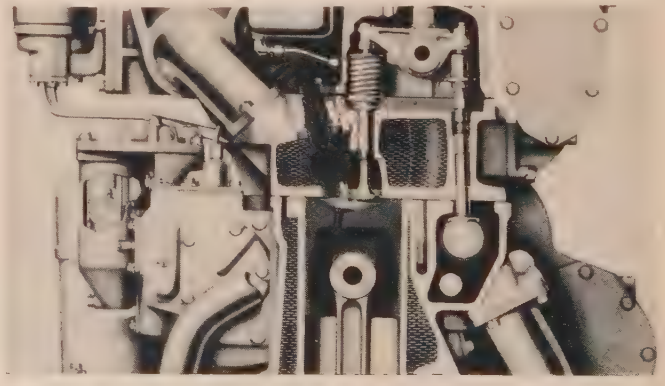


Fig. 2—Precombustion-chamber engine used in testing

shown in Table 1. The first column shows the bore and stroke of this group of engines. Notice that the bore size was held constant. The second column shows the change in displacement as a result of the stroke changes. The stroke/bore ratio for each is shown in the third column. The last column shows the ratio of combustion-chamber wall surface to volume of the combustion chamber at top center. A high surface/-volume ratio is an indication that more heat may be lost from the compressed gases, and the combustion-chamber proportions may be less desirable, which will be discussed later. The first and fourth engines shown in the table had no relation to a production engine. They were made only to extend the stroke change and thereby to establish a trend. The second engine, 5.12 x 6 is a single-cylinder version of the Caterpillar D326 engine. The third engine, 5.12 x 6.5 is a single-cylinder version of the D326 Series F engine.

Fig. 3 shows the profiles of the combustion chambers used in this series of engines. The same cylinder head, ports, valves, and camshaft were used. This gave the shortest stroke engine the advantage of better breathing for a given rpm as compared to the longest stroke engine, since the displacement was only 71% of the longest stroke engine. The combustion chambers had to be varied to adjust for the variable displacement. The profiles of the shortest and longest stroke engines are shown. The other two combustion chambers fall in between these two extremes. The fuel system was similar throughout. The rate of injection of fuel delivered to the cylinder was changed to match the displacement.

The second group of engines had the stroke/bore ratio as the main variable. The dimensions of the four engines are shown in Table 2. You will recognize the first engine, the 5.12 x 7.41, as the longest stroke engine from the first group. The bore and stroke were both changed for the three new engines, as shown in the first column, so that the displacement of the cylinder remained nearly constant. As a result, we had a series of engines in the same size range, with stroke/bore ratios as shown of 1.45, 1.30, 1.04, and 0.85.

In each test engine of this second group, the cylinder heads were similar to the head used in the first group, having two valves which moved parallel

Fig. 3—Profile of combustion chambers of constant bore series

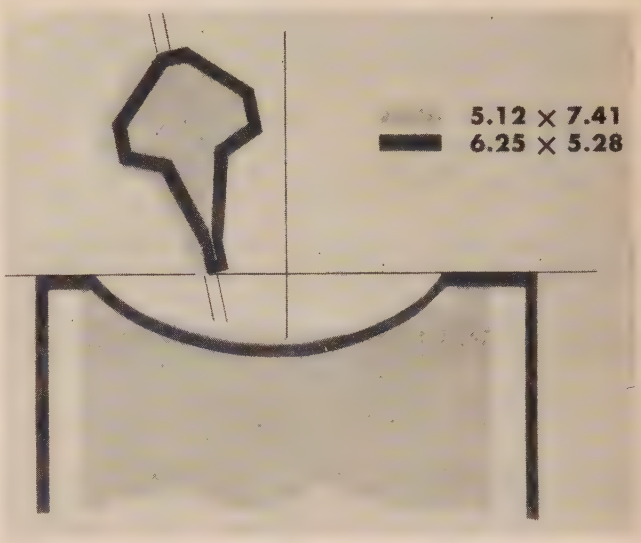


Fig. 4—Profile of combustion chambers of constant displacement series

to the cylinder centerline. The valve heads were contained within the inside diameter of the cylinder bore. The cylinder heads were also designed to allow the use of larger inlet and exhaust valves and ports as the bore was increased, which is considered most advantageous to the large-bore, short-stroke engine. All of the ports were made according to our present design practice for intake and exhaust ports, with the exception of the 5.75 x 6, which used the head from our single-cylinder 5.75 x 8 test engine, with ports originally intended for speeds up to 900 rpm. It became apparent that the restriction of these ports was the limiting factor when the performance of this engine dropped out of line above 1600 rpm.

Where in the first group of engines it became necessary to change the precombustion-chamber volume to maintain the optimum relation to the displacement and to the main combustion-chamber volume at the

Table 1—Single Cylinder Test Engines
(Constant Bore and Variable Stroke)

Bore and Stroke	Displacement	Stroke/Bore Ratio	Surface/Volume Ratio
5.12 x 5.28	109	1.03	9
5.12 x 6.00	124	1.17	8
5.12 x 6.50	134	1.27	7.1
5.12 x 7.41	153	1.45	5.8

Table 2—Single Cylinder Test Engines
(Constant Displacement, Variable Bore and Stroke)

Bore and Stroke	Displacement	Stroke/Bore Ratio
5.12 x 7.41	153	1.45
5.38 x 7.00	159	1.30
5.75 x 6.00	156	1.04
6.25 x 5.28	161	0.85

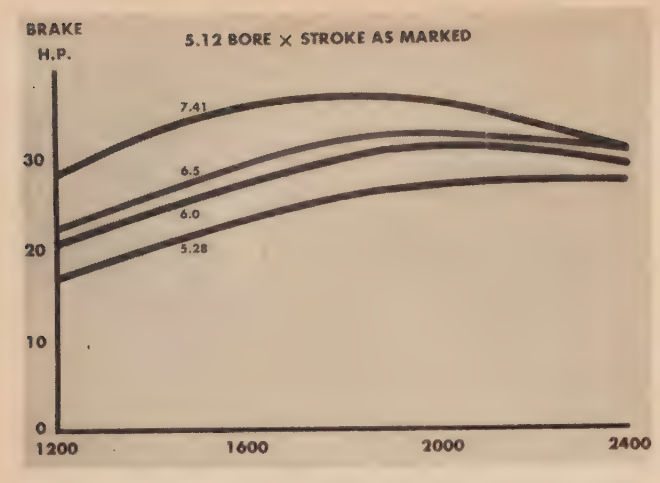


Fig. 5—Actual maximum bhp for each engine of constant bore series

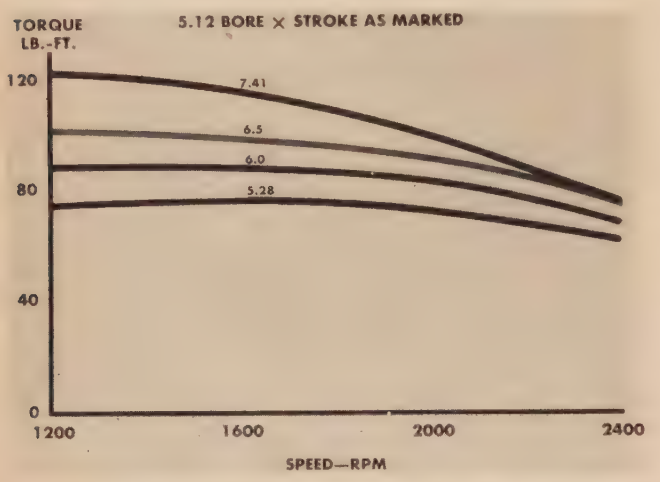


Fig. 6—Maximum torque curves for each engine of constant bore series

Table 3—Comparison of Long- and Short-Stroke Test Engines

	Long	Short
Bore	5.12	5.12
Stroke	7.41	5.28
Displacement	153	109
Bmep	100	100
Piston Speed	2225	2225
Rpm	1800	2530
Bhp	34.8	34.8

top of the stroke, the same precombustion chamber was used in all tests in this second group of engines. It has been well establish, through experimental work with the Caterpillar combustion system, that the volume of the precombustion chamber must be held in reasonably close relationship to the total clearance volume, and is not dependent upon bore or displacement of the cylinder. However, the main chamber crater was adjusted according to Caterpillar design constants to conform best to the diameter of each piston. Fig. 4 shows the two extreme stroke/-bore ratio cylinder profiles. Although both craters or piston bowls are proportioned to give optimum performance for each engine, it becomes difficult to design the combustion chamber for the 6.25 x 5.28 engine. The problem may be considered analogous to

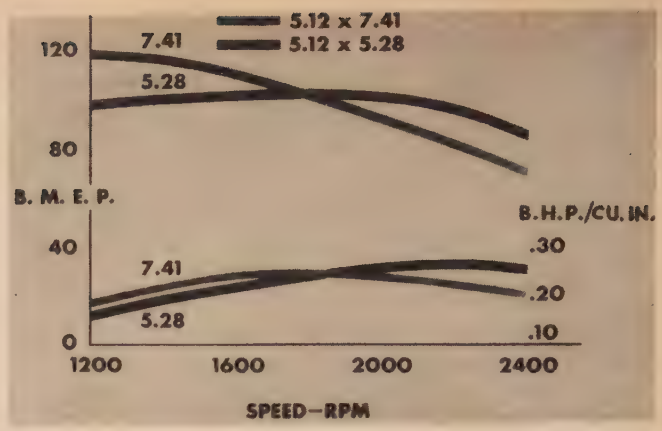


Fig. 7—Bmep and bhp per cu in. versus speed for two extremes of stroke

using a platter as a mixing bowl. A certain minimum piston-to-head clearance is essential, and any air left in this space probably plays a very small part in combustion. However, it can be seen that the "lost volume" is much greater on the large-bore engine. To avoid this by spreading the crater wider across the piston results in a flatter, pancake shape, in which mixing of fuel and air becomes more difficult. It is also obvious that the surface/volume ratio with the piston at top center is high for the large-bore, short-stroke engine, regardless of the crater shape. This volume of compressed air for diesel engines is small compared to the current spark-ignition engines, because the compression ratio is higher.

The fuel injection system, including the camshaft, pump, fuel line, and nozzle were the same. Timing tests were run, and the optimum timing was selected for each test engine, based on maximum power and minimum fuel consumption.

The test results of the engines having the same diameter cylinder, will be discussed first. Before looking at the data we might speculate either, that the long-stroke engine should develop the highest actual bhp, because it has the largest displacement, or that the shortest stroke engine should develop the highest bhp, because it would be able to run at a much faster rpm when developing maximum power. The actual maximum bhp for each engine is shown in Fig. 5. Power increased with each increase in stroke at all speeds throughout the 1200–2400-rpm speed range, varying from a value of 28.2 bhp for the 5.28-in. stroke engine, to 36.5 bhp for the 7.41-in. stroke engine. Notice that at 2400 rpm, the 6.5-in. stroke engine developed nearly as much power as the 7.41-in. stroke engine. The shortest stroke engine did develop its maximum power at a higher speed but was not able to equal the maximum of any of the longer stroke engines.

Theoretically, the bhp would be the same regardless of stroke for a fixed bore size, if we assume that the bmep would be constant for a given piston speed. Since,

$$\text{mean piston speed} = \text{rpm} \times 2 \times \text{stroke}$$

we can substitute this into the bhp equation to read,

bhp = bmep × area × piston speed × constant

A comparison of the long- and short-stroke test engines, according to these assumption, would be as shown in Table 3.

If this were true, then the short-stroke engine could be reduced in size relative to the long-stroke engine, and a weight reduction could be made. The theoretical analysis, therefore, makes the short-stroke engine look attractive.

The actual data shown, for an engine of our type, and for the conditions outlined for the first group, indicate that the bmep was higher at constant piston speed for the longer stroke engines. It could also be said that the optimum piston speed for maximum bhp was higher for the longer stroke.

Fig. 6 shows the maximum torque curves. The outstanding difference shown here, is the excellent performance of the longer strokes as the engine is "lugged down" from high speeds. The "flat" performance as indicated by the 5.28-in. curve would have to be reduced at the high-speed end by governing devices, to obtain a torque rise best suited to the load demand of our machines. Not only did the longest stroke engine develop more power at 2000 rpm than the shortest stroke engine, but it also had 25% torque rise on lug as compared to 5%. This may not seem too important a point to those concerned with constant-speed diesel engines or for loads which fall off rapidly with a drop in speed such as marine engine propeller loads. In most of our engine applications it is essential to be able to provide a generous amount of torque increase as load demand increases.

So long as conditions exist which make "torque-rise" important, we will continue to look for good engine performance at both ends of the useful speed range. It is, therefore, a desirable situation to find adequate power at rated speed backed up by a considerable increase in torque or bmep at lower speeds. The only alternative, if more "torque-rise" is wanted than the engine naturally has, is to derate the high-speed end and add displacement if necessary to obtain the rated speed power. The drop in performance of the 7.41-in. stroke engine at high speeds indicates that if we would continue to increase the stroke indefinitely, a point would be reached where the displacement would be too great for good breathing at this speed range. Other design considerations eliminate the need to be concerned about this.

Bmep plotted versus speed is shown in Fig. 7. The two extremes of stroke are shown only, the other two falling in between these two curves, and crossing at 1800–2000 rpm. Bmep and bhp per cu in. were highest for the 7.41-in. stroke engine at 1200 rpm and for the 5.28-in. stroke at 2400 rpm. The actual values of maximum bhp per cu in. for the four engines were 0.257, 0.250, 0.245, and 0.238 going from the shortest to the longest stroke, while maximum bmep in the same order was 105 at 1600, 109 at 1200, 112 at 1200, and 120 at 1200 rpm. The increase in bmep and displacement as the stroke was increased more than offset the slight decrease in bhp per cu in. shown. The net result was the increased power with each increase in stroke which was shown in Figs. 5 and 6.

The longer stroke, larger displacement engines were more economical to operate as illustrated in Fig. 8. These curves show the lowest fuel consumption that was obtained for a required power output. The curves are made up of points from various engine speeds from 1200 to 2400 rpm. Notice that the fuel consumption dropped with each increase in stroke ranging from 0.47 down to 0.426 lbs per bhp-hr from the short- to the long-stroke engine.

One way to use this chart, would be to compare the bhp which could be obtained on a fixed amount of fuel. On 12 lb of fuel, the shortest stroke engine will deliver 22.5 bhp for 1 hr, and the longest stroke engine, 27.5 bhp, or 21% more power for your fuel dollar. The fuel consumption was high on all the single-cylinder engines at 2400 rpm. High friction at this speed was partially responsible for this.

Friction is shown in Fig. 9 by the bar graphs, with

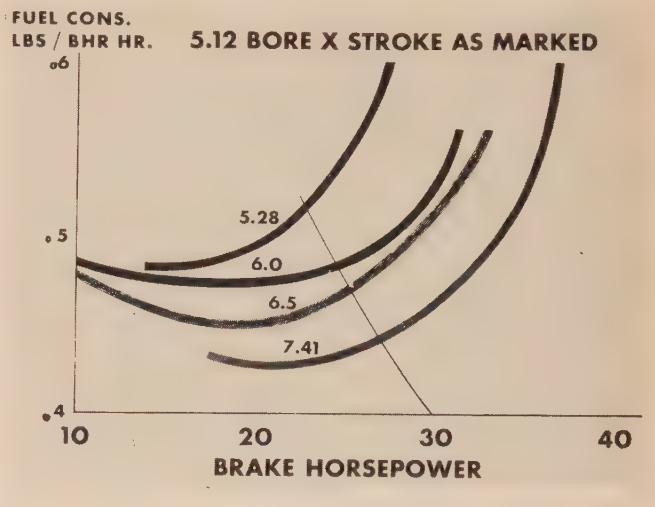


Fig. 8—Comparison of fuel consumption for fixed-bore, variable-stroke engines

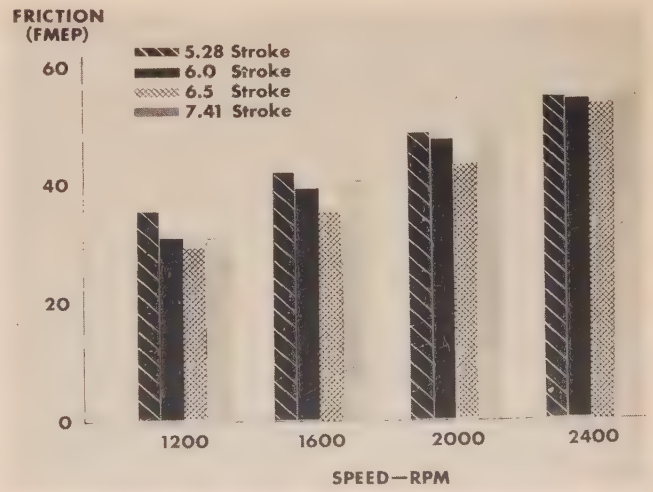


Fig. 9—Friction comparisons for engines at different speeds

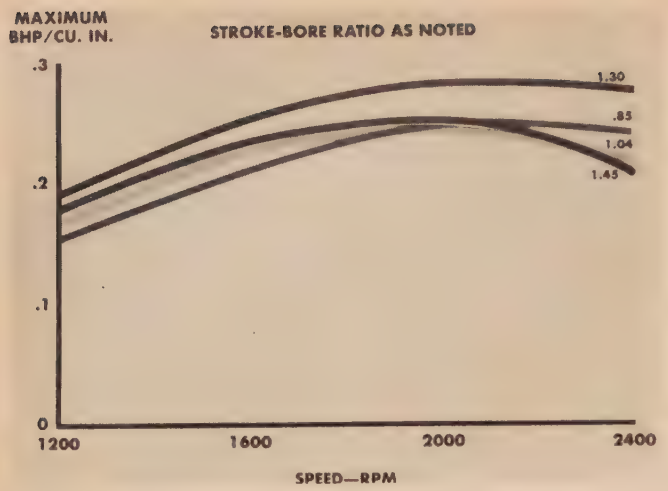


Fig. 10—Maximum bhp per cu in. developed throughout speed range

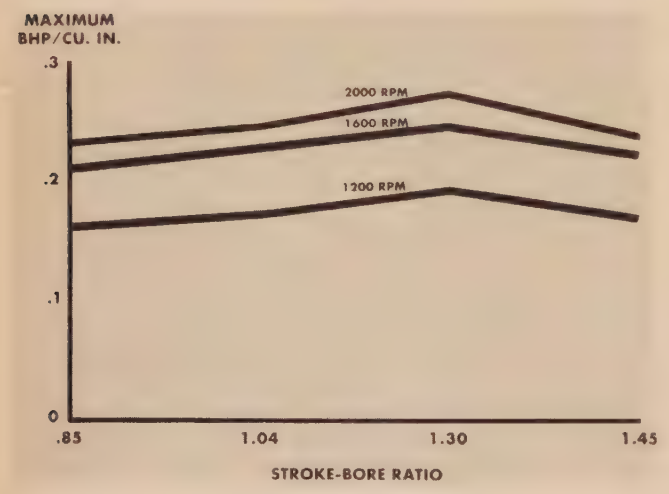


Fig. 11—Comparison of maximum power versus stroke/bore ratio for fixed speeds

the results of all four engines grouped at each speed. While the actual values of fuel consumption and friction may be high because this is single-cylinder data, the comparison between engines should be valid. The left-hand or first bar of each group is for the shortest stroke, and each one to the right is for the next longer stroke. The shortest stroke engine had the highest fmep or "dead load" per cubic inch at low speeds, while the longest stroke engine was slightly higher at top speeds. The 6.5-in. stroke engine showed the least amount of friction at all speeds. Fmep and heat rejection to the cooling water, being lowest on the 6.5-in. stroke engine, indicate less load on the cooling system, lower fuel consumption, and longer life of critical parts. The highest friction on the 5.28-in. engine correlated with high heat rejection to the cooling water. This might be expected for an engine having a relatively high surface/volume ratio at top center where the gases are turbulent and hottest.

To summarize this first series of tests we can conclude that:

1. Actual bhp and torque increased with each increase in stroke.

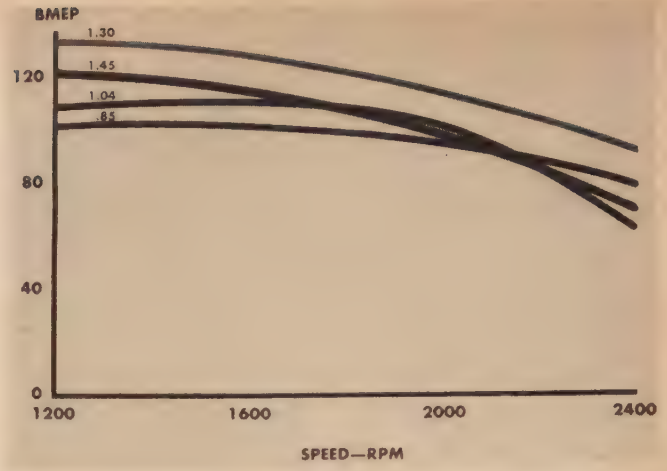


Fig. 12—Maximum bmep for fixed stroke/bore ratios

2. Bmep (and bhp per cu in.) were highest for the long-stroke engine up to about 1800 rpm, and for the shortest stroke at 2400 rpm.

3. More power at a lower fuel consumption rate could be obtained with a longer stroke.

4. Friction was the lowest on the 6.5-in. stroke, 1.27 stroke/bore ratio engine.

5. Torque increase characteristics improved with each increase in stroke.

In the second series of engines tested, the stroke and the bore were allowed to vary while the displacement remained nearly the same. One author [1] suggested this as a fair way to compare engines of different stroke/bore ratios. There appeared to be an optimum stroke/bore ratio for a Caterpillar diesel engine of this size.

Fig. 10 shows the maximum bhp per cu in. obtained from the four engines throughout the speed range. All of these engines developed peak bhp near 2000 rpm. Note that in the normal operating range of 1200 to 2000 rpm, the 0.85 and 1.04 ratio engines were lowest, the 1.45 ratio engine, which was the long-stroke engine from the first series, was somewhat higher, but the 1.30 ratio engine appeared to be the best of this group. The 0.85 and 1.30 ratio curves parallel each other throughout the speed range. The rapid drop of the 1.04 ratio curve above 2000 rpm is believed to be due to the port design mentioned previously, and the drop of the 1.45 ratio curve appears to be a normal consequence of the small bore and long stroke. The highest bhp per cu in. was 0.273 in this series versus 0.257 in the first series of tests. In order to study this relationship of specific output to stroke/bore ratio, the same data is replotted as shown in Fig. 11. Here the lowest curve shows the maximum output for all engines at 1200 rpm, the middle curve at 1600 rpm, and the top curve at 2000 rpm. The 2400 rpm curve was omitted because the values were less in each case than the 2000 rpm points; in other words, it was past the peak bhp points, and the 1.04 stroke/bore ratio engine, as previously mentioned, was unduly handicapped by the choking

effect of the intake and exhaust ports at 2400 rpm. Some evidence of this is shown by the sagging of this point on the 2000 rpm curve. The general indication is that the 1.30 ratio is near the optimum point for best performance, the lower ratios probably suffering from heat losses and unfavorable mixing of the fuel and the air, while the 1.45 ratio was limited by friction and breathing.

Bmep versus rpm is shown in Fig. 12. Here again, the 1.30 ratio looks best, and the 1.45 ratio is next for applications where torque increase characteristics are of importance. The 0.85 ratio showed some promise at 2400 rpm only.

The measurement of exhaust-smoke density is one way in which we can rate diesel engines. A reading of 0.3 smoke density on our smokemeter, represents a point just beyond visible smoke, sometimes referred to as "light" or "just visible." A well-developed engine of the precombustion-chamber type will deliver approximately 95% of maximum power before reaching this arbitrary smoke point. We have found this to be a reasonable point to consider as a rated maximum, allowing the engine to run with much lower temperatures, less carbon formation, and much lower fuel consumption, than it would if allowed to run at absolute maximum. Fig. 13 shows graphically, the relation of these 0.3 smoke points to absolute maximums previously mentioned. The 1.30 ratio fell out of line relative to its maximum at 1200 rpm, indicating need for further development to reduce smoke at this speed. The small difference between the smoke-rated maximum and absolute maximum at 2000 and 2400 rpm, indicates that the combustion was good and that power was limited by other factors.

The minimum fuel consumption is shown in Fig. 14. The 1.30 ratio gave the lowest fuel rate of all engines tested.

Comparative friction of these engines is shown by the bar graph in Fig. 15. The 1.45 ratio showed the highest friction throughout. With the exception of 1200 rpm, there was an indication that friction increased with each increase in stroke/bore ratio. Friction was measured by the "motoring method."

Another method of studying friction is to compare the amount of fuel required to run the engine without external load, or idling. The measurement is therefore made while the engine is operated and might more nearly measure the friction which prevails when the engine is in normal use. When idling, the only load the engine has is overcoming its own friction, and the fuel rate should be very sensitive to small changes in friction. Unlike gasoline engines, the diesel is operating with excess air, and combustion of the fuel at idle should be very good on all engines. The idle fuel rates are shown in Fig. 16. In comparing the engines by this method, the 1.30 ratio engine was the lowest at 1200, 1600, and 2000 rpm, and the 0.85 ratio engine raised slightly when compared to friction data previously shown. This change may be due in part, to the differences in combustion. However, the fuel-rate data seem to follow the general pattern of measured friction, but varying with speed at a different rate than the motored friction indicated.

To summarize the second series of tests we can conclude that:

1. When the displacement was held nearly constant (range of 153–161 cu in.), the stroke/bore

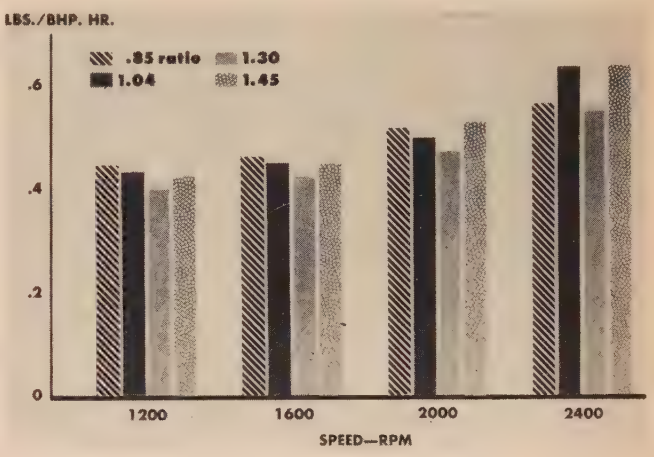


Fig. 14—Minimum fuel consumption

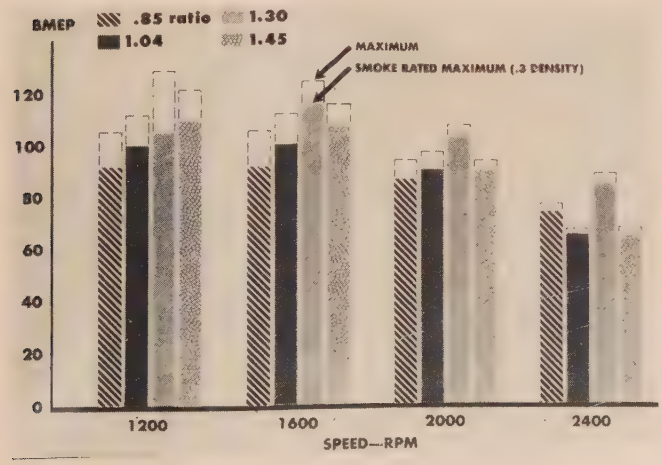


Fig. 13—Power rating by smoke density

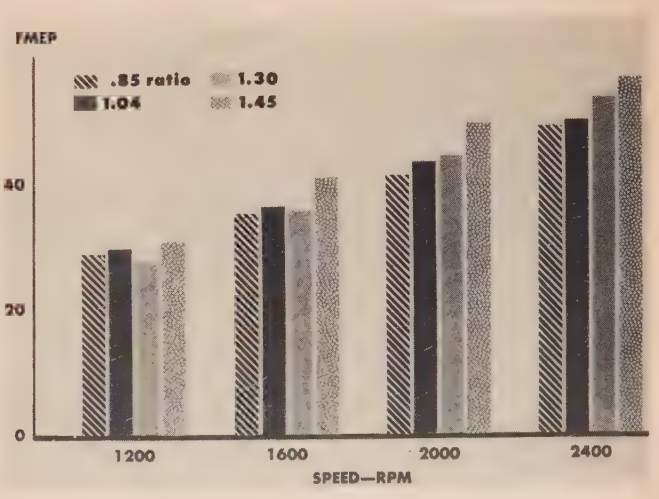


Fig. 15—Friction comparisons

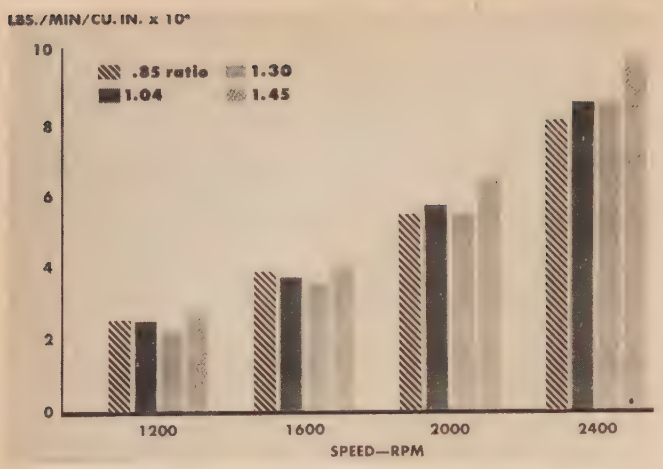


Fig. 16—Comparison of idle fuel rates

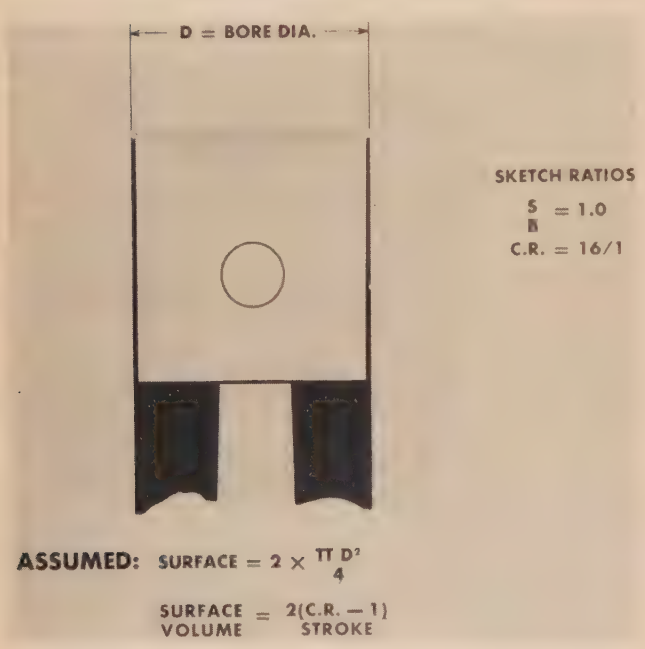


Fig. 17—Theoretical combustion chamber, case 1

ratio of 1.30 gave the best performance of those tested.

2. The 1.45 stroke/bore ratio engine, while performing well, showed excessive friction and a higher rate of power loss at the high speeds.

3. The shortest stroke engine was both low in friction as well as power, indicating that its overall thermal efficiency was relatively low.

The results of these two series of tests indicated that there were changes in engine performance that might be attributed, at least in part, to the change made in the cylinder dimensions. The foremost purpose of this test was to establish a trend. It is quite possible that the performance of any one of these engines could be improved with further development. However, it is our belief that we have successfully resisted the temptation to "play any favorites" unless it was for the underdog. Some of the engines which were low in power were modified and rerun before the data was accepted. The best data obtained were then used. However, a rather consistent pattern resulted in the first series of tests when combining the data from two more or less undeveloped engines with two engines similar to production engines.

After collecting data such as this on experimental tests, it is frequently desirable to ask, "Why these results?" Theoretical analysis "comes to life" if you have some test data to relate to it. One of the variables previously mentioned is the surface/volume ratio. It influences not only heat rejection but also the design of the combustion chamber. Therefore, it might be worth some theoretical consideration. We would like to consider the situation with the piston at top center during the compression-expansion strokes only for the following reasons. It is at or near this point that the compression and combustion temperatures are the hottest and the turbulence of the gases is greatest. Since the heat will transfer at the fastest rate under these conditions, any changes which might have an effect on this rate of heat loss could easily have an effect on ignition, combustion, and friction. As previously mentioned, the process of making a combustible mixture and burning it occurs near this top-center point. A simple expression for this surface/volume ratio can be made if we assume that the surface mainly in contact with the compressed air is equal to the area of two circles with diameters equal to the diameter of the cylinder bore as illustrated in Fig. 17.

$$\frac{\text{Surface}}{\text{Volume}} = \frac{2 \times \frac{\pi D^2}{4}}{\frac{\pi D^2}{4} \times \text{stroke} \times \frac{1}{(\text{compression ratio} - 1)}}$$

or,

$$\frac{\text{Surface}}{\text{Volume}} = \frac{2 \times (\text{compression ratio} - 1)}{\text{stroke}}$$

In this perhaps overly simplified equation, the bore diameter cancels out, bore size does not matter, and surface/volume ratio varies directly with compression ratio and inversely with stroke length. It also says that, for a given surface/volume ratio, if the compression ratio is increased, the stroke should be increased.

Another analysis can be made, giving some consideration to bore, by assuming a simple cylinder with a "flat-top" piston exposing the compressed gases to a narrow band of cylinder-wall surface, as illustracted in Fig. 18. The formula then becomes:

$$\frac{\text{Surface}}{\text{Volume}} = \frac{2 \times (\text{compression ratio} - 1)}{\text{stroke}} + \frac{4}{\text{bore}}$$

This is similar to the last formula except for the addition of the factor $\frac{4}{\text{bore}}$, throwing a penalty on a small-bore engine. From this formula of surface/-volume for a theoretical engine, curves can be plotted, showing its relation to stroke/bore ratio, to compression ratio, and to bore diameter.

Fig. 19 shows curves for three series of engines

with the bore held at 4, 5, and 6 in., all assumed to have a compression ratio of 16/1. Notice that for a constant surface/volume ratio, the stroke/bore ratio can be reduced as the piston diameter is increased. Points 1, 2, 3, and 4 are for theoretical engines with the same bore and stroke as the first series of test engines, and points 4, 5, 6, and 7 are for the second series of test engines. Notice that the constant displacement for the second series of engines forms a flatter curve than the constant bore group. This may explain why the second group appeared to have an optimum stroke/bore ratio of 1.30 while the first group indicated that an even greater ratio might be desirable. From actual data, it also appeared that the 0.85 ratio, large-displacement engine operated as well, or better than, the 1.03 ratio small-displacement engine. This is indicated by comparing the position of engines 1 and 7 on the theoretical curves.

Fig. 20 shows two series of theoretical 4-in. bore engines, one with 16/1 and one with 8/1 compression ratio. These curves illustrate the point previously made that, for a constant surface/volume ratio, when the compression ratio is reduced, the stroke/bore ratio can be reduced.

While all the test data reported here were run under naturally aspirated, 100 F air temperature, sea-level conditions, a large number of the 5⅛-in. bore, 6½-in. stroke Series F engines are supercharged by an exhaust driven turbocharger. Some data were run on the first series of engines under simulated supercharged conditions, indicating the same trend that was observed on the unsupercharged engine tests. It is normal to expect the engine which operates most efficiently unsupercharged to also be a good blown engine. The rated output of engines is often limited by the "thermal loading" on critical parts, especially those which are highly supercharged. It is therefore more important than ever to pay considerable attention to fuel consumption on the engines we intend to supercharge. Supercharging is usually associated with high-speed, maximum-bhp engines. However, another important advantage of our new turbocharged engines is the opportunity to obtain high outputs at lower speeds with less friction, lower fuel consumption, and lower thermal loads. The supercharged engines often operate at lower fuel rates up to considerably higher loads than can be obtained on the same engine unsupercharged. Similarly, when the customer has some freedom of choice between sizes of unsupercharged engines, it is often to his advantage to select a large displacement per cylinder, optimum stroke, engine with high output, and to operate at less than maximum speeds, in order to take advantage of lower friction, lower fuel consumption, and longer life.

In conclusion, I would like to emphasize a few points concerning this subject. As mentioned before, the influence that stroke and bore changes have on performance is only one of the factors which must be considered in the final design of an engine. The driven equipment requirements of torque and speeds influence or even dictate the design. The engine must be considered as a part of an entire package with consideration given to the machine we are driving and the types of transmissions and gear reductions involved, and then it must be tailored to meet these requirements. For these and other design considerations, we may not build our future engines to conform

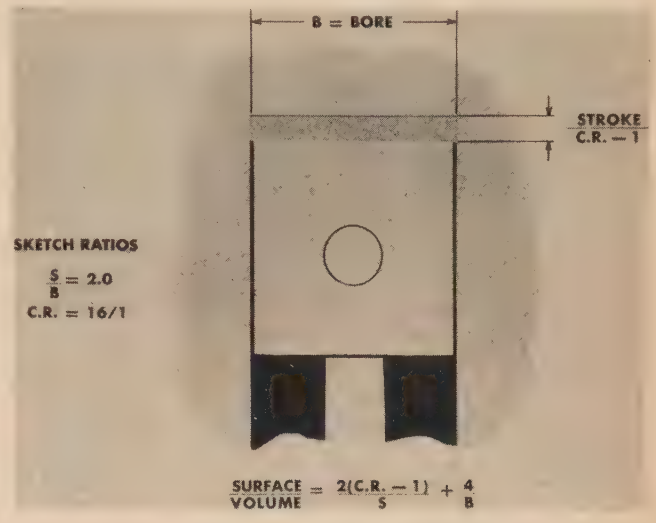


Fig. 18—Theoretical combustion chamber, case 2

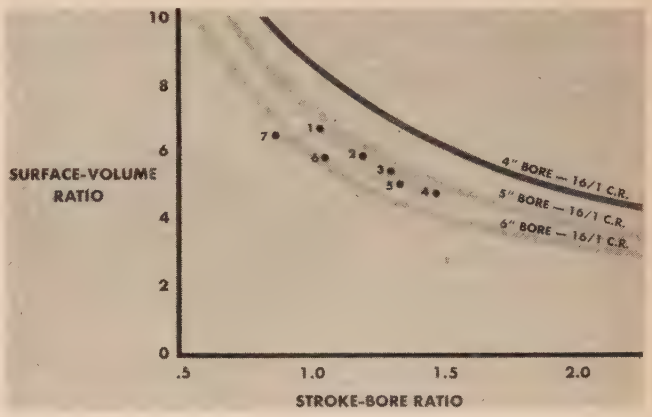


Fig. 19—Effect of bore on theoretical surface/volume ratio

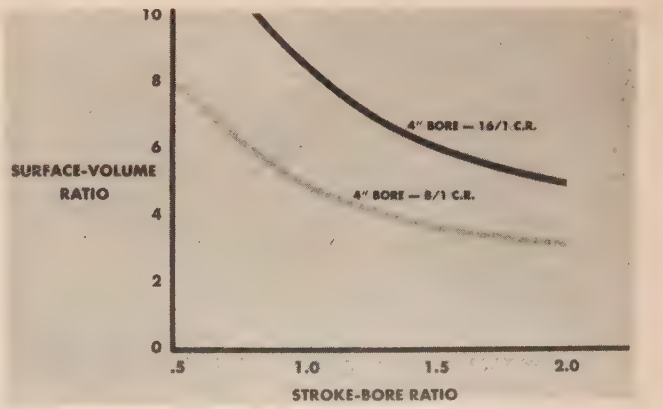


Fig. 20—Effect of compression ratio on theoretical surface/volume ratio

Fig. 21—Applications of Series F engines

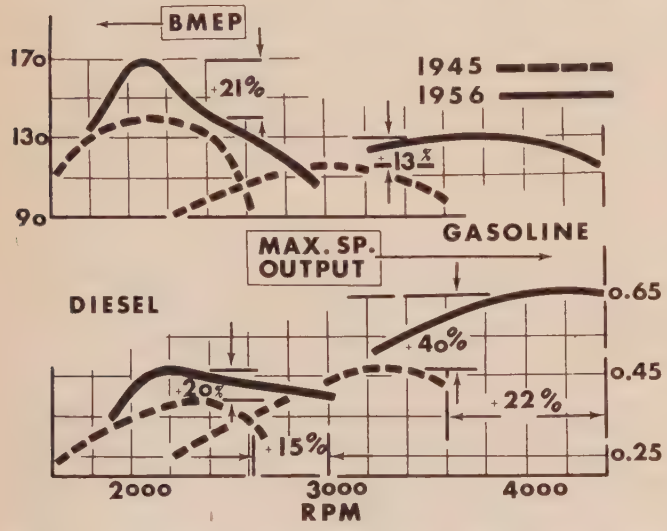


Fig. A—Comparison of automotive engine design trends, 1945 and 1956

exactly to the results of these tests. We would like to reemphasize that these results may fit our type of diesel engine only, and optimum ratios may be different for other types of diesel engines. There must be considerable data in the diesel-engine industry on this subject which would be of interest to this group.

A general conclusion can be made from the test data presented, that the D326 and D337, Series F, 5⅛-in. bore by 6½-in. stroke, 6-cyl engines give better performance than the 6-in. stroke engines. The size and weight of the engines remained nearly the same, and could be fitted into installations where the D326 and D337 had been used, making it more efficient in every respect. Some of the applications of these engines in Caterpillar machines are shown in Fig. 21. The Series F engines have been in production for more than a year, and reports of excellent performance and long service life from the field substantiate our conclusions that the longer stroke was the right direction to go in, for the DW20 and DW21 turbocharged earthmoving machines, as well as the naturally aspirated and turbocharged industrial engines.

Acknowledgment

I would like to acknowledge and thank the many people with whom I am associated for help and suggestions on this project, and particularly D. W. Knopf and W. C. Loveridge who conducted the engine testing.

DISCUSSION

Discusses Limitations On Diesel-Engine Performance

—E. R. Klinge and H. H. Haas

Continental Aviation and Engineering Corp.

THE author and his company have made a definite contribution to a subject which is very complex but at the same time one of the most important matters that faces engine designers. The author has briefly mentioned some of the factors which influence stroke and bore selection other than the ones he was concerned with in his work. Very clearly, the engine configuration itself exerts a very strong influence on these decisions. Analysis of minimum bulk and minimum weight is certainly a part of this story which the author has chosen to omit. It is not possible in this discussion to add such an analysis, but merely to corroborate the *performance* discussion already presented by the author.

It appears that spark-ignition engines and compression-ignition engines are developing in different directions as can be seen by analysis of catalog data. Fig. A shows a compilation of data since the war years comparing 1945 to 1956 engines. It shows the maximum bmep and horsepower per cubic inch versus speed of automotive (that is, truck and tractor) diesel and gasoline engines for 1945 and 1956. It is noted that most of the increase in gasoline-engine output has been achieved through speed increases of 22% whereas the bmep has gone up only 13%, resulting in a net increase of 33%. The diesel engines, on the other hand, have a much different story. Here the big increase was 21% in bmep, accomplished to a great extent with turbosuperchargers, and a 15% increase in speed for a net increase of 20%. It should be noted also that the bmep falls off rapidly with speed.

This trend in diesel-engine design (that is, reduction of bmep with increased speed) is further substantiated by Fig. B. Here is plotted the maximum-rated bmep versus maximum-rated speed of a number of actual engines. They readily fall into three groups, namely, those that are naturally-aspirated, supercharged production engines, and experimental engines. The range covers marine, locomotive, and stationary engines at one extreme and automotive engines at the other end. These curves do not necessarily say that this is the ceiling on diesel-engine design; rather, they say this is as far as anyone has gone today and that a great deal of effort must be expended before these lines can be raised. It is interesting to note that one manufacturer has already made a breakthrough and has a production engine developing 255 bmep at 1800 rpm.

At first glance, Fig. B appears to be carrying a torch for the long-stroke, slow-speed engine. Careful analysis proves otherwise. The slope of the supercharged engine curve is 3.5 bmep per 100 rpm. Using this load factor and varying the rpm with the stroke/bore ratio so as to maintain a constant piston speed of 2200 fpm, it can be shown that for any cylinder displacement there is an optimum stroke/bore ratio. Two different cylinder sizes have been calculated, 50- and 150-cu in. displacement. Similar data was computed for spark-ignition cylinders operating with a constant bmep of 140 psi. Fig. C is the result of this calculation.

To restate the basis of Fig. C, it is:

For diesel: 2200-fpm mean piston speed, bmep related to speed by Fig. B.

For spark ignition: 2200-fpm mean piston speed, 140 bmep at any speed.

This results in the following relations:

For diesel

$$\text{hp per cu in.} = \frac{42}{12 \times \text{stroke}} - \frac{220}{28.57\ (\text{stroke})^2}$$

For spark ignition

$$\text{hp per cu in.} = \frac{7}{3 \times \text{stroke}}$$

These arbitrary relations, of course, have their limitations, but they seem to be in accord with our experience. Fig. C seems to tell us that:

1. Stroke/bore ratio on diesel engines does not principally control maximum power possibilities. The choice must be made primarily on other grounds.

2. Stroke/bore ratio on spark-ignition engines has a primary influence on power possibilities.

3. Attractive power output is possible with diesel engines over a wide range of sizes.

4. High output in spark-ignition engines requires small cylinders and very large spark-ignition engines with conventional combustion processes are probably not feasible at all.

It should be pointed out that the author has almost confirmed the conclusion that stroke/bore ratio is not important on diesel-engine power in his Figs. 10 and 11. It seems that the 1.30 stroke/bore ratio engine is generally superior to the others in all respects and perhaps must be ignored. Regarding the other three engines, then, of 1.45, 1.04, and 0.85 stroke/bore ratio, little can be concluded.

It is likely that almost everyone can agree with these general conclusions from their own experience and that the data presented merely ties these experiences together. This leads to speculation upon why these limitations apply, which is a separate discussion in itself. To summarize briefly, the dominant factors in the diesel engine probably are:

1. Raising diesel speeds is more difficult than raising bmep because of the combustion system and injection system limitations. In diesel engines, bmep is dependent upon the richest fuel/air ratio which can be used without smoke. Because it is more and more difficult to control the injection at high speed, the smoke limit is reached at a lower bmep at higher speed. This is even more evident in supercharged engines where the fuel quantity per stroke is higher. If the injection duration is to be kept short at high speeds, the limit of the system is reached mechanically and hydromechanically. From this discussion, it is apparent that combustion is dominant, whether open or divided chamber, whether quiescent or turbulent, and so on.

2. Thermal loads are already the limitation in high-output diesels, and raising the bore size and shortening the stroke are likely to aggravate the problem as the author has pointed out.

3. As the author again has pointed out, large-bore, short-stroke diesels have an acute problem in combustion-chamber design in preventing wasted air which never participates in the combustion process.

The information gathered from this paper is very pertinent and valuable and is generally in agreement with our experience. It must be pointed out again, however, that all of the foregoing discussion and the author's paper are related to *potential performance* only. Conclusions reached cannot be applied and might even conflict with other major requirements of the engine. Nevertheless, this paper will assist us in concentrating upon the real limits to diesel progress and in pushing them forward.

Wants Further Surface/Volume Analysis

—Vladimir L. Maleev

IN view of the present trend to a small stroke/bore ratio in automotive gasoline engines, the topic of this paper is very timely and of practical interest.

The two series of engines, or rather of engine sizes, selected for the tests were well chosen. The results of the tests are consistent and quite conclusive since they showed that for the tested type of diesel engines there exists a definite optimum stroke/bore ratio of about 1.3. It is of interest to note that this stroke/bore ratio is commonly used in many diesel engines of various designs and sizes.

In respect to the surface/volume analysis the writer wonders, why did the author bother with the formula which neglects the influence of the side walls, Fig. 17, which he himself calls oversimplified. The error introduced by the oversimplification, as compared with the more accurate formula that takes into account the side walls, Fig. 18, is from 15 to 20%, certainly not negligible. It is also regrettable that the author did not carry his surface/volume analysis further and did not explain why the trend towards small stroke/bore ratios existing in spark-ignition gasoline engines cannot spread to diesel engines. In diesel engines a decrease of the surface/volume ratio is desirable for the sake of reducing heat losses to the walls and, as illustrated by Fig. 20, is obtained by an increase of the stroke/bore ratio. Conversely, in gasoline engines an increased surface/volume ratio and the resulting increase in heat rejection to the walls helps to reduce the danger of detonation and thus permits use of a higher output and thermal efficiency. Fig. 20 shows that a modern spark-ignition engine with a 4-in. bore x 3-in. stroke, or a 0.75 stroke/bore ratio, and an 8/1 compression ratio has a surface/volume ratio of about 6. In order to have the same surface/volume ratio a 4-in. bore diesel engine with a 16/1 compression ratio must have, as shown in Fig. 20, a stroke/bore ratio of 1.5 which is in line with the results of the discussed tests.

This paper brings out one more interesting fact, namely that

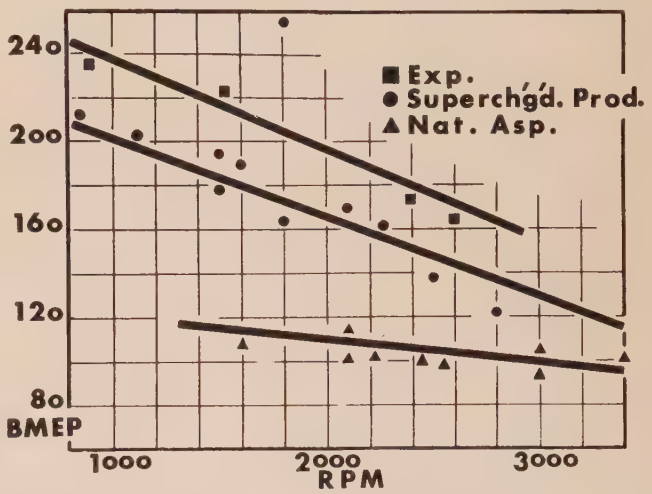


Fig. B—Effect of rpm on bmep of high-output, 4-cycle diesel engines

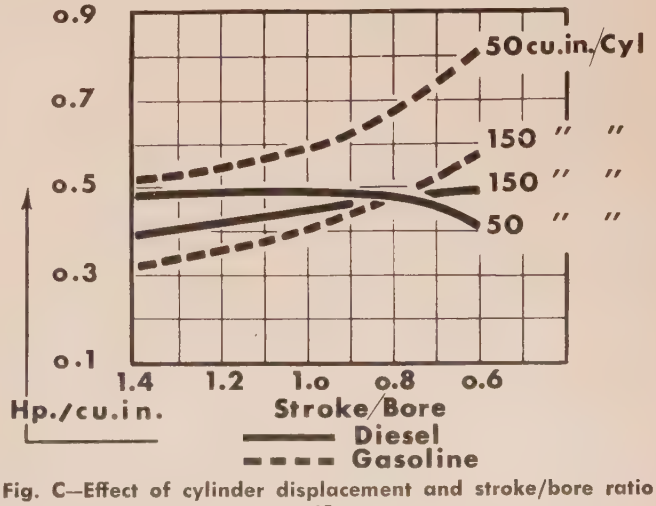


Fig. C—Effect of cylinder displacement and stroke/bore ratio on specific output

the lately often-heard claim of automobile engine designers that a decrease in the stroke/bore ratio reduces friction losses is at least doubtful. While Fig. 5 may be interpreted to support partly this claim at higher speeds, Fig. 9 seems to disprove it entirely.

Notes Importance of Correct Stroke/Bore Ratio

—P. O. Nelson and R. J. Smutny
International Harvester Co.

IN our discussion we will try to emphasize some of the points that the author makes, ask a few questions concerning the text, and perhaps express some of our own ideas on this many-sided task of picking the correct stroke/bore ratio.

On the subject of surface/volume ratios, we are also of the opinion and would like to emphasize that it is important to minimize this ratio at or near top dead-center.

Considering another theoretical engine of constant displacement and compression ratio, it is not very difficult to show with the use of maximum-minimum theories that the surface/volume ratio is at a minimum when the diameter equals the height.

As Mr. Fleck has pointed out, the heat transferred from the hot gases during this portion of the cycle may affect the combustion process considerably. Now, considering our above-mentioned engine with the piston in the neighborhood of top dead-center, the only change the designer could make, regarding bore-stroke, in order to cause the combustion chamber to conform to the (diameter = height) configuration, would be to decrease the bore of the engine which, in our case, would also lengthen the stroke and increase the stroke/bore ratio.

If we consider the first 10 deg after top dead-center on the firing stroke we know that we experience high turbulence, temperatures, and pressures thus aiding heat transfer out of the combustion chamber; we also know that comparing the short-stroke to the long-stroke engine at the same rpm the piston of the longer stroke engine has traveled downward a greater distance and will approach the ideal surface/volume ratio for minimum heat transfer sooner than the short-stroke engine. When compared at the same piston speed, (that is, pistons moving identical increments), the change in volume is greater in the short-stroke engine. Therefore, the heat transfer out is lower due to a lower pressure and temperature. The difference in this case is that the short-stroke engine has the advantage of operating at a higher rpm for the same piston speed.

Another important performance characteristic related to the surface/volume ratio near top dead-center is cold-starting. To start an engine at subzero temperatures, one must minimize heat transfer out of the combustion chamber and therefore minimize surface/volume ratio.

Past experience has shown that engines with large stroke/bore ratios give the best performance at relatively low speeds. However, the recent trends to light-weight, more compact, inexpensive power packages have indicated that higher speeds are desirable. From this point of view, the short-stroke engine must be considered.

Of the two series of engines tested, we feel that the second series of tests are the more valid. The only way to make a fair comparison between short- and long-stroke engines is to compare them on the basis of equal piston displacements per cylinder. If the engines are compared on some other basis, the cylinder size enters the picture due to the effects on efficiencies, peaking speed, and outputs. It also is suggested, that in order to obtain a more complete determination of the stroke/bore ratio effects, the engines should be compared on the basis of equal piston speeds as well as equal rpm. If engines of unequal displacements are to be compared it might be advisable to compare their specific output as suggested by Taylor and Taylor.[a]

By cross-plotting data from the second series of tests, Mr. Fleck obtains Fig. 11. This graph shows that the maximum power occurs at the same stroke/bore ratio for different engine speeds. Several points of interest are noted in regard to this graph. If we consider the most important factors influencing the peak points (that is, heat transfer, air capacity, and combustion and mixing) we might see that the data presented is not entirely conclusive. Considering each factor independently, we can discern the effect on the peaking point as far as the stroke/bore ratio is concerned. First, if the combustion and mixing remain unchanged between engines, the peaking point would tend towards a higher stroke/bore ratio as the speed is increased. By improving the combustion and mixing, as the speed increases, the peak points probably would occur at the same stroke/bore ratio. It is granted that at high speeds the combustion and mixing problem will become quite involved. Second, from a heat-transfer standpoint, as mentioned previously the engine prefers a high stroke/bore ratio. Third, from an air-capacity standpoint, the engine prefers a low stroke/bore ratio. Maintaining optimum combustion and mixing, it is our opinion that the air-capacity advantages might offset the heat-transfer losses at high speeds. Thus, the peak points would tend toward a lower stroke/bore ratio as the speed is increased. More test points between 1.04 and 1.30 stroke/bore ratio would prove or disprove this point.

In conclusion we would like to ask Mr. Fleck a question. In reading the paper, since nothing specific was mentioned about compression ratio, we assumed that the compression ratio was kept constant for the two series of engines. Is this assumption correct?

Stroke/Bore Ratio Affects Present, Future Trends of Diesel-Engine Development

—C. N. Guerasimoff
Allis-Chalmers Mfg. Co.

THE analysis of the two groups of trend-performance curves, one covering the results of the test of a single-cylinder engine in which the bore was kept constant and the length of the stroke and consequently the displacement was varied, and the other where the displacement of the cylinder was kept substantially constant and the stroke/bore ratio was varied, proves conclusively that the Caterpillar D326 and D337, Series F, 6-cyl, 5⅛-in. bore by 6½-in. stroke engines have demonstrated much better all-around performance due to increased stroke/bore ratio than their 6-in. stroke predecessors.

The author's conscientious effort in the investigation of this controversial stroke/bore ratio subject not only enlightens us on what can be accomplished with diesel engines at the current level of their progress, but also sheds some suggestive light on the probable trend of future diesel-engine development.

It should be noted that historically, in the early stages of its development, not only the diesel but the gasoline engine as well, had been a long-stroke engine.

Being a predecessor of the diesel engine, the gasoline engine, in the course of technological evolution had undergone a change of its gradually diminishing stroke/bore ratio at an earlier date than the diesel engine which is lagging behind it by at least a quarter of a century.

Comparing, for example, the stroke/bore ratio of five leading automotive engines, Cadillac, Chrysler, Lincoln, Buick, and Packard using their largest models, we find that in 1929, the average stroke/bore ratio was 1.45. In 1939, this average ratio had changed to 1.344. Twelve years later, in 1951, it dropped to 1.13, and in 1956, the average figure for engines of the same make was 0.908.

This gradual change in stroke/bore ratio has been prompted by growing automotive field requirements for lighter powerplants of higher operating speed and higher specific output.

In this process of progressive improvement, the gasoline engine has been gradually changing in its thermodynamic characteristics as well as physical structure. The combustion chamber had changed from L-head to a more efficient overhead design, and cylinder-block construction had changed from in-line to a more compact and more economical V-type.

In going back to our diesel engine and trend curves presented by the author, we find that although, under the given set of experimental conditions, some performance shortcomings related to the short-stroke engines were observed, several desirable features were also revealed in these tests in connection with the short-stroke/bore ratio engines which are as follows:

1. Higher specific output at maximum bhp.

[a] "Internal-Combustion Engine," by C. F. Taylor and E. S. Taylor. Pub. by International Textbook Co., 1943.

2. Lower frictional losses in terms of fmep almost through the entire engine speed range of the second group of engines of practically constant displacement and variable stroke/bore ratio.

3. Lowering of stroke/bore ratio by increasing the bore and keeping stroke constant results in a better performing engine, a condition no doubt observed by many engine manufacturers on bored-out engines and confirmed by the experimental data of the paper.

4. Although the performance of the long-stroke engine in the first group suits tractor application very well, where 25% torque rise on lug is very desirable, it indicates at the same time that deterioration of volumetric or mechanical efficiencies or both in long-stroke engines is proceeding at a considerably higher rate toward the high-speed end, than in an engine of lower stroke/bore ratio. It should be noted that this feature is well recognized by the author and accepted as justifiable in the use of a short-stroke engine where a combination of reasons and design economics prevail over some optimum performance characteristics.

Speaking of shortcomings of a low stroke/bore ratio engine, unfavorable surface/volume relation of the combustion chamber can be effectively remedied by the use of several helpful expedients:

1. In discussion of the paper, it has been stated that necessary changes have been made in the size of the precombustion- and main combustion-chamber proportions to assure the same 16/1 compression ratio on all engines tested. Since comparisons were made of short-stroke engine operation at higher speeds than that of the long-stroke model, an additional adjustment could have been made to assure combustion efficiency at higher speeds on the short-stroke engine to be the same as that of the longer stroke engine at lower speed. This we found to be necessary in connection with the Lanova-type chamber where, for higher operating speed, it is necessary to increase the energy-cell orifice to keep the rate of discharge of gases from the energy cell into the main chamber in step with a shorter time element available at the increased engine speeds.

This is true of any type of diesel combustion process employed, be it precombustion chamber, energy cell, or direct-injection type. In a direct-injection type the increased rate of the combustion process can be achieved either by higher fuel atomization and better distribution or by the use of induced turbulence in addition to the normal rate of flame propagation.

Under the circumstances, a comparison of performances of the two opposing engines in question made on an indicated, instead of a brake power basis, would have illustrated my point most convincingly.

2. Suppressing the effect on power and fuel economy of the piston to cylinder-head clearance volume, or "lost volume," using the author's terminology in short-stroke engines can be materially reduced by keeping the clearance to a minimum, consistent with practical aspects of engine operation. In the course of development of the G/M 4–51 engine of 4.1-in. bore and stroke, it has been found that by reducing piston to cylinder-head clearance from 0.039-in. to 0.020–0.025-in. limit, a gain of approximately 7% was made both in power and fuel economy at maximum engine speed and load rating.

3. If surface/volume ratio of engine clearance is used as a yardstick of comparison of thermal efficiencies of short- and long-stroke engines, rather than comparing for example, the engines of 1.21 and 1.45 stroke/bore ratio, there is no reason at all why the surface/volume ratio of a 1.21 stroke/bore ratio engine cannot be made equal or much smaller than that of the longer stroke 1.45 ratio model. This improvement can be attained through turbocharging, requiring increased combustion-chamber volume with consequent reduction of the surface/volume ratio figure. If turbocharging is carried far enough as in the case of Napier Nomad engine, some phenomenal results can be attained both in power and fuel economy. In a case of the Nomad 6 x 7⅜ bore and stroke, 12 cyl, 2-cycle engine with 1.21 stroke/bore ratio and turbocharged to 89-psia intake manifold pressure, the maximum power output of 3100 hp at 2050 rpm was attained corresponding to 240-psi bmep and specific fuel consumption of 0.34 lb per bhp-hr.

[b] NACA ARR 4E24, February, 1945, "Effect of Changing Stroke on Air Capacity, Power Output, and Detonation of Single-Cylinder Engine," by J. C. Livengood and J. V. D. Eppes.

Discusses Effect of Stroke/Bore Ratio on Performance

—C. Fayette Taylor
MIT

MR. FLECK'S paper attacks a very fundamental problem in engine design, which has received all too little scientific attention and has been the subject of altogether too much uninformed controversy.

As the author very properly points out, many factors influence the optimum stroke/bore ratio. This discussion will be confined to questions of performance only.

The indicated output of a cylinder is proportional to the product of mass rate of airflow, fuel/air ratio, and indicated efficiency. In studying the effect of stroke/bore ratio on performance, it would be well to separate these factors as far as possible. Since fuel/air ratio is not controlled by stroke/bore ratio, this quantity should be held constant.

Fig. D shows results of test made at MIT [b] on a single-cylinder, spark-ignition engine where the stroke was changed, together with appropriate changes in top-center position of the piston to keep compression ratio constant, and in connecting-rod length to keep the crank/rod ratio constant. Fuel/air ratio, temperatures, and inlet and exhaust pressures were held constant.

The striking feature of the results shown in Fig. D is that volumetric efficiency and air consumption are independent of stroke at any given piston speed. This result would indicate that mean velocity through the valves is the controlling factor in volumetric efficiency, provided there are no serious pressure fluctuations in the inlet system. In our tests such fluctuations were avoided by the use of large inlet and exhaust receivers and very short inlet and exhaust pipes.

Indicator cards taken with the three strokes at various piston speeds showed that, with fuel/air ratio constant and spark advance for best power, indicated efficiency remained measurably constant. Thus, indicated power output was the same for each stroke at a given piston speed.

Differences in brake output were due to differences in friction. The friction mep with a short stroke was higher than that with a larger stroke, at a given piston speed, on account of the higher rpm of the crankshaft bearings.

These results appear to justify the conclusion that maximum breathing capacity, and therefore maximum *potential* output, is independent of stroke, provided only that the same maximum piston speed can be used with all strokes.

These results also seem to justify the conclusion that, at least in these tests, changes in heat loss due to changes in combustion-chamber shape were too small to effect efficiency.

For those who may be interested in spark-ignition engines, as well as in diesel engines, Fig. E shows the knock-limited inlet pressure to be highest with the short-stroke cylinder, at a given piston speed.

Limitations on piston speed as stroke gets shorter and rpm correspondingly higher might be:

1. Inertia stresses in the valve gear, which increase as rpm^2, with a given bore.

2. Inertia stresses in the crank-rod-piston system, which depend on mass × radius × $rpm.^2$ Since piston mass is little affected by shortening the stroke on a given cylinder, it is obvious that such stresses increase as stroke is shortened at a given piston speed.

3. Combustion limitations in a diesel engine, which might limit rpm rather than piston speed or might reduce efficiency as rpm increases.

Regarding Mr. Fleck's paper, it appears that the results show the combined effects of changes in breathing capacity, mechanical friction, pumping loss, combustion efficiency, heat loss, and possibly also in fuel/air ratio. Heat loss is included here, because with the type of combustion chamber used, changes in geometry can conceivably have much larger effects than in the case of the MIT spark-ignition cylinders.

Before the results of such tests can be generalized, it would be necessary to try to separate these various factors. Such separation could be accomplished provided heavy- and light-spring indicator cards and measurements of air consumption and heat to jacket water are available.

I would like to ask Mr. Fleck if such data could be made

available. I would also like to ask if the crank/rod ratio was the same for the various cylinders tested, and if the fuel/air ratio was held constant during the tests.

Discusses Effect of Air/Fuel Ratio on Diesel Engine Performance

—R. H. Miller
Nordberg Mfg. Co.

MR. TAYLOR told us that as far as the gasoline engine is concerned, the only effect of stroke/bore ratio is ability to get the engine under the hood of an automobile. Mechanical and thermodynamic advantages claimed for the stroke/bore ratio which meets this space requirement are largely nonexistent.

Assuming that combustion-chamber efficiency remains constant at all stroke/bore ratios, the same will hold for the diesel engine on the basis that all stroke/bore ratios are operated at the same rotative speed.

The combustion efficiency and, therefore, the thermal efficiency of the diesel engine falls off with rotative speed but is even more sensitive to air/fuel ratio.

The effect of air/fuel ratio is shown in Fig. F. This typical curve has been plotted from test data at 500 rpm. Plotting other rotative speeds would produce a family of curves showing decreasing thermal efficiency at any air/fuel ratio with increasing rotative speed.

The failure of the diesel engine to operate at its otherwise inherent high thermal efficiency at low excess-air ratios is due to the short time available for mixing of fuel and air in the combustion chamber.

Hesselman accomplished thorough fuel/air mixing by injecting the fuel into a highly turbulent air some 80-deg before top-center in the compression stroke. However, to prevent autoignition, the compression ratio had to be reduced to about six and the mixture fired with an electric spark.

This engine operated with high combustion efficiency at low air/fuel ratios but suffered from low cycle efficiency due to the small expansion ratio which was equal to the compression ratio as in all conventional engines.

The Atkinson cycle combines the advantages of high combustion efficiency with low excess air of the Hesselman engine and the high cycle efficiency of the diesel engine.

Miller System Supercharged Atkinson cycle engines of high power output, operating at 190 bmep with an indicated thermal efficiency of 50%, are being built in this country by us.

Vehicles equipped with engines built in accordance with this system would operate with twice the mileage per gallon now obtained with our present-day thermodynamically crude and mechanically superb automotive engines.

For equal horsepower output, the engine is of smaller displacement and lighter weight.

Since the compression ratio is a matter of choice and does not affect output or thermal efficiency, low-octane fuels or diesel fuels can be used successfully.

The cost of developing this supercharged Atkinson engine for an automobile would probably be not more than the cost of the annual change of the body tail fins but would benefit the operator considerably more.

Shows Stroke/Bore Ratio Effects on Efficiencies

—A. Raymond White
Chrysler Corp.

THIS is a very timely paper, and its careful documentation gives it an authority which such a subject might otherwise lack. It would seem, however, that the author has been unfortunate in his choice of references, for their paucity and lack of emphasis lead one to suppose that stroke/bore ratio has had little attention in engine design. Actually this factor has been one of the most bitterly controversial questions in automotive history, and references could range all the way from a 1907 classic by F. W. Lanchester [c] (that demon anticipator of everybody) to a 1938 paper which sought to prove that the optimum stroke/bore ratio should equal the ratio of specific heats of air.[d] If the discussion has been less heated in recent years, it may be ascribed to a better perspective, which recognizes that stroke/bore ratio is merely one facet of a larger problem and not an end in itself.

As Mr. Fleck points out, the compression-ignition engine traditionally has been a long-stroke engine. In the past there has been a suspicion that this might be due to uncritical acceptance of European dictation, as happened in the passenger-car field 40 years ago. However, it may be shown quite convincingly that the compression-ignition engine, because of its high compression and low speed, can profitably use a larger stroke/bore ratio than the Otto cycle engine. Since this fact may be overlooked in analyzing the author's results it is hoped that a few remarks directed specifically toward passenger-car engines may be in order.

Whatever may be said about compression-ignition engines, it can be stated unequivocally that in Otto cycle engines there is remarkable correlation between the surface/volume ratio of a combustion chamber and its thermal efficiency. This ratio:

$$R = S/V = \frac{k(n-1)}{qD} + \frac{4}{D} \qquad (1)$$

where:

k = Coefficient depending on chamber shape
n = Compression ratio
q = Stroke/bore ratio
D = Cylinder bore

In this equation D is not an independent variable, but since we are discussing cylinders of constant displacement it is apparent that D^3q = constant, or $D = \left(\frac{C}{q}\right)^{1/3}$. Substituting in equation (1), differentiating, and equating to zero leads to the result that the ratio S/V is a minimum when the stroke/bore ratio:

$$q = \frac{k}{2}(n-1) \qquad (2)$$

When the combustion chamber is an extension of the cylinder, the value of k is 2; for a spherical segment chamber it is less (about 1.7, though this cannot be generalized); and for a side-valve chamber it is 3.55 $\left(= \frac{8}{\pi} + 1\right)$. This last value was derived theoretically and then confirmed by analysis of a dozen chambers, all made to the same design philosophy but varying widely in size.

From equation (2) it is apparent that, if thermal efficiency were the only design criterion, stroke/bore ratios could profitably be increased far beyond present values. It is improbable that these considerations had anything to do with the engines used in the early "Coupe de l'Auto" races; but in the light of this equation the stroke/bore ratios of 3.5 to 4.0 used on these monstrosities seem much less ridiculous.

But thermal efficiency is *not* the only design criterion. In pasenger-car engines stroke/bore ratio has a substantial effect on volumetric efficiency and a marked effect on mechanical efficiency. Though these factors may exert minor influence in low-speed compression-ignition engines, they rapidly assume major importance as the engine speed increases, and in passenger-car engines they drastically modify the results of equation (2). About 12 years ago an attempt was made to determine optimum stroke/bore ratio for high-speed engines by analysis and test. Engines with stroke/bore ratios from 0.5 to 1.5 were built and tested with the result that maximum economy seemed to be obtained with cylinder proportions close to the lower value. The analysis of this problem admittedly involves the difficult task of predicting fmep, and one cannot afford to be too dogmatic about the results. However, it is the writer's personal opinion that for engines of 5000-rpm peak power the optimum stroke/bore ratio is between 0.5 and 1.0, and the present vogue of cylinders with stroke/bore ratios close to 0.75 tends to confirm it. This result certainly is not new. At the time when Lanchester presented the paper mentioned above his company was building 4- and 6-cyl engines with 4-in. x 3-in. cylinders, and the fact that we have caught up with this type of thinking merely confirms the admitted genius of the man.

With regard to Mr. Fleck's published results, it is felt that the first series of results with varying displacement of cylinders is not applicable to the investigation of optimum stroke/bore ratio. In equation (1) above both bore and stroke are in the denominator; consequently, any increase in cylinder displacement will im-

[c] *Proceedings of IAE*, Vol. 1, 1906–1907, pp. 155–200: "Horsepower of Petrol Motor in its Relation to Bore, Strke, and Weight," by F. W. Lanchester.
[d] *Automobile Engineer*, Vol. 28, September, 1938, pp. 339–340: "Stock-Bore Ratio," by J. Bowie.

prove thermal efficiency regardless of stroke/bore ratio. This fact is too well confirmed to need laboring. By the same token the author's statement that "Theoretically, the brake horsepower would be the same regardless of stroke for a fixed bore size, etc." is a direct contradiction of his thesis that surface/volume ratio affects thermal efficiency.

The second set of results for constant displacement cylinders is very interesting and valuable though one might wish for better correlation. The conflicting conditions for thermal efficiency and mechanical efficiency should be continuous functions of speed, and there is no apparent explanation for the outstanding performance of the 1.30 ratio engine. Before accepting this result it would be well to determine if this engine was a sport, either in bearing fits or otherwise to explain the reason for its low friction.

ORAL DISCUSSION

Reported by E. H. Middendorf

International Harvester Co.

R. Miller, Nordberg Mfg. Co.: A gasoline engine operates best with about 25 to 30% excess air.

C. F. Taylor, MIT: Indicated horsepower is proportional to the engine air capacity as shown by tests conducted on a spark-ignition engine. Also, inertia forces become greater with larger bore engine due to heavier pistons. Combustion limitations are also encountered on diesel engines at higher speeds when the bore diameter is increased.

S. K. Chen, International Harvester Co.: Surface/volume ratio is less in an open-chamber engine than in a precombustion-type chamber. If the surface/volume ratio is so important, then why are the precombustion-chamber type engines built when the open-chamber type has less surface/volume ratio? Do the combustion characteristics remain the same at higher engine speeds? Are the same injection nozzles, combustion conditions, orifices, and timing used?

Mr. Fleck: I do not guarantee that the data were obtained running at absolutely the same combustion conditions. The orifices were changed, but the combustion chambers remained the same. The injection system also remained the same in the second series of engines, but the timing was changed.

P. H. Schweitzer, Pennsylvania State University: The 1.30 stroke/bore ratio is also well adapted for 2-cycle engines. There is no magic number in making the stroke shorter and shorter.

Mr. Fleck: It is desirable to gather some data at other than top dead-center considering surface/volume ratio. Also, a lower surface/volume ratio would improve cold-starting, but with the use of starting aids this problem is not serious.

Although bmep falls off with speed in the diesel engine, gasoline engines also encounter the same trouble. The 1.30 stroke/bore ratio was selected because it looked favorable from certain standpoints. Friction, rather than combustion, is a detrimental factor in diesel performance.

Author's Closure To Discussion

Regarding Mr. Klinge's discussion, he expressed willingness to throw out the 1.30 ratio because it was good, and also because by doing so, it became easier to say that stroke/bore is not a factor. The designer could then go ahead with a free hand and leave the performance details up to the development engineer. In checking over our results, we can find nothing unusual about this particular engine favorable to it except the stroke/bore ratio. However, as mentioned in the paper, we do question the maximum power data of the 1.04 ratio engine at the higher speeds, because of some definite porting restrictions. By ignoring the 1.04 points in Fig. 11, curves can be drawn through the remaining three points, which may indicate an optimum stroke/bore ratio some place between 1.04 and 1.30 as suggested by another discusser.

Mr. Klinge mentioned that he believes the combustion and injection system is one of the dominant factors in limiting the speed of diesel engines. In the speed range of our tests, combustion did not appear to us to be a limiting factor. Fig. 13 indicates only a small per cent of difference between 0.3 smoke (light) and engine maximum (infinitely dense smoke), at 2400 rpm, which is one indication of good combustion. We have made small diesel engines which were run with no apparent combustion difficulties up to twice the speeds indicated here.

In reply to Prof. Taylor's comments and questions, the results shown do represent the combined effects of breathing capacity, mechanical friction, pumping loss, combustion efficiency, and heat loss. The possibility of fuel/air ratio affecting the results of the absolute maximum brake horsepower shown is slight because of our procedure in running maximum which is to gradually increase the fuel/air ratio until the addition of fuel results in a constant or even a slight decrease in engine brake horsepower. Complete heat-rejection data were run. We normally do

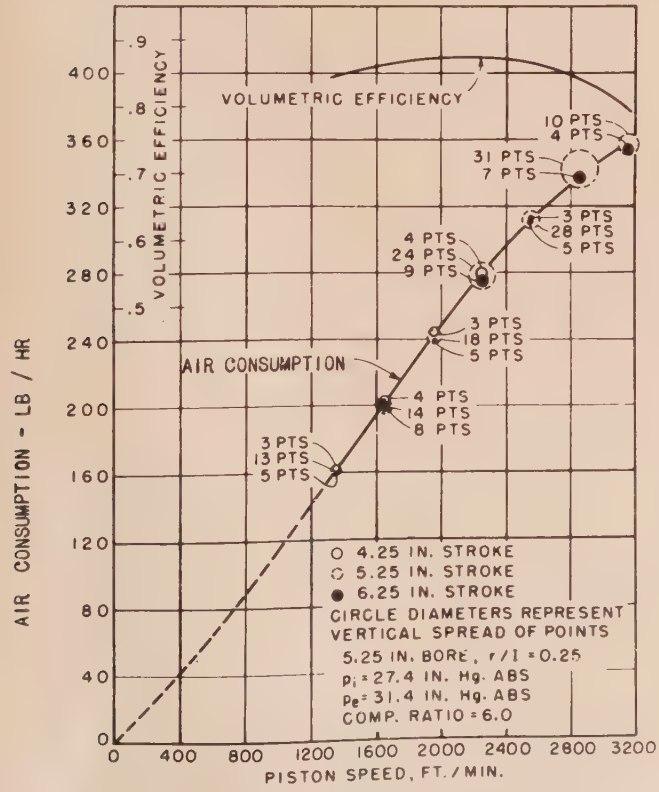


Fig. D—Effect of changing stroke/bore ratio on volumetric efficiency and air consumption

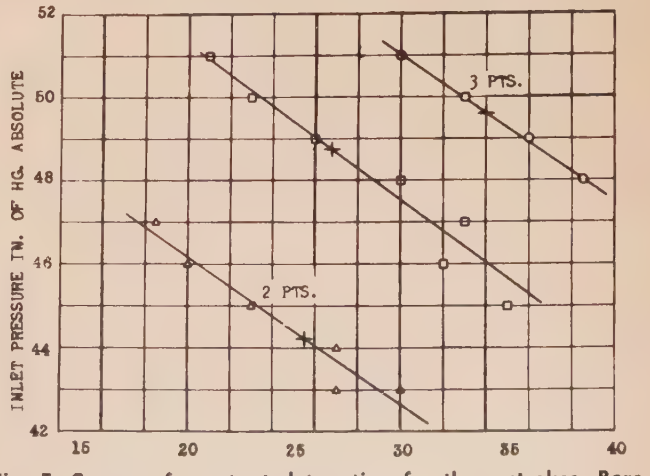


Fig. E—Curves of constant detonation for three strokes. Bore = 5.25 in., $r/_2 = 0.25$, compression ratio = 6.0, P_e = 31.4-in. Hg. absolute, piston speed = 2400 fpm

not take indicator cards suitable to calculate indicated horsepower, but will be glad to supply those interested with the exact readings of brake horsepower and friction horsepower for computing purposes. The crank/rod ratio varied between the various engines to permit the use of some available experimental parts.

The actual ratios of crank radius/rod length for the second series of engines, where the displacement was nearly constant, were, from short to long stroke—0.16, 0.19, 0.25, 0.33. This gave the following combustion-chamber volume increases in cubic inches at 20-deg atc of 5.6, 5.6, 5.9, 6.1. In the first series of engines where the displacement varied with stroke, the ratios were 0.23, 0.27, 0.30, 0.33 from the shortest to longest stroke, giving volume increases at 20-deg atc of 4.0, 4.7, 5.2, 6.9 cu in. or a per cent increase relative to displacement of 3.7, 3.8, 3.8, 3.9. Therefore, the rate of volume change per degree of crank angle near top center was nearly proportional to total displacement, and when comparing engines at the same rpm, the time factor would be nearly constant when compared with the per cent of volume change.

I would like to say that I can heartily agree with most of the comments made in the discussion by Mr. Nelson and Mr. Smutny. We agree that there are conflicting factors in the needs of the engine which wants both a small and a large stroke/bore ratio. The problem seems to be one of determining how low you can go, rather than to conclude that a 1.3 ratio is optimum. As indicated in Fig. 19, the optimum ratio may become smaller as the cylinder displacement is increased.

In answer to the question concerning compression ratios, it was held nearly constant on the second series of engines. The ratio was changed on the first series of test engines to conform to our development experience of compression ratio requirements as related to cylinder displacement. In this series, the ratios varied from 18.0 to 16.5 from the smallest to the largest displacement engine.

In answer to Mr. Maleev's question about why I included Fig. 17, it actually represents more closely the situation which exists in diesel engines at top center than Fig. 18. Efforts are usually made to minimize any access to the cylinder wall at this point. In other words, considerable advantage can be gained with various combustion-chamber configurations to minimize the surface in contact with the hot gases near top center. Mr. Maleev explained one of the uses for curves such as Fig. 20.

Mr. Guerasimoff has suggested several remedies for shortcomings of the low stroke/bore ratio engine. While it is true that a larger orifice is needed for higher speeds, it is also our experience that overdoing this will result in considerable loss in power at lower speeds. A look at the slope of the bmep curves in Fig. 7 indicates that the low ratio engine has very little excess bmep at the low-speed end. In my opinion, the other engines in the first group would have more chance to gain power at 2400 rpm with an over-sized orifice, rather than the low stroke/bore ratio engine.

In regard to piston-to-head clearance, we try to keep this down to a practical minimum. This dimension is probably, as Mr. Guerasimoff indicates, more critical on a low stroke/bore ratio engine, and should be controlled closely.

The application of turbocharging does not appear to me to be a factor in reducing surface/volume ratio. Any reduction in compression ratio anticipated due to supercharging could be applied to all the engines under consideration.

Mr. White's remarks were very interesting. His formula (1) looked familiar and, if the value 2 is used for K, is the same as that used to construct the curves in Fig. 19. The formula (2) presents another slant on the subject. It indicates that for the very minimum in surface/volume ratio, compression ratio alone determines the stroke/bore ratio. Immediately this suggests approximately a 15/1 stroke/bore ratio for a diesel engine which can be visualized by extending the horizontal scale to the right in Fig. 19. After considering this, it makes one want to see how far we can swing to the left before getting into difficulties.

One of the most appreciated suggestions made by Mr. White was to check on the remarks of Lanchester made in 1906–1907. We did not have this reference but were able to order a copy of it from his company library. For anyone interested in reading the remarks of a man speaking 50 years ago, who was perhaps more than that much ahead of his time, along with the remarks of his friends and critics, I would recommend that the reader look

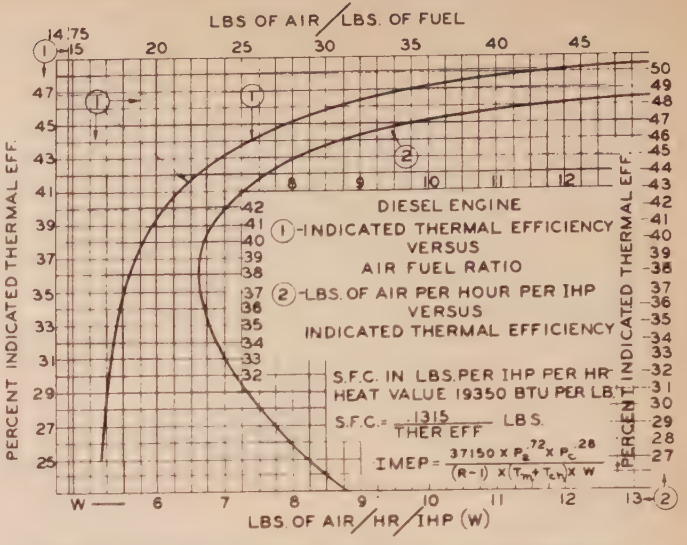


Fig. F—Effect of air/fuel ratio on thermal efficiency

Table A. Test Engine Characteristics

Test Engine	Displacement	Actual bhp	Calculated $B^{1.3} \times S^{0.7}$
5.12 x 5.28	109	28.2	28.8
5.12 x 6.00	124	31.0	31.2
5.12 x 6.50	134	32.7	33.2
5.12 x 7.41[a]	153	36.5	36.5
5.38 x 7.00	159	43.3	37.4
5.75 x 6.00	156	37.5	36.2
6.25 x 5.28	156	38.2	37.2

[a] Taken as standard for basis of calculating the other engines.

Table B. Test Engine Characteristics Using Lanchester's Formula

Test Engine	Displacement	Stroke Needed as per Lanchester Rating Formula ($N = 1.5$)	Displacement
5.12 x 7.41	153	7.41	153
5.38 x 7.00	159	6.37	144
5.75 x 6.00	156	5.24	136
6.25 x 5.28	156	4.10	121

into this paper. If it were published today, it would probably be called an "Advanced Theory of Engine Dimensions."

One theory developed which may be familiar to the reader was that

$$\text{hp} \propto \sqrt{B^3 S}$$

or

$$\text{hp} \quad B^N \times S^{2-N}$$

with the sum of the two indices always equal to 2, or square rather than cubic.

Modifying N to fit our data presented, gives the formula

$$\text{hp} = B^{1.3} \times S^{0.7} \times C$$

where:

B = Bore
S = Stroke
C = Constant

This is illustrated in Table A

This shows close correlation with Lanchester's law that hp $\propto L^2$. This conception indicates that, if we are only interested in maximum bhp, we should not compare engines, either on a constant bore, or constant displacement basis, but that the displacement should be changed as the bore is increased, at a rate depending on the value used for N.

Using Lanchester's formula hp varies as $\sqrt{B^3 S}$, and the strokes to give equal bhp are calculated for the bore sizes used in the second series of test engines as shown in Table B.

This table might explain why the second engine was better than the first, but it then suggests that the last two engines were worse than our constant displacement comparison indicated, possibly due to other circumstances already discussed which could cause a compression ignition engine to need a stroke/bore ratio other than that considered most ideal from theoretical analyses.

SOUND and FUROR

THE JET NOISE SUPPRESSION AGE

M. M. Miller, Douglas Aircraft Co., Inc.

This paper was presented at the SAE National Aeronautic Meeting, Los Angeles, Oct. 4, 1956.

THE problem of jet noise in the aircraft industry has grown in importance so suddenly that standard definitions, methods of measurement, and methods of data presentation have not been developed. This paper points out some of the areas subject to misunderstanding and makes recommendations for a few simple methods of data presentation which offer a consistent method of evaluating jet noise suppression gains. Using these methods the results of some jet nozzle suppressors tested by the Douglas Aircraft Co. are rated. Several promising methods of achieving adequate acoustic results are shown and are recommended for consideration in the growing list of jet noise suppression devices.

Sound

The acoustics engineer recognized long ago that the sound of jet engines would create a major problem in his future. Early noise measurements made out of curiosity indicated that, pound for pound, jet engines as a class were among the most efficient noise makers of all power-producing devices. The first attempts at noise suppression were associated with ground run-up and engine test and development operations, for even the early low-thrust engines wrought havoc with speech communication in the vicinity of the engine. The need for controlling this new noise resulted in the formation of study and information exchange committees (such as the Aircraft Industries Association Noise Control Committee); the arousal of professional interest on the part of physicists, physiologists, and engineers; and the establishment of an entirely new industry to provide ground suppression devices on a commercial basis. With these began the first stage of the Jet Noise Suppression Age.

In order to design these ground suppressors, it was necessary to be able to calculate the noise of a jet as a mathematical function of some known engine parameter or parameters. The search for this relationship truly began our quest for the understanding of the physics of jet noise. The author recalls calculating noise versus thrust, noise versus fuel flow, noise versus turbine-out temperature, noise versus kinetic

BEFORE an evaluation of suppressors can be made or before a sensible discussion of suppression goals can be carried on, terms, conditions, and possibilities must be understood. A five-number system for evaluating suppressors is proposed.

Methods of jet noise suppression undergoing general development today include frequency shifting, jet spreading, and jet velocity reduction.

Basic types of jet spreaders and velocity reducers are shown and ratings given. Combinations of types have resulted in increased suppression in many instances.

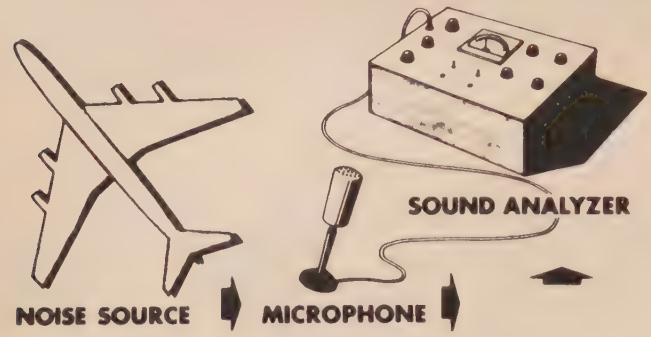


Fig. 1—Typical impression of how noise is measured

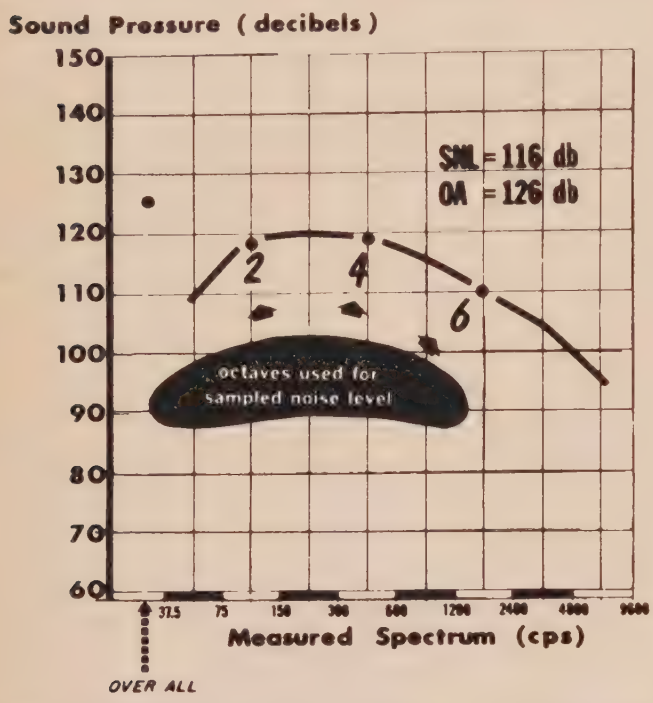


Fig. 2—Estimated octave band analysis of J-75 engine noise in direction of maximum intensity showing overall sound pressure level and octaves used in SNL calculation. Engine is in static operation

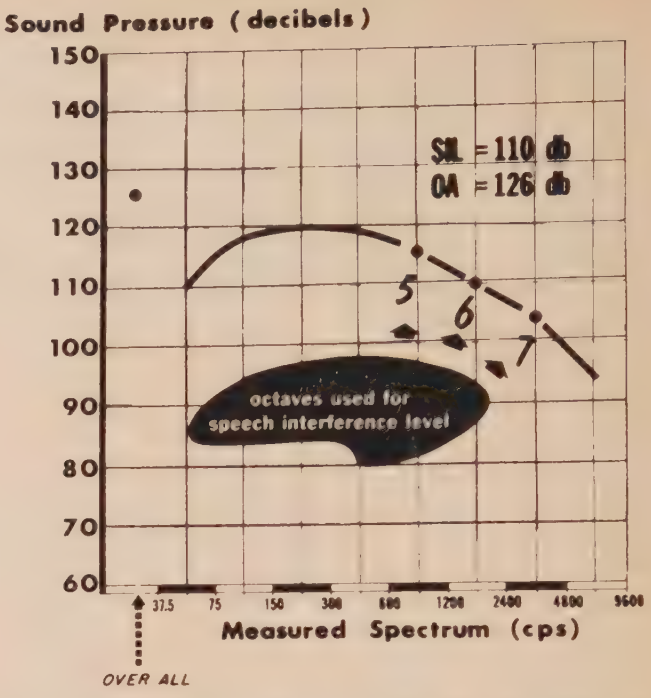


Fig. 3—Same analysis as Fig. 2 showing overall sound pressure level and octaves used in SIL calculation. Engine is in static operation

energy in the thrust-producing gases, and finally noise versus the parameter ($\rho A V^8$), a parameter more accurate than most, for which we owe considerable gratitude to Professor Lighthill of Great Britain where ρ = density of jet exaust gases, A = area of jet exhaust, and V = relative velocity of exhaust gases and outside airstream.

Stage Two of the Jet Noise Suppression Age arrived with the planning of commercial jet operations. Experience with communities closely surrounding the airports from which these airlines must operate has clearly indicated that suppression of the noise from these newer and more powerful aircraft is necessary if friendly relations are to be maintained between the airport and the neighbors over which its airborne transports must fly.

The fundamental axiom of the acoustics engineer is that "noise is best reduced at the source." This is not attempted in most ground noise suppressors, in which massive enclosures are used merely to prevent the already generated noise from getting to the outside world. The need for suppressors for flight operation has emphasized the importance of methods which prevent the generation of noise. This has in turn required more fundamental information concerning the mechanism of jet noise production and has in consequence completely opened the broad, extremely difficult field of jet noise suppression.

A jet engine produces noise differently from a current propeller recriprocating engine in three simple ways: (1) it produces more noise; (2) it produces noise of a different kind (or quality); and (3) the maximum noise is radiated in a different direction. Here the simplicity ends. In order to compare these sounds, to evaluate suppressors, and to express differences in the response of the interested human in calculable engineering numbers, a myriad of terms and expressions are available. One can begin with purely physical terms such as overall sound pressure level, octane band sound pressure level, power level, spectrum level, and peak octane level. Then he can add the response of the human being to these physical terms and continue with speech interference level, loudness level, sampled noise level, acoustical comfort index, articulation index, and others almost ad infinitum. These terms, even if properly used, may serve to confuse and confound the layman and effec-

tively prevent practically any understanding except in terms of the individual response of the human being.

Furor

Fig. 1 represents the average man's notion of the simplicity of noise measurement: "Place a microphone somewhere near a noise source, read a number off the sound meter, and the noise is adequately defined." If only it were that simple! Actually the microphone "hears" a different noise at each location around the source, and what is worse, there is no single number readable from the meter, even for a single microphone position, which adequately defines the sound. The steady insistence of those too busy to study noise that single numbers be used to evaluate and distinguish sound sources is responsible for the furor surrounding the current evaluation of jet noise suppression needs and suppressor benefits.

Three widely used one-number evaluations of a sound spectrum are the overall sound pressure level (OA SPL or OA), speech interference level (SIL), and sampled noise level (SNL).[1] Each is useful for certain purposes, each is misleading considered by itself.

Fig. 2 shows an estimated plot of sound pressure levels in octave bands of the maximum noise produced by a J-75 engine at 500 ft. This octave band analysis requires eight meter readings. The overall sound pressure level (OA) requires one meter reading. The sampled noise level (SNL), although expressed as one number is the average of three meter readings, the second, fourth, and sixth octaves. Fig. 3, representing the same spectrum, gives the speech interference level, a one-number expression representing the average of three meter readings of the fifth, sixth, and seventh octaves.

Which of these three numbers most adequately differentiates between two aircraft sounds when the frequency compositions are not similar?

Fig. 4 shows octave band analyses of the sounds from the DC-6B and DC-7 obtained at the time of greatest overall sound pressure level as the airplanes pass by the microphone (slant range 500 ft). Under these conditions, most observers will comment that the DC-7 sounds "much quieter" than the DC-6B. Yet the overall reading is essentially the same for both sounds.

Maximum noise when the airplane is operated statically, and maximum climb-out noise of the RB-66 twin-engine, J-71 jet-engine-powered airplane are shown on Fig. 5. In this case the speech interference level does not discriminate between the sounds, although observers agree that the static case is the more annoying. In both Figs. 4 and 5, only the SNL distinguishes the noise consistently with observers' judgments.

It is possible that acoustical specialists may someday agree on a satisfactory method of calculating and representing auditory responses such as loudness, for example. Until that time, however, it appears that the SNL is the only consistently calculable single number of the three terms OA, SIL, and SNL which is well-known and which accounts for the effects of low

[1] NACA TN 3379, December, 1954, "Survey of Background and Aircraft Noise in Communities near Airports," by K. N. Stevens.

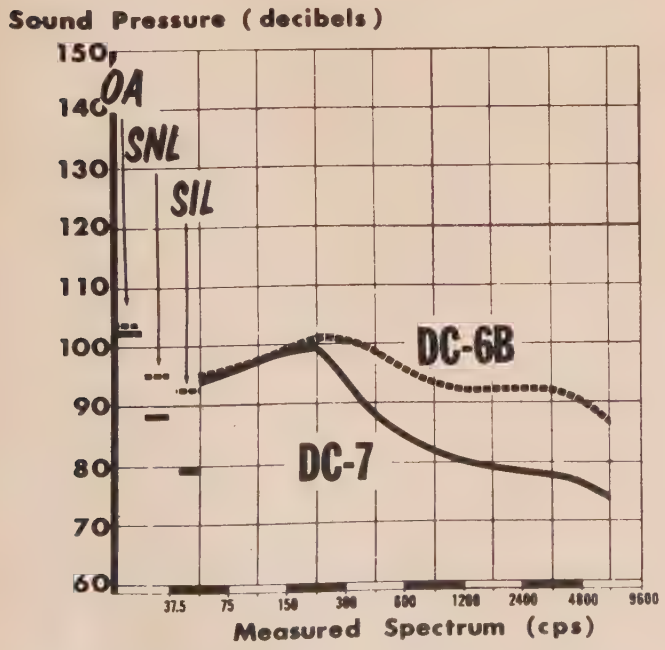


Fig. 4—Octave band analyses of noise spectra of DC-6B and DC-7 airplanes operating at initial climbout at 500 ft

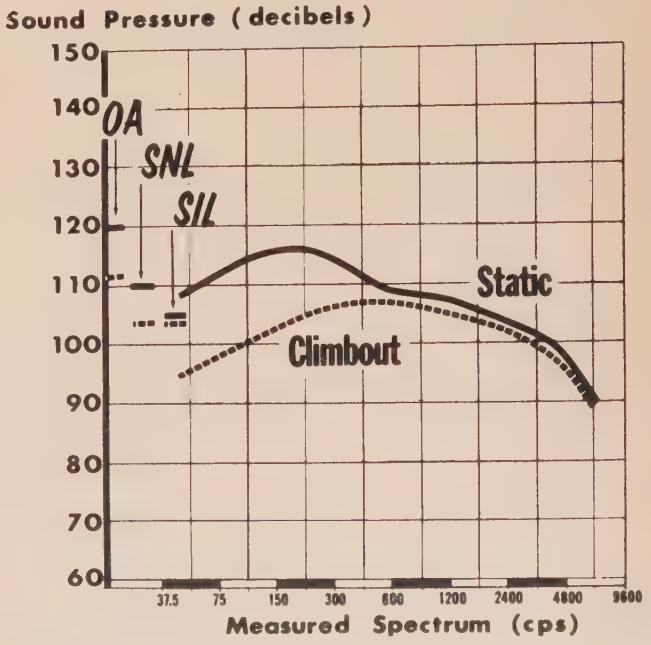


Fig. 5—Octave band analyses of noise spectra of RB-66 airplane measured with airplane operating statically and at initial climbout at 500 ft

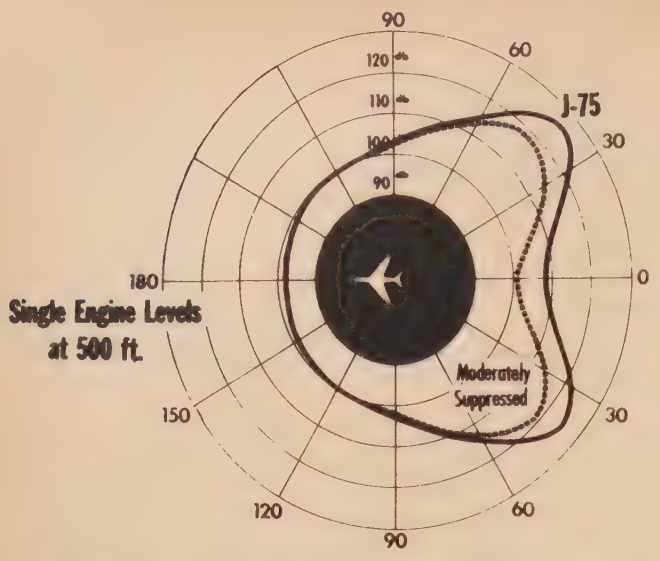


Fig. 6—Polar plot of overall sound pressure levels estimated for J-75 engine with and without moderate hypothetical suppressor of corrugated nozzle type

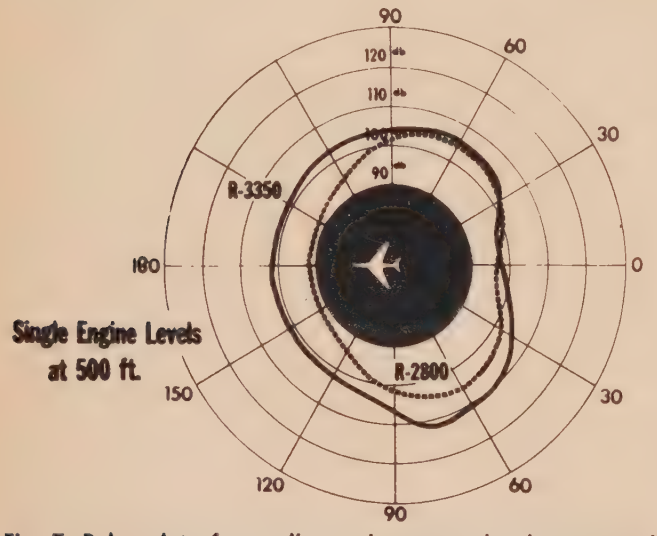


Fig. 7—Polar plot of overall sound pressure levels measured around powerplants of DC-6B and DC-7 airplanes. Engines are operating at take-off power

(vibration), middle, and high (speech interference) frequencies in the evaluation of a sound at a point.

The polar plots of Figs. 6 and 7 show the radial distribution of the estimated overall sound pressure levels of the J-75 jet engine, the J-75 engine equipped with a hypothetical corrugated nozzle suppressor (offering moderate noise reductions), and the R-2800 (DC-6B) and R-3350 (DC-7) engines. The hypothetical suppressor is introduced in the text only for discussion purposes. It does not represent recommended or expected noise reductions for the DC-8 airplane. The top half of the reciprocating-engine plots is derived from measurements made around the right sides of the DC-6B and DC-7 airplanes with only the outboard right-hand engine operating at

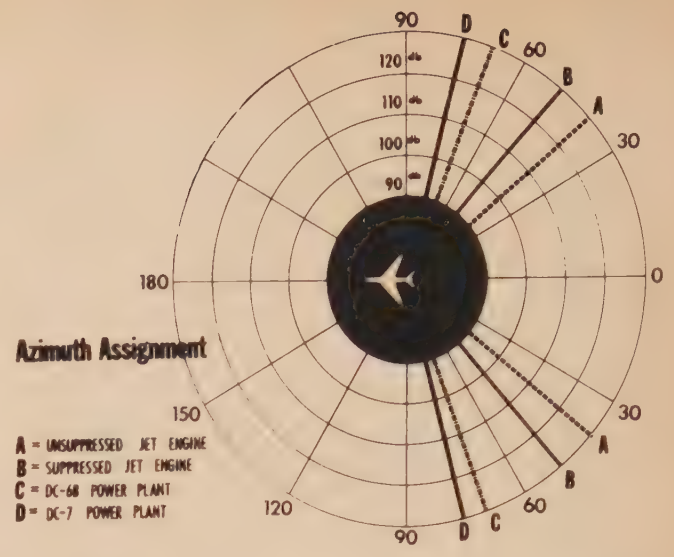


Fig. 8—Definition of azimuths of maximum noise from DC-6B, DC-7 powerplants, and jet engine with and without suppression

take-off power. The lower half of these plots represents the noise on the left side of the airplanes with only the outboard left-hand engine operating. The asymmetry is due to the differences in ground-reflection effects on the propeller noise and the asymmetry of the exhaust outlets on the DC-7 airplanes.

From these polar plots it is apparent that measurements made at a constant azimuth angle to the engines will not indicate the true differences in the maximum noise produced by the engines, since each engine produces its greatest noise in a different direction (azimuth angle).

Directions of maximum noise are indicated by the letters A, B, C, and D as shown on Fig. 8. A represents the azimuth of maximum OA of the unsuppressed jet engine. When the hypothetical suppressor is applied, the "maximum" azimuth shifts to direction B. C represents the azimuth of maximum noise of the DC-6B powerplant, and D that of the DC-7 powerplant.

Fig. 9 shows an octave band analysis and OA SPL for the J-75 engine, J-75 hypothetical suppressor, and the engines for the DC-6B and DC-7 airplanes. Each of these spectra is shown for the same azimuth angle and distance. At this fixed position, much larger differential values are shown than on Fig. 10, which shows the same spectra, but at the angles from the engines at which the noise has the greatest OA level.

Polar plots and octave band analyses of the noise at selected points on these plots are very useful in making studies of noise near airports due to ground run-ups. If it is found that jet noise suppressors reduce jet noise by the same amount in motion as when the engine is motionless, static polar measurements may be the easiest method to use for comparative purposes. Polar plots do not, however, indicate the variation in noise which the airport neighbor hears

from an airplane during take-off operations. Under these circumstances the fixed neighborhood observer effectively moves relative to the airplane along a line which lies at a constant distance from the airplane's path, rather than at a fixed distance from the airplane as a polar plot would indicate.

Jet noise is reduced due to the forward motion of the airplane. This reduction has been measured during a series of take-offs of the RB-66, the DC-6, and the DC-7. RB-66 flight speed at take-off is about 180 knots, and the DC-6, DC-7 take-off speed is about 120 knots.

Measured sideline data for the DC-6B and DC-7 airplanes are shown on Fig. 11. The recorded data for the RB-66, corrected by calculation to J-75 single-engine level, are shown on Fig. 12. The hypothetical suppressor is assumed to function as well on the J-75 during the take-off operation as it does statically.

Octave band analyses at the positions of the maximum OA levels on Figs. 11 and 12, are shown on

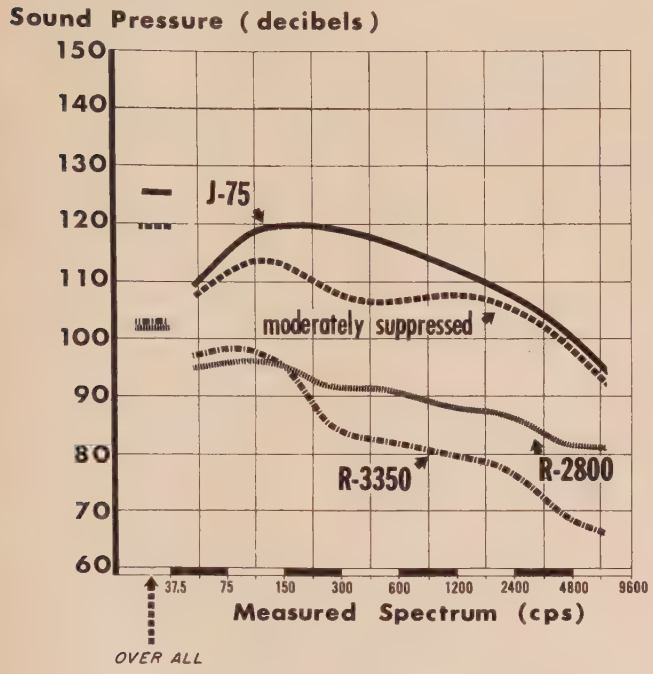


Fig. 9—Octave band analyses at static single-engine levels at azimuth A at 500 ft. Moderate suppressor is hypothetical suppressor of corrugated nozzle type

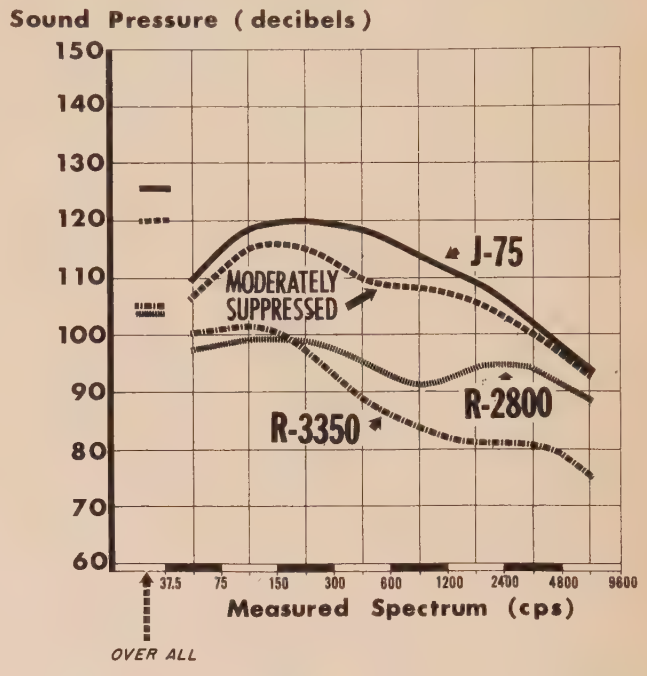


Fig. 10—Octave band analyses for static single-engine levels at maximum noise azimuth of 500 ft. Moderate suppressor is hypothetical suppressor of corrugated nozzle type

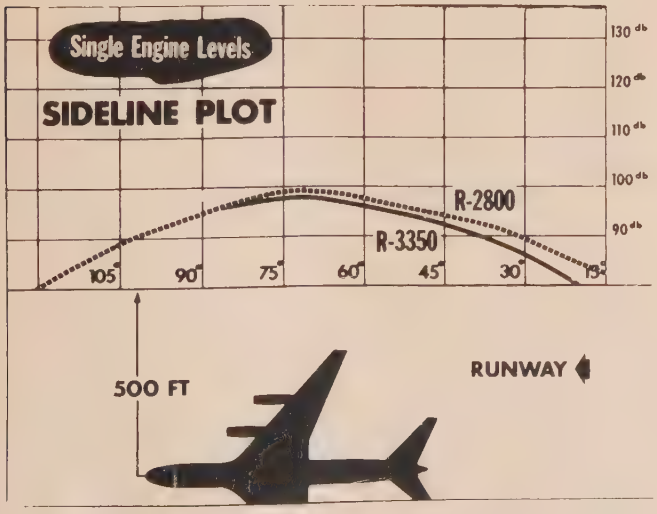


Fig. 11—Sideline plots of overall sound pressure levels of single R-2800 and R-3350 engine calculated from four engine data obtained during airplane take-offs

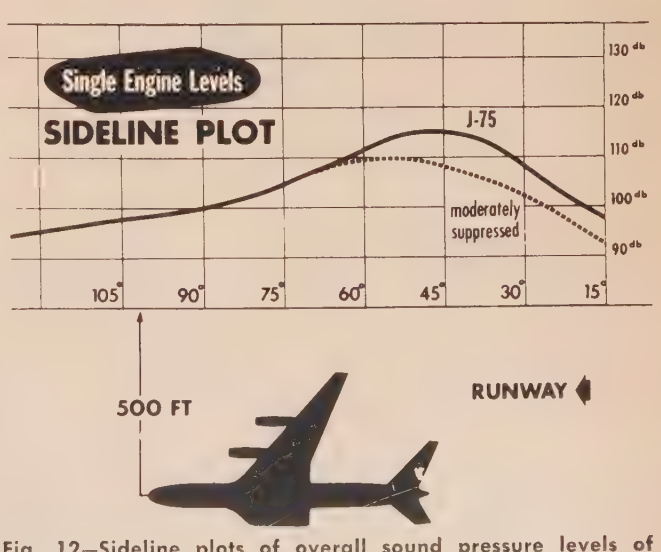


Fig. 12—Sideline plots of overall sound pressure levels of single J-75 engine with and without moderate hypothetical suppressor calculated from twin-engine take-off data from RB-66 airplane

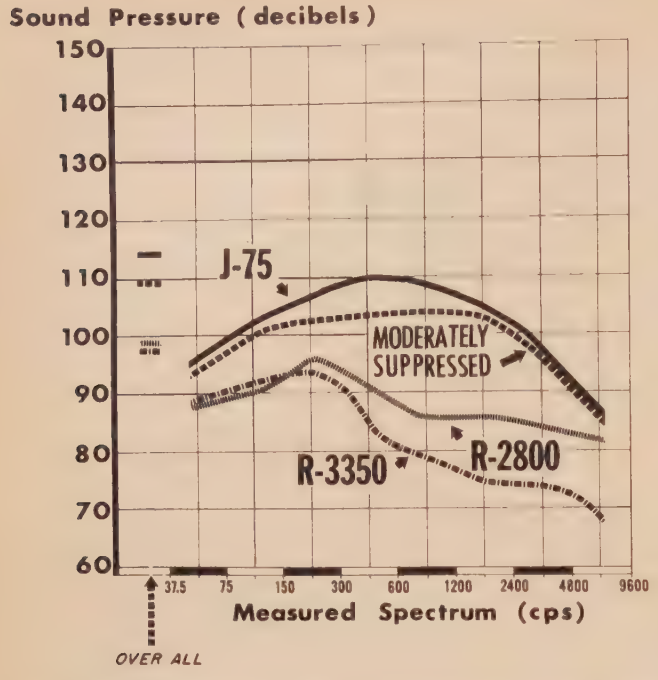


Fig. 13—Octave band analyses for single-engine levels at sideline maximum noise position of initial climbout. R-2800 and R-3350 engine data are calculated from DC-6B and DC-7 take-off data. J-75 data are calculated from RB-66 take-off measurements. Suppressed data assume moderate suppressor of corrugated nozzle type

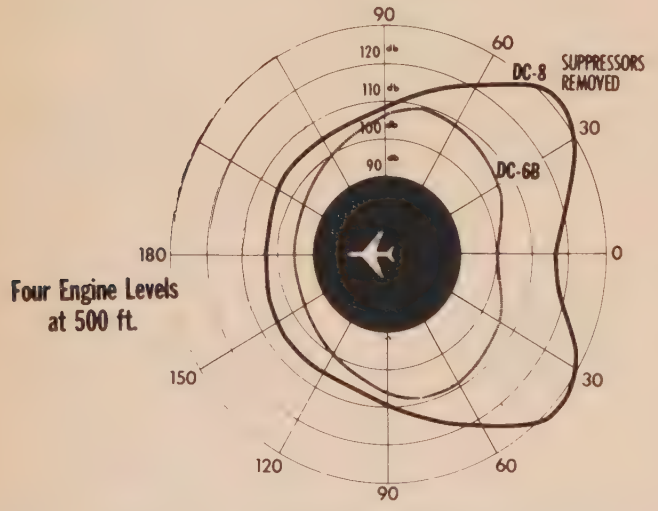


Fig. 14—Polar plot of overall sound pressure levels estimated for DC-8 (J-75 engines) operating at static take-off thrust and measured for DC-6B at static take-off thrust

Fig. 13, on which the reduction of low-frequency sound energy due to motion is evident for both the jet- and propeller-driven airplanes.

A close examination of Figs. 9, 10, 11, and 12 will disclose the reasons behind the "furor" associated with the jet noise suppression program. The critical questions are: (1) how much suppression is needed and (2) how much suppression is obtained. From these data, no clear and unambiguous answer is evident. Nevertheless, several figures of merit may be derived from the curves.

Since the OA SPL has become a widely used number for evaluations, it will be used (although reluctantly) for an example determination of suppression values. The OA SPL difference between the J-75 engine and the R-2800 engine on Fig. 9 is 24 db (at the azimuth of greatest noise of the J-75). The OA difference on Fig. 10 is 22 db. The relevant values for Fig. 10 are taken from polar plots, each in the direction of maximum noise at equal distances from the plane. When the comparison is made as the take-off observer hears it (Figs. 11 and 12), the difference is 17 db instead of 22 db.

Another method of rating a suppressor is to consider the maximum octave band reduction it achieves. On this basis, Fig. 9 rates the hypothetical suppressor at 12 db, the reduction realized in the 300–600-cps octave band. Fig. 10 shows this value as 8 db and Fig. 13 as 7 db. Suppressor evaluation on the basis of the OA SPL gives this suppressor a 7-db rating on Fig. 9, a 6-db rating on Fig. 10, but as the take-off observer hears it, 5 db (Fig. 12).

It is easy to see how confusion of terms and definitions of ratings can permit a 5-db suppressor to be mistakenly shown to supply 12 db of the required reduction in noise.

All previous discussion has been concerned with the sound from single engines. Figs. 14, 15, and 16 are included to show the measured and estimated differences between the respective unsuppressed airplanes with all engines operating at take-off power. Fig. 14 establishes a base line for use in problems involving static operation of the airplane; Fig. 15 establishes a comparable base line for comparison of neighborhood noise during take-offs.

Although the DC-8 airplane will be equipped with noise suppressors, values with the suppressors removed are shown since unsuppressed values afford the most uniform base line for suppressor comparisons.

Whenever the convenience of a single-number evaluation demands it, the author proposes to use the SNL method of single-number evaluation until a better method is suggested and generally accepted. Therefore, maximum SNL values are also shown on Fig. 16.

Proposed Suppressor Rating System

A large share of the discussion so far has been concerned with showing that no single-number system can give a complete picture of the effectiveness of a suppressor. Is there a minimum number of single numbers which can give all the essential information necessary to differentiate suppressor benefits? It is here proposed that five numbers can tell the story of static benefits, and an equivalent five numbers can portray the sideline or neighborhood benefits.

For the static case, there are the polar differences in:

Sound power level (PWL): The reduction in total sound power (an integrated value of the total energy radiated in all directions from the jet engine). This term is a measure of the integrated efficacy of the suppressor.[2]

Sampled noise level (SNL): A term already defined as differentiating in terms of the opinion of an average person as a reactor to noise. The ΔSNL given in the rating should be the SNL difference of maximum energies observed on equal circles around the engines.

Overall sound pressure level (OA): The difference between maximum OA's at a constant distance from the engine. This represents the difference of maximum energies observed on equal circles around the engine.

There are also differences in the following two angles:

ψ: The azimuth of maximum OA on the polar plot of the unsuppressed jet engine.

ϕ: The azimuth of maximum OA on the polar plot of the suppressed jet engine.

For the sideline case, the equivalent numbers become the differences in:

Total fly-by energy (TFE): An integration of the sideline plots, this total expresses the total energy felt at an observer's location during passage of the airplane in flight. The exact definition of this term has been given by K. M. Stevens and L. L. Beranek who have suggested this as a method to account for the integrated effects of the number of flights per unit time from an airfield.[3]

Sampled noise level (SNL): Valuable in rating individual passages of jet engines (airplanes). This SNL rating should be the SNL difference corresponding to the OA SPL difference at points of maximum OA's during passage of the airplane with and without suppressors.

Overall sound pressure level (OA): The difference of the maximum OA's occurring during passage of the airplanes past a point with and without suppressors.

There are also differences in the following angles:

ψ: The azimuth of maximum OA SPL on the sideline plot relative to the position of the unsuppressed airplane at the time this maximum noise was produced.

ϕ: The azimuth of maximum OA SPL on the sideline plot relative to the position of the suppressed airplane at the time this maximum noise was produced. With these definitions in mind, suppressor ratings could be written as polar: PWL, SNL, OA, ψ, ϕ (the static case); and sideline: TFE, SNL, OA, ψ, ϕ (the flight case). In the remainder of this discussion, suppressors will be classified by this method.

Airport Neighborhood

When the Acoustics Section at Douglas was first asked to present information to airline and airport personnel which would permit them to evaluate their neighborhood noise problem, no precedents either as

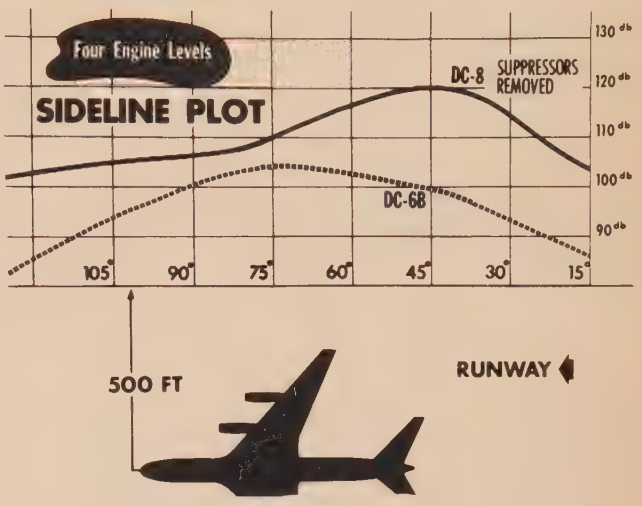


Fig. 15—Sideline plot of overall sound pressure levels compares estimated level of DC-8 with measured data of DC-6B if no suppressors were installed on DC-8

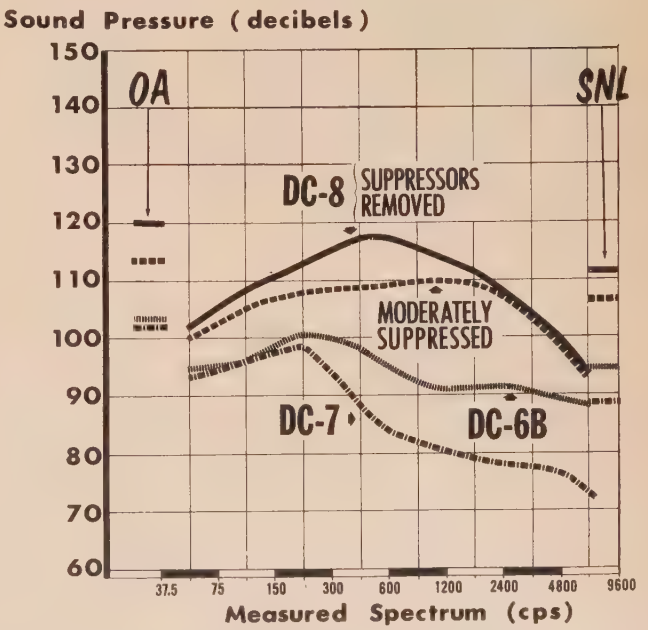


Fig. 16—Octave band analyses for four engine levels at sideline maximum noise position of initial climbout. DC-6B and DC-7 data are from take-off measurements. DC-8 data are calculated from RB-66 take-off measurements. Suppressed data which assume moderate suppressor of corrugated nozzle type presented for illustration purposes only

2 WADC Technical Report 52-204, December, 1952, "Handbook of Acoustical Noise Control, Vol I, Physical Acoustics," by R. H. Bolt, L. L. Beranek, and R. B. Newman.

3 "Effects of Airport Noise on Airport Planning," by K. N. Stevens and L. L. Beranek. Paper presented at IAS National Turbine-Powered Transportation Meeting, San Diego, Aug. 16, 1956.

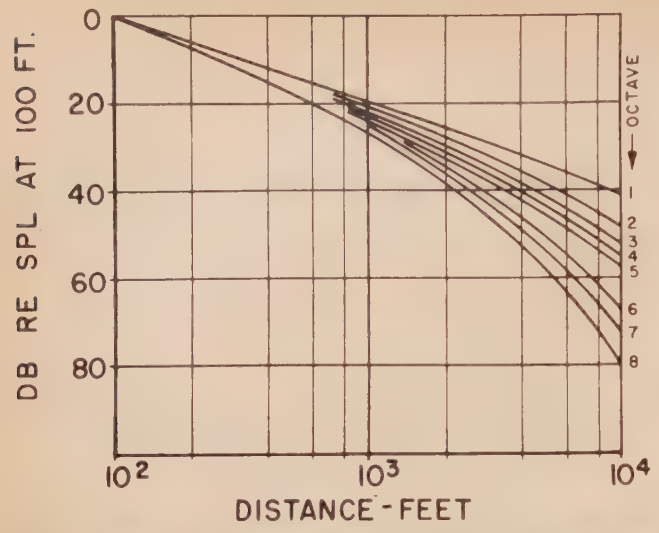


Fig. 17—Octave sound pressure level reductions with distances used in calculations when airplane is in contact with flat terrain

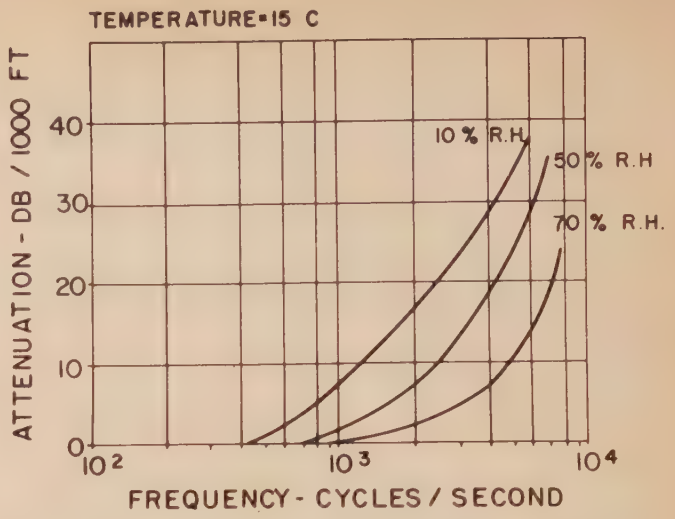


Fig. 18—Sound attenuations used when airplane is airborne to account for effects of relative humidity. Attenuations are added to normal inverse square reductions with distance

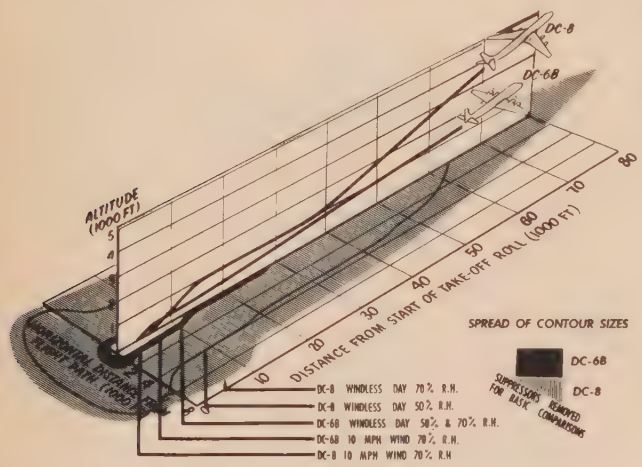


Fig. 19—Annoyance contours of 87-db SNL without suppressors on DC-8

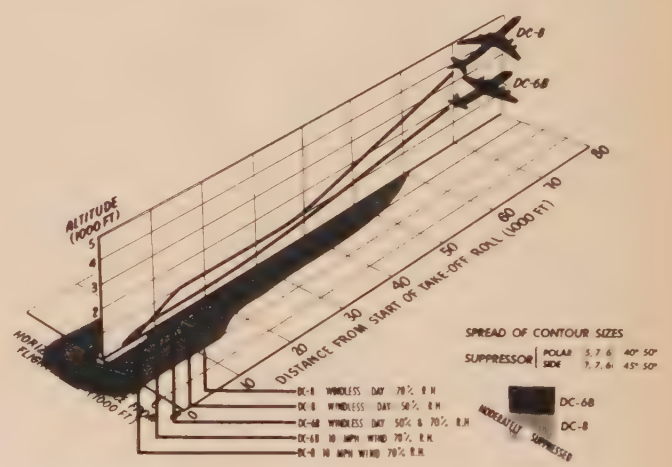


Fig. 20—Annoyance contours of 87-db SNL with one type of moderate suppressors

to criteria or methods of calculation had been established.

First, a criterion was needed. The report of the National Opinion Research Center,[4] based on a series of sample type polls, indicated variations in annoyance versus SNL for people living in airport neighborhoods. The problem was to determine from this a single SNL value which could reasonably be used as a working limit of acceptability. This value was taken as 87 db. Corroboration of this number is obtained from another source [5] which associates a jet-engine SNL of 87 db with a 50% articulation index (AI). This number is conventionally used in aircraft acoustics design as the maximum spectrum for conversational comfort. A thorough discussion of this question [6] presents correlations of SNL, AI, and SIL with actual experience in airport neighborhoods and demonstrates that the choice of 87-db SNL is a sound one.

Pending the results of further studies now in progress, and because of the lack of other valid statistics, we are assuming that an 87-db SNL represents the borderline for acceptable community reaction to the aircraft noise situation.

After establishing a criterion, the next major problem was to calculate the spread of noise from the source. It is well known that wind, turbulence, temperature gradients, and wind gradients in the air can drastically alter the transmission of sound. But how

[4] National Opinion Research Center Report No. 54, University of Chicago, Chicago, "Community Aspects of Aircraft Annoyance," by P. N. Borsky.

[5] *Journal of the Acoustical Society of America*, Vol. 19, January, 1947, pp. 90–119: "Factors Governing Intelligibility of Speech Sounds," by N. R. French and J. C. Steinberg.

[6] WADC Technical Report 52-204, June, 1953, "Handbook of Acoustic Noise Control, Vol. II, Noise and Man," by W. A. Rosenblith, K. N. Stevens, and staff of Bolt Beranek and Newman.

much? A search of the literature revealed nothing useful—and still doesn't. If wind and temperature conditions were assumed which would yield optimistic calculations, there was no restraint or standard to prevent more drastic assumptions leading to still more optimistic predictions.

It was obvious that the determination of wind and temperature effects by a statistical processing of numerous measurements would be an extremely long and tedious chore and one for which equipment and personnel were not available. Therefore it was decided to omit all consideration of wind- and temperature-gradient conditions in standard evaluation procedures until substantial data concerning these effects were acquired. It should be emphasized that actual noise will almost always be less than that calculated without consideration of wind effects, but that under any condition chosen it cannot honestly be said exactly how much less.

Other phenomena which must be taken into consideration are the effects of temperature and humidity on the propagation of sound through the atmosphere and the effects of terrain on the spread of sound. Although the acoustical literature failed to reveal universally accepted values, the curves of Figure 17 [7] are applied as corrections to the airplane noise contours when the airplane is on the runway. This accounts for the effect of flat terrain. The curves of Figure 18 [8] are applied to the noise data when the airplane is airborne, to account for attenuation in the atmosphere.

To arrive at the annoyance contours of Figs. 19, 20, 21, and 22, initial use was made of the reference polar plot of noise for the static condition. The 87-db SNL in the direction of maximum OA SPL, using the ground attenuation curves of Fig. 17, was calculated. After the airplane was airborne, the reference sideline plot was used to calculate the 87-db SNL variation with distance according to the atmospheric attenuation curves of Fig. 18. Further out in the flight path, corrections were made for power reductions and speed increases, using a $\rho A V^8$ correction. Finally all points were connected together to form a smooth contour.

Neighborhood Contours

Figs. 20, 21, and 22 compare 87-db SNL contours for the DC-6B and for the DC-8 equipped with various suppressors. For comparison purposes Fig. 19 shows contours with suppressors removed from the DC-8.

The outer boundary of the DC-6B shaded area represents its 87-db SNL at a temperature of 15 C and 50% (and also 70%) relative humidity. This is based on a series of measurements made when there was virtually no wind. The outer boundary of the DC-8 shaded area is the 70% relative humidity, 15 C contour, calculated from flight data obtained on A3D and RB-66 airplanes, again under virtually no wind conditions. The single heavy contour is a calculated DC-8 curve for 50% relative humidity and 15 C. It is the standard curve used for evaluation purposes.

The contours of Figs. 19, 20, 21, and 22 also show the marked effect of one set of weather conditions on sound transmission from aircraft, especially on the noise from jets. The inner boundaries of both the DC-8 and DC-6B result from measurements made on a gusty day, 70% relative humidity, 20 C, with average winds slightly exceeding 10 mph. The most noticeable effect of the wind was to reduce low-frequency sounds (below 300 cps) as much as 15 db. Jet noise between 300 and 600 cps was reduced

[7] Unpublished curves depicting measured data by R. H. Bolt, L. L. Beranek, and staff of Bolt Beranek and Newman.

[8] WADC Technical Report 54-602, May, 1955, "Review of Sound Propagation in Lower Atmosphere," by W. L. Nyborg and D. Mintzer.

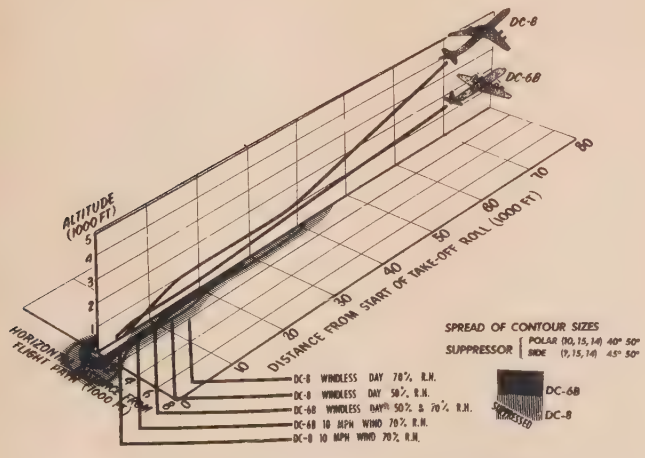


Fig. 21—Annoyance contours of 87-db SNL with suppressors which reduce noise level

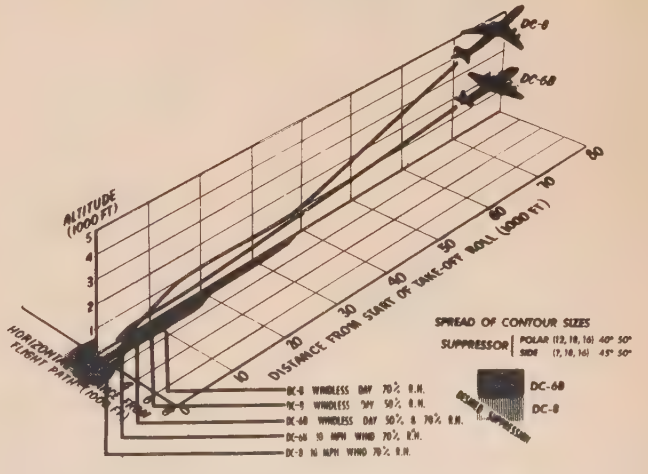


Fig. 22—Annoyance contours of 87-db SNL with desired suppressors showing further reduction in noise level

considerably more than was propeller noise. This indicates that under such wind conditions the jet airplane, even unsuppressed, produces a noise comparable to that of modern propeller-driven airplanes.

Since it is during hot, humid, still-air weather conditions that the spread of sound is greatest, one should be cautious about basing estimates on statistical data taken around an airfield where average wind conditions may exceed 5 or 10 mph.

Fig. 23 offers a single look at the large changes in relative areas enclosed by the 87-db SNL contours which can occur under windy conditions and when suppressors are installed. Since this one condition of wind and its observed acoustic effects account for noise reductions equivalent to large amounts of mechanical suppression, it is imperative that a complete study of the effects of weather conditions be made. When such a study is made, analyses of each individual airport's noise problem based on statistical weather conditions can be completed. Such analyses will probably indicate that there will be only a few days a year when the desired suppressions noted in this paper will be necessary. If such suppression is accomplished, the jet transports will be quieter than modern propeller transports over the remaining larger portion of the year.

Fig. 22 shows that a "polar (12, 18, 16) 40 deg 50 deg" suppressor gives J-75 engine noise levels comparable to the DC-6B, even with the extremely conservative still-air and humidity conditions assumed in the calculation. If the airplane is equipped with Conway bypass engines, a "polar (9, 15, 13) 40 deg 50 deg" suppressor will accomplish the same desirable result.

Now that methods of suppression evaluation have been defined, a look at methods of suppression is in order.

Fig. 23—Simplified areas enclosed by 87-db SNL contours which can occur under windy conditions and when suppressors are installed

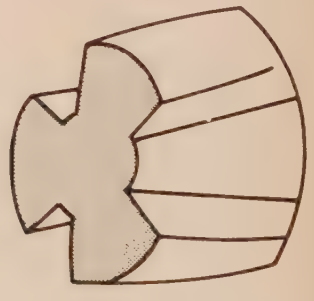
Noise Suppressors

Three fundamental methods of suppressing the irritating properties of normal jet noise have emerged from basic studies of the jet noise problem. They are to:

1. Shift frequencies: A method first proposed at the Pratt & Whitney Aircraft Corp., it utilizes the fact that the predominant sound energy produced by a jet shifts to higher frequencies as the dimensions of the jet are reduced. The resulting increased effective noise reduction occurs because of the greater loss of high-frequency sound energy in passing through the atmosphere; the frequency shift also takes advantage of the decrease in sensitivity of the human ear at frequencies above 4000 cps.

2. Alter jet mixing: Researchers at the University of Southampton College of Aeronautics and Engineers at Rolls Royce found that devices attached at the jet-engine nozzle to encourage rapid early mixing of the jet stream with the outside air also caused an appreciable noise reduction; the frequency at which the greatest reduction occurs can be governed by the geometry of the mixer.

3. Reduce velocities: This principle was evident as soon as it was shown that the noise of a jet was proportional to a very high power of the jet exhaust

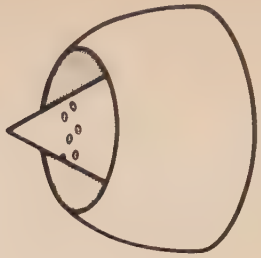


SCALE	PWL	SNL	O.A.	Ψ°	Φ°	F_n%
1/10	—	4.5	6.0	30	35	-0.6

Fig. 24—Suppressor rating typical of corrugated nozzle suppressors of approximately half-depth corrugations

SCALE	PWL	SNL	O.A.	Ψ°	Φ°	F_n%
1/10	—	2.5	3.5	30	32	-1.7

Fig. 25—Suppressor rating typical of circumferential vortex generators



	SCALE	PWL	SNL	O.A.	Ψ°	Φ°	F_n%
AIR	1/10	—	.5	2.0	30	30	—
WATER	1/10	—	2.5	3.5	30	30	-8

Fig. 26—Suppressor ratings typical of air and water jets injected into engine exhaust gases from bullet body of bulleted nozzle

velocity (ρAV^8). When the jet velocity can be reduced and the thrust maintained by increasing the density of the exhaust gases or the area of the exhaust, a substantial noise decrease is accomplished.

An extreme form of the frequency shift is represented by the long tail pipe developed by Pratt & Whitney which contains many thousands of small holes. In this form, it is difficult to visualize the device as a practical flight suppressor, although it has been developed into an economical and efficient ground suppressor.

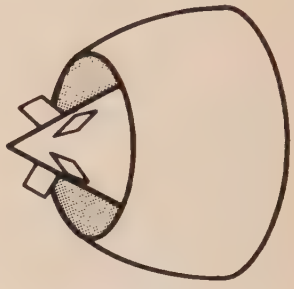
The original form of the jet mixing devices was presented by Rolls Royce as the well-known corrugated nozzle. Such a device accomplishes a worthwhile noise reduction but will have to be developed further so as not to penalize thrust performance at cruise conditions.

The bypass engine is a form of a velocity-reducing device but the engine would have to be inordinately large if it supplied enough cold secondary air (to change ρAV^8 enough) to obtain the desired noise reduction. The ejector is another form of velocity-reducing device, but when used with the plain conical jet nozzle it may have to be extremely long to mix the jet and induced air properly in order to provide the required ρAV^8 change at the ejector exit.

With these simple elements the Jet Noise Suppression Age began its mechanical development.

When engineering ingenuity was applied to these elements, new forms appeared. The elements were combined to yield increased noise suppression. To the author's knowledge, only pairs of suppression elements have been combined at the present time, but it is his belief that the best noise suppressor may result from an optimum combination of all three principles or some new principle yet to be discovered.

The Douglas Aircraft Co. assigned a team of aerodynamicists, powerplant engineers and acoustics engineers to the noise suppression task. They decided



SCALE	PWL	SNL	O.A.	Ψ°	Φ°	F_n%
1/10	—	3.5	5.0	30	32	-2.5

Fig. 27—Suppressor rating typical of radial vortex generators. Paired vanes have opposite aerodynamic lift

at the beginning of the program to develop first the mechanically most simple type of suppressor to have as a "bird in the hand." Moreover, it was felt that the device should, while realizing the required noise suppression, cause a minimum thrust loss at take-off and no change at all in the cruise performance of the airplane.

Of the three basic principles, a change of jet mixing seemed the most promising. Beginning with the corrugated nozzle for comparison, tests were made of many types of mixing devices. These were all capable of being turned off or retracted in cruise to prevent the same small loss of thrust in cruise which may occur during take-off operations.

Basic elements of mixing nozzle suppressors and the results of tests of these elements are shown in Figs. 24 through 29. SNL's for scale models have been obtained by normalizing the scale-model spectra to the spectra of full-size nozzles.

Further tests disclosed that these configurations

SCALE	PWL	SNL	O.A.	Ψ°	Φ°	F_n%
1/10	—	3.5	5.0	30	35	-3.6

Fig. 28—Suppressor rating typical of swirl vanes. All vanes have aerodynamic lift in same radial direction

SCALE	PWL	SNL	O.A.	Ψ°	Φ°	F_n%
1/5	1.7	2.2	2.2	40	45	0
FULL	2.1	4.1	2.5	40	40	-0.4

Fig. 29—Suppressor rating typical of scalloped nozzles. Note that this is not a toothed nozzle

could sometimes be combined with additive noise suppression results. Fig. 30 shows photographs of a full-scale nozzle combining radial vortex generators and a scalloped nozzle. Data in the table at the bottom of the figure show the correlation between the full-scale nozzle and its 1/5-scale test model. Recent tests of more advanced combinations indicate that a "polar (6, 9, 10) ? deg ? deg" suppressor is probable utilizing these simple mixing-type devices.

As a part of the simple suppressor development program, Douglas has investigated the possibilities of using an ejector to reduce noise. Fortunately, the ejector principle and the mixing principle were found to supplement each other. The mixing devices applied to the primary nozzle of the engine improve the noise suppression properties of the ejector so that it can be shortened to a more practical length and still produce very significant noise reductions.

Examples of data from ejector tests are shown on Fig. 31. Full-scale data were obtained from F. B. Greatrex of Rolls Royce, who fed the exhaust (with six corrugation primary) from an Avon engine into a 93-in.-long ejector to obtain the results shown. Note how the corrugated primary nozzle improved the noise reduction.

The ejector data included in the bottom row of Fig. 31 were obtained from tests of a Douglas ejector, two diameters long, 1.6 diameters across, and with an optimum mixing primary nozzle. It is clear from this that noise suppressions of the magnitude shown in this figure are very close to the suppressions desired on future jet transports.

This paper has dealt with noise suppression devices without particular regard to the mechanical and aerodynamic considerations which must be satisfied to produce the final installation. A number of additional suppressor ideas are being experimented with by Douglas and other companies. It is too early to tell whether the performance of these new ideas will exceed the eventual capabilities of those already tested, or be more adaptable to practical installations. It is certain, however, that good progress is being made, and that with coordinated and cooperative effort on the part of the many agencies engaged in this problem, a timely, satisfactory solution will be found.

DISCUSSION

Wants Better Method For Measuring Noise

—F. Mintz
Lockheed Aircraft Corp.

MR. MILLER is entirely correct in emphasizing that a multitude of numbers is required to define a noise problem adequately and in deploring the insistence of the layman in acoustics on trying to characterize such a problem by one number, preferably in decibels. He correctly points out, also, that the so-called speech interference level (SIL), the arithmetic average of the decibel levels of the fifth, sixth, and seventh octaves, is not an adequate measure of a sound spectrum, since it ignores the low-frequency content entirely and that the overall sound level in decibels can be very misleading since it gives no clue at all as to frequency content.

He suggests, as an alternative approach, for a reasonable single-number description, the use of the sample noise level (SNL) which is the arithmetic average of the levels in decibels in the second, fourth, and sixth octaves.

One of the reasons for the popularity of the SIL and for the proposed popularity of the SNL is that by arithmetically averaging the levels in decibels for several bands one avoids the assignment of excessive weight to one band which may be considerably higher than the others. Whatever this may mean objectively, it does seem to provide a more satisfactory subjective estimate of the nature of a sound (if any one number can be considered to do this). There are many valid objections, however, to the attempt to characterize a sound by using only selected octave bands. Consequently, if there is indeed any validity to the use of arithmetic averaging of octave band levels, the most sensible way to do this would be to take the numerical average of the levels of all eight octave bands, not merely of a selected three.

An immediate obvious objection to the use of a simple aver-

age of all band levels is that certain bands are more significant in terms of subjective effect (loudness, annoyance, and the like) than others. The immediate modification that suggests itself is to use a weighted average. It requires little insight to raise the next question, "How does one weight the bands?" The most rational answer seems to be, according to loudness. And this represents no great obstacle. There is available in the literature a simple formula such as that of Stevens [a] for determining loudness from octave-band data. By the use of this formula and/or some simple chart (as for example, the Mintz-Tyzzer chart [b]) a loudness value can be obtained quite readily from octave-band sound levels.

If, therefore, it is considered necessary to use single numbers to characterize a noise for purposes of discussion or comparison or simply to provide a basis of understanding for the layman, the two alternative methods of converting octave-band levels to a single descriptive number (particularly the loudness method) seem to merit consideration as substitutes for or as supplements to the SIL and SNL. The trial use of these numbers in some typical aircraft noise problems is suggested for evaluation purposes.

ORAL DISCUSSION

Reported by F. Mintz

Lockheed Aircraft Corp.

J. T. Dyment, Trans-Canada Air Lines: Is Convair working along the same lines as Boeing with respect to jet noise suppression?

G. Getline, Convair: Convair is exploring all significant aspects of the problem.

H. W. Nevin, Jr., United Air Lines, Inc: Are there any results available on the recent measurements in the San Francisco area on jet taxiing noise?

Mr. Miller: Detailed results are not available, but the general impression is favorable. In the tests, an A3D was used without noise suppressors to simulate operation of a DC-8. Inside the terminal, no appreciable disturbance was observed when the doors and windows were closed.

R. K. Rourke, Trans World Airlines, Inc.: Have any measurements of jet noise been made at elevated temperature and humidity conditions?

Mr. Miller: The effect of temperature on noise is small. Winds have been found to have a surprisingly large effect, but no explanation has been found for this. Relative humidity is an important factor in jet noise, and the greatest effect of temperature is probably due to changes in relative humidity. In any case, it cannot be emphasized too strongly that the noise problem cannot be presented or summarized adequately by a single-number quantity.

H. D. Hoekstra, CAA: Could you comment further on the fact that one of the contributors to gas-turbine noise is the compressor, as for example, the obvious compressor noise associated with the Viscount.

Mr. Miller: The problem of compressor noise can be handled by present conventional means, such as sound barriers, and ear protectors for working personnel. It is not considered as difficult a problem to solve in practice as tail pipe noise.

R. D. Kelly, United Airlines, Inc.: Is there any good physical explanation of the striking effect of wind in reducing jet-engine noise disturbance on the ground?

Mr. Miller: No good explanation is yet available for the observed effect of wind on jet noise. Some work by Ingard indicates that air turbulence diffuses noise.

Mr. Kelly: Isn't the speech interference level very important?

Mr. Miller: It is very important. However, it ignores the low-frequency range which is also important. The sample noise level gives a better total picture, since it also includes the low frequencies.

Mr. Kelly: If the "ejector" type of unit is as effective as it appears, wouldn't increasing the tail pipe outlet diameter have the same effect?

Mr. Miller: The variable tail pipe would not provide the same effect as the ejector, which induces flow of external air into the jet exhaust, thus decreasing effective exhaust velocity without loss of momentum.

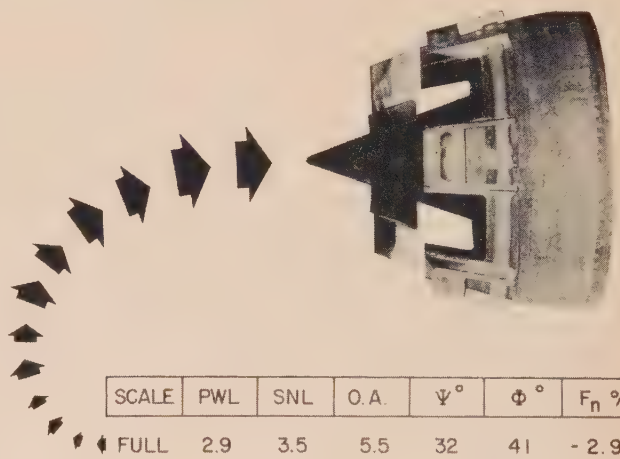
Mr. Kelly: Would not the use of suitable holes in the structure, as in acoustical materials, help in reducing the noise?

Mr. Miller: If upstream turbulence in the engine affected the noise, then the use of holes (resonators and the like) might be useful. This does not, however, seem to be the case, as most of the noise is generated downstream in the exhaust. Such treatment might be useful in reducing compressor noise ahead of the engine.

Author's Closure To Discussion

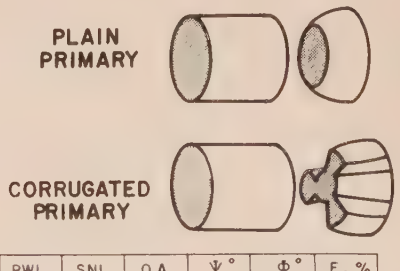
I concur wholeheartedly with the comments of Mr. Mintz so long as these comments are directed toward future investigations and future methods of presenting data. Neighborhood reactions should be correlated with both the loudness functions and the integrated sideline energy.

The principal reasons for my proposal to use the SNL in present evaluations are (1) it does partially account for the whole spectrum, but, even more importantly (2) it is the only method to date which has been correlated extensively with neighborhood reactions.[4]



SCALE	PWL	SNL	O.A.	Ψ°	Φ°	F_n %
FULL	2.9	3.5	5.5	32	41	-2.9
1/5	3.2	4.8	5.2	45	45	-2.7

Fig. 30—Suppressor ratings typical of paired combination of mixing devices, full and 1/5 scales



	SCALE	PWL	SNL	O.A.	Ψ°	Φ°	F_n %	
PLAIN	1/10	—	4.5	5.0	30	60	+11.6	
CORRUGATED	1/10	—	10.8	11.0	30	60	+13.8	
PLAIN	FULL	6.0	9.0	8.0	40	40	—	
CORRUGATED	FULL	12.0	14.6	14.0	40	70	—	SIX LOBE PRIMARY
BEST COMBINATION	1/10	—	16.0	14.0	30	70	+2.8	

Fig. 31—Suppressor ratings typical of ejectors with plain and corrugated primary nozzles, full and 1/10 scales

[a] *Journal of the Acoustical Society of America*, Vol. 28, September, 1956: "Calculation of Loudness of Complex Noise," by S. S. Stevens.

[b] *Journal of the Acoustical Society of America*, Vol. 24, January, 1952, pp. 80–82: "Loudness Chart for Octave-Band Data on Complex Sounds," by F. Mintz and F. G Tyzzer.

Diesel Exhaust Odor — Its Evaluation and

WHEN the diesel engine was first introduced in urban motor coaches and trucks, a few whiffs told the public there was something new under the sun. What people were smelling was the safest exhaust gas from any internal-combustion engine then in existence, but still they did not like the odor. Even today many people do not like diesel exhaust odor, but it is usually tolerated because of the economy and durability of diesel engines. In some applications, however, the odor has restricted use of diesel engines. For this and other reasons, the problem of diesel odor is being investigated actively by several groups in General Motors. This paper describes work that has been done by the Research Staff on developing techniques for evaluating diesel odor. Also included are limited data showing how these techniques have been used in the search for ways to reduce or eliminate diesel odor.

TECHNIQUES, based on panel estimates, were developed for evaluating the odor and irritation intensities of undiluted diesel-engine exhaust gases or of various dilutions of these gases in air. Along with the estimates, chemical analyses were made to determine the concentrations of total aldehydes, formaldehyde, and oxides of nitrogen.

Statistically significant correlations were found between odor or irritation intensity estimates and the analytical data, but these correlations were too weak to permit accurate prediction of odor or irritation from chemical analyses.

Effects of some engine variables on diesel odor were studied. Possible means of reducing diesel odor are discussed.

Many people fail to distinguish between exhaust odor and air pollution. As a result the diesel engine has been accused of contributing more to air pollution than other types of powerplants whose exhaust gases have less recognizable odors. In fact, it has even been suggested that converting all gasoline and diesel engines to operation on liquified petroleum gas might significantly decrease air pollution in some major cities.[1,2] The merit of such a suggestion is best judged on the basis of factual information such as that obtained in a recent test program in which exhaust gases from representative Chicago Transit Authority coaches powered by comparable gasoline, propane, and diesel engines [3] were analyzed.

The results of the Chicago investigation, summaized in Table 1, showed that (a) the emission of

[1] Air Pollution Foundation Report No. 12, Vol. 1, No. 12, November, 1955, "Second Technical Progress Report."
[2] *National Petroleum News*, Vol. 46, March, 1954, pp. 18–21: "How Oil Industry Fights Smear of Its Good Name."
[3] *APCA Journal*, Vol. 5, August, 1955: "Composition of Exhaust Gases from Diesel, Gasoline, and Propane-Powered Motor Coaches," by M. A. Elliott, G. J. Nebel, and F. G. Rounds.

Relation to Exhaust Gas Composition

F. G. Rounds and H. W. Pearsall,
General Motors Corp.

This paper was presented at the SAE National Diesel Engine Meeting, Chicago, Nov. 2, 1956.

toxic carbon monoxide from the diesel coaches was only a fraction of that from the gasoline and propane coaches and (b) the observed differences in the emission of oxides of nitrogen, formaldehyde, and hydrocarbons by the three engine types were relatively small. These and other available data, therefore, do not indicate that wholesale conversion of gasoline and diesel engines to liquified petroleum gas operation would help appreciably to control the air pollution problem. No formal measurements of odor were made during the Chicago investigation, but incidental observations of the exhaust odors of the three types of engines indicated that all were objectionable to some degree. The propane-powered coaches were generally the least objectionable whereas either the diesel or gasoline-powered coaches were the most objectionable depending upon the driving condition. Hence, the Chicago results suggest that there is no close relationship between the exhaust odor of an engine and its contribution to air pollution.

Odor And Its Measurement—No physical or chemical methods have yet been found to "measure" odor, and consequently the primary standard in all odor work remains the human nose.[4] When an individual sniffs a given odor sample, he integrates several independent factors: (a) the odor quality (whether they are such odors as flowery, pungent, or acrid), (b) the odor intensity (whether strong, easily noticeable, faint, or other characteristics), (c) the irritation quality (whether the nose, throat, and/or eyes are affected), and (d) the irritation intensity (whether strong, easily noticeable, faint, or the like). Since conscious evaluation of all these factors in a test program may lead to confusion, most odor studies are confined to one or at most two of these factors.

The many techniques described in the literature for

Table 1—Composition of Exhaust Gases from Diesel, Gasoline, and Propane-Powered Motor Coaches

Driving Condition	Carbon Monoxide, vol %	Oxides of Nitrogen, ppm	Formaldehyde, ppm	Hydrocarbons, ppm
	Observed Concentrations of Exhaust Gas Constituents			
Idle				
Diesel	0.0	59	9	390
Gasoline	11.7	33	30	4830
Propane	5.1	47	30	2410
Acceleration				
Diesel	0.05	849	17	210
Gasoline	3.0	1347	16	960
Propane	3.5	1290	18	390
Cruise				
Diesel	0.0	237	11	90
Gasoline	3.4	653	7	320
Propane	1.75	2052	23	330
Deceleration				
Diesel	0.0	30	29	330
Gasoline	5.5	18	286	16,750
Propane	4.2	56	172	19,030

Total Hourly Emission Based on the Chicago Transit Driving Pattern

Type of Coach	Carbon Monoxide, SCFH	Oxides of Nitrogen, SCFH	Formaldehyde, SCFH	Hydrocarbons, lb per hr
Diesel	2	5.4	0.24	0.47
Gasoline	194	3.8	0.17	1.24
Propane	96	4.9	0.11	0.57

4 "Odors, Physiology and Control," by C. P. McCord and W. N. Witheridge. Pub. by McGraw-Hill Book Co., New York, 1949.

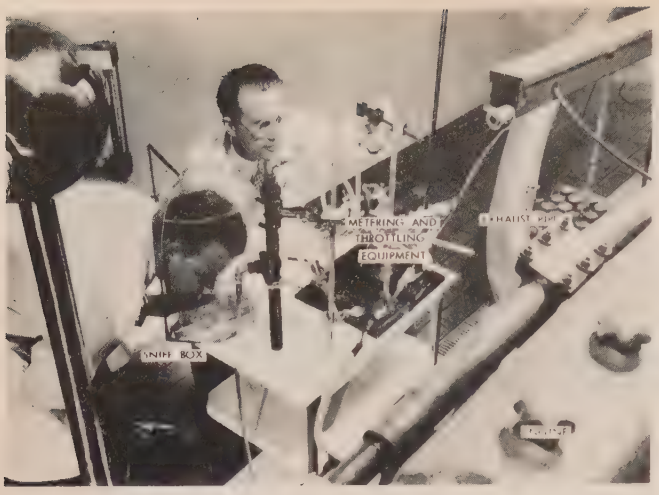


Fig. 1—Equipment used to evaluate diesel odor intensity by dilution technique

evaluating odor fall into three categories: threshold techniques,[5,6] direct techniques,[4,7] and chemical techniques.[4] The characteristic feature of the threshold methods is that the odorant is evaluated at a series of dilutions in a neutral gas (such as air) until a concentration is found at which the odorant is just detectible. This is the "odor threshold." In the performance of the tests the odorant may in some cases be dissolved in a neutral liquid such as water. Odor thresholds for some materials may be as low as 1 ppm (by volume) or even less. The relative odor intensities of odorous materials such as exhaust gases may be estimated by comparing their odor thresholds. The direct technique requires the evaluation of undiluted odorant or some relatively high concentration of the odorant in a neutral diluent gas. (This technique is used extensively in odor quality work as well as in odor intensity studies.) The odor intensities of samples are compared on the basis of intensity estimates made at the same concentration. The chemical analysis technique depends upon measuring the concentration of either the true odorant or some material whose concentration varies in a known relation with the concentration of the odorant. However, before the chemical analysis technique can be used, it must be calibrated by one of the other two techniques. Thereafter, all that is needed to compare the intensities of samples is a knowledge of the concentrations of the constituent in the samples.

All three techniques require that the human nose be used either as the test instrument or as the calibration medium. Although the average person can detect extremely small amounts of some odorous materials, the psychological reaction to that material will vary appreciably from one individual to another. Moreover, the reaction of a given person varies from time to time. In an attempt to compensate for these fluctuations of an individual with time, most odor investigators rely upon a team of individuals or "panel." The rating of the panel on an odor sample can be arrived at by having each member of the panel sniff the sample individually and then averaging the independent ratings of the panelists.

Test Equipment and Procedure

All three general approaches to the problem of odor measurement were used in our studies. Evaluation of odor quality, as distinguished from odor intensity, is difficult and probably belongs more in the realm of the aesthete than the engineer. Moreover, it is often found that differences in odor quality are unimportant if the intensity is low enough. Therefore, the present studies were designed to evaluate merely the intensity of the exhaust gas odor and irritation. Rather than determining the true odor thresholds, it was found more convenient to evaluate a series of dilutions of exhaust gas in air with a panel to measure the reduction of the odor or irritation of diesel exhaust gas with increasing dilution. When the direct technique was used, the panel made its estimates on undiluted exhaust gas. In our study of the chemical analysis technique, the exhaust gas constituents considered were total aldehydes, formaldehyde, and oxides of nitrogen (except N_2O). Most of the odor and irritation intensity evaluations were made on undiluted exhaust gas.

The samples for odor and irritation intensity evaluation were taken at a tee in the 4-in. exhaust pipe about 16 ft from the exhaust manifold. From the tee, the sample passed through 4 ft of $\frac{7}{16}$-in. copper tubing, a water-cooled exchanger, 10 in. of $\frac{7}{16}$-in. Tygon tubing, and finally entered the bottom of a so-called "sniff box" through a metal diffuser plate. In all tests, the flow of exhaust gas was controlled by a restriction between the heat exchanger and the sniff box.

The sniff box, shown in Fig. 1, was a clear plastic box of 0.5-cu ft capacity. The box was so designed that a panelist could insert his head comfortably for an odor evaluation whereas stray air currents would not easily enter. Clear plastic was used because it was easy to clean and it partially relieved the psychological fear of putting one's head in a dark box. Between evaluations by individual panelists, the loose-fitting door was closed, and the box was flushed with a continuous stream of gas to renew the test sample.

In order to obtain maximum repeatability of the odor and irritation intensity estimates, the following precautions were observed:

1. Small-diameter tubes and passages through which the exhaust gas passed were made as short as possible.

2. The sniff box and associated equipment were cleaned frequently.

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[5] *Chemical and Engineering News*, Vol. 34, June, 1956, pp. 2794–2796: "Odor Audit and Control."
[6] *Tappi*, Vol. 36, April, 1953, pp. 154–161: "Abatement of Sulphate Pulp Mill Malodors by Odor-Masking Agents," by W. C. Meuly and B. K. Tremaine.
[7] *The Lamp*, Vol. 38, Spring Issue, 1956, p. 24: "The Sniffers."

Table 2—Dynamometer-Mounted Diesel Engines Used in Odor Studies

Engine	Type	Displacement, cu in.	Number of Cylinders	Nominal Compression Ratio	Combustion—Chamber Design	Service Life
A	2-cycle	426	6	16.0	Open chamber	300 hr on dynamometer stand
B	2-cycle	426	6	16.0	Open chamber	About 300,000 miles total. About 100,000 miles since overhaul
C	4-cycle	673	6	16.6	Open chamber	New engine after run on dynamometer
D	4-cycle	505	6	17.0	Open chamber[a]	About 200 hr on dynamometer stand

[a] The injected fuel impinges on the piston at about a 30-deg angle.

3. The temperature of the gases entering the sniff box was kept between 140 and 160 F by applying external heat if necessary.

4. The gas flow rate into the sniff box was maintained at about 1.0 to 1.5 cfm.

5. The membership of the odor panel was not changed during a given series of tests.

For the chemical analyses the exhaust gas was sampled after it had passed through 16 ft of 4-in. exhaust pipe and 5 ft of ¼-in. copper tubing. The samples were obtained in evacuated 500-ml bottles containing appropriate absorbing liquids. The oxides of nitrogen (except N_2O) were determined by the phenoldisulfonic acid method of Beatty, Berger, and Schrenk.[8] Total aldehydes were determined by the acid Schiff's reagent method of Busch and Berger.[9] Formaldehyde was determined by the phenyl hydrazine-ferricyanide method described by Kersey, Maddock, and Johnson,[10] with modifications to improve the precision. All three analysis methods are colorimetric. The phenoldisulfonic acid method is accurate and reliable for oxides of nitrogen except N_2O. The formaldehyde determination may be affected to a small extent by higher aldehydes. In the case of the total aldehyde analyses, there is some question as to the completeness of response for certain higher aldehydes. Although the selected aldehyde methods may not be completely reliable, they are considered at least as good as other available methods for aldehydes, and we had had previous experience with them. Nevertheless, further development of analytical methods for aldehydes is needed.

The 2-cycle and 4-cycle diesel engines used in the study are described briefly in Table 2. All were tested on the dynamometer stand. Engine B, previously used in an intercity bus, had been removed because of high oil consumption and excessive smoke. The other engines were relatively new at the time of the test.

Engine speeds from 400 to 2000 rpm and engine loads from no load to full load were investigated. Fractional loads as mentioned in this paper refer to fractions of the full throttle brake horsepower at the stated engine rpm. Prior to sampling of the exhaust gas at a particular test condition, the engine was stabilized by operation at that condition for 30 min.

Panel—The most important measuring tool used in these studies was the rating panel. After an engine test condition had been established, each member of the panel took one or two sniffs of the gas in the sniff box and independently recorded his impression of the odor and irritation intensity of the sample. As a guide for the panelists, the descriptive terms of Table 3 were used.

After a little practice, many of the panelists chose to report their estimates to the nearest 0.5 unit and were encouraged to do so. The ratings of the individual panelists were given numerical weights as shown in the table, and these were averaged to obtain the final value.

When an individual rates the intensity of an odor, he compares it with his own mental concept of what is meant by such terms as "strong" and "easily noticeable." These mental standards undoubtedly vary from one person to the next. In addition, many factors such as the odor quality, the engine noise, and the individual's health might also affect his subjective standards. Although attempts were made to prepare reproducible odor standards by vaporizing known amounts of pure chemicals in air, the resulting odors were so unlike diesel odor as to be useless. The idea of obtaining a "constant" supply of exhaust gas from a second diesel engine was abandoned because of inability to ensure that the "standard odor" would remain constant. Therefore, it was necessary to depend upon the panelist's mental definitions of the odor and irritation intensity levels and to rely upon the averaging of several ratings to smooth out random variations.

In the selection of a panel, there are two important considerations, the size of the panel and its membership. For the most reproducible results, a very large panel would probably be best. For convenience and practicability, a small panel is desirable. For our study, the optimum number appeared to be about five, and this size panel was used in most of our tests.

Our panelists included dynamometer operators, engineers, and secretaries. However, not all people

Table 3—Odor and Irritation Intensity Descriptions

Odor Intensity		Irritation Intensity	
Definition	Odor Units	Definition	Irritation Units
Very strong	5	Strong	4
Strong	4	Easily noticeable	3
Easily noticeable	3	Faint	2
Faint	2	Very faint	1
Very faint	1	No irritation	0
No odor	0		

8 Bureau of Mines Report of Investigations No. 3687, 1943, "Determination of Oxides of Nitrogen by Phenoldisulfonic Acid Method," by R. L. Beatty, L. B. Berger, and H. H. Schrenk.

9 Bureau of Mines Report of Investigations No. 4531, 1949, "Sampling and Determination of Aldehydes in Diesel-Engine Exhaust Gas and in Mine Air," by H. W. Busch and L. B. Berger.

10 *The Analyst*, Vol. 65, April, 1940, pp. 203–206: "Determination of Small Amounts of Formaldehyde in Air," by R. W. Kersey, J. R. Maddocks, and T. E. Johnson.

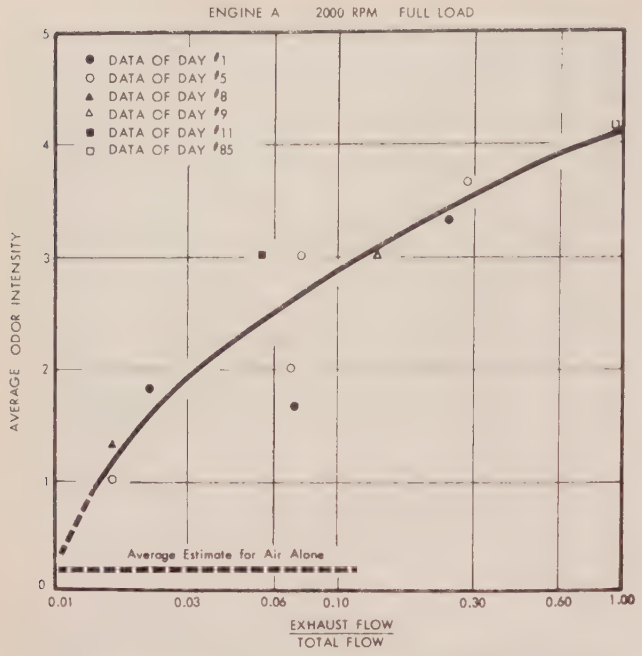

Fig. 2—Reproducibility of odor data obtained by dilution technique

were equally desirable as panelists. A few tended to rate all samples at one end of the scale or the other and some tended to rate the samples erratically compared to their own previous estimates. Such individuals were not asked to serve on the panel. Except for about one person in ten whose ratings were erratic, everyone tested seemed to be able to make consistent ratings. The age or sex of the panelists appeared to be unimportant factors.

Once the panel was selected, its membership was not changed during a given series of tests. When the membership changed from one series of tests to another, the average estimates for a given engine condition sometimes differed, but the trends of the estimates stayed the same as conditions were varied.

Odor and Irritation Intensity Evaluation Methods

Odor Evaluation By Dilution Techniques—When the "man in the street" (or in a car following a diesel coach) smells exhaust gas, he is evaluating a mixture of exhaust gas and air. Ideally, then, a laboratory evaluation method should utilize air-exhaust gas mixtures rather than straight exhaust gas. Therefore, our first attempts were directed along this line. Our approach was to evaluate increasing dilutions of exhaust gas in air until the characteristic odor of diesel exhaust could no longer be detected. In the individual tests, a small flow of exhaust gas was measured, mixed with a measured flow of clean air, and the resulting mixture was allowed to pass into the sniff box for evaluation by the panel.

Finding a method for measuring small flows of exhaust gas without changing the composition was a problem that was never resolved with complete satisfaction. The difficulties arose from the fact that exhaust gas is a complex mixture of gases, carbonaceous solids, fuel and oil droplets, other organic compounds, and water. When conventional metering systems like orifices or area-type meters were used, enough particulate matter and condensate collected at the metering point to change the flow rate markedly in a few minutes. This trouble was especially great at the low flow rates corresponding to high dilutions. Traps to remove particulate matter and condensate could not be used since part of the odorant is also removed. In the metering system finally adopted, a water-displacement device introducing zero back pressure was used. Flow measurements were then made only at intervals, and most of the time the throttled exhaust gas was bypassed around the metering device and led directly to the diluting system and sniff box. The exhaust sampling line and throttling valve were maintained at temperatures between 140 and 160 F to prevent condensation of water and consequent absorption of water-soluble compounds. In order to prevent accumulation of odorous materials, the metering and mixing systems were cleaned frequently with acetone.

An independent check on the dilution as calculated from air and exhaust flow rates was made by determining the concentration of oxides of nitrogen in the raw exhaust gas and in the diluted mixture. The agreement between the values of dilution as calculated by the two methods was satisfactory.

An indication of the overall reproducibility of the odor intensity estimates made by the dilution technique is given by the data in Fig. 2. A dilution ratio of 1.0 denotes undiluted exhaust gas, a dilution ratio of 0.10 signifies a tenfold dilution of exhaust gas in air (1 volume of exhaust gas to 9 of air) and so on.

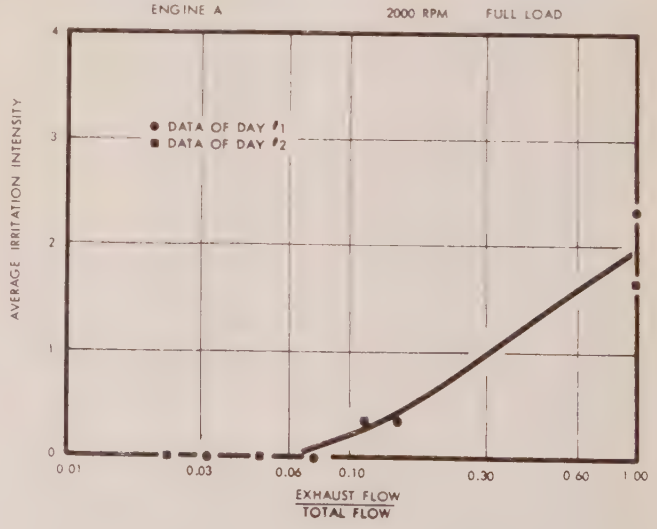

Fig. 3—Reproducibility of irritation data obtained by dilution technique

In general the reproducibility of the estimates was reasonably good. The data in Fig. 2 show that the characteristic odor of diesel exhaust was not eliminated until a dilution ratio of about 0.01 was reached. It is of interest to note that even though the air from the laboratory air system was filtered and passed through activated charcoal before it entered the sniff box, it was judged by some panelists to have a detectibie odor.

Similar data on the reproducibility of irritation intensity estimates are shown in Fig. 3. The reproducibility is again seen to be reasonably good. The data show that the irritant properties of these exhaust gases disappeared at dilutions beyond about 0.06. A comparison of Figs. 2 and 3 shows that substantially less dilution was required to remove the irritant properties of diesel exhaust gas than to eliminate the odor.

The type of data obtained by the dilution method is illustrated by Fig. 4 which shows the effect of engine load at 2000 rpm on the odor intensity of exhaust gases from Engine A at various dilutions. The curves for the four loads studied have similar slopes throughout the dilution range shown. The data also show that full load was the most odorous condition and one-third load the least odorous. The difference in odor intensity between the most odorous and least odorous engine conditions is similar to the day-to-day variation in test results at a single engine condition as shown in Fig. 2. However, these and other reproducibility test results indicate that the difference in odor intensity between the extremes shown in Fig. 4 is probably real. Engine load appears to have had little effect on the rate at which the odor intensity fell off with increasing air dilution, as indicated by the similarity of the slopes of the curves in Fig. 4.

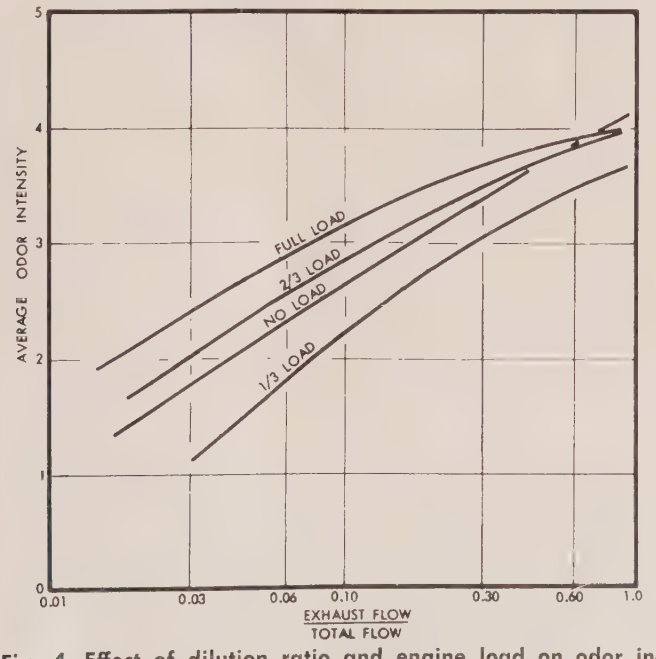

Fig. 4—Effect of dilution ratio and engine load on odor intensity of exhaust gases for Engine A at 2000 rpm

Table 4—Reproducibility of Direct Odor and Irritation Intensity Estimates for Engine C

Engine Speed, rpm	Load	Number of Tests	Average Intensity	Standard Deviation
Odor Intensity (5-Man Panel)				
2000	Full	8	2.76 odor units	0.26 odor unit
2000	Half	9	2.42 odor units	0.27 odor unit
1000	Full	7	2.77 odor units	0.33 odor unit
1000	Half	6	2.47 odor units	0.24 odor unit
Irritation Intensity				
2000	Full	8	2.84 irritation units	0.25 irritation unit
2000	Half	9	2.14 irritation units	0.39 irritation unit
1000	Full	7	2.93 irritation units	0.22 irritation unit
1000	Half	6	2.29 irritation units	0.22 irritation unit

In the work just described, three important disadvantages of the dilution method became evident:

1. Metering of small exhaust flows is difficult and subject to error unless extreme precautions are observed.

2. The metering apparatus and sniff box must be cleaned very frequently, particularly at high exhaust gas dilutions.

3. Maintenance of constant engine conditions for the time required to study several dilutions may be difficult.

Direct Odor Or Irritation Intensity Estimate Techniques—The exhaust gas-air samples evaluated by the dilution method were undoubtedly more like those encountered by the "man in the street" than were the undiluted exhaust gas samples. However, repeated tests similar to those shown in Fig. 4 showed that the order in which engine test conditions rank with respect to odor intensity was generally the same whether estimates were made on equally diluted samples (say a dilution ratio of 0.30) or on undiluted samples. Therefore, the more laborious dilution method was abandoned in favor of the undiluted or direct technique for the majority of the data reported here.

To determine the reproducibility of estimates made by the direct method, repetitive tests were run in Engine C at four conditions. The results are summarized in Table 4. The standard deviations of the odor estimates were similar for all four engine conditions, and the numerical values happen to be about the same for both the odor intensity and irritation intensity estimates. From the data in Table 4, it was concluded that panel-average estimates of odor or irritation intensity, based on limited observations of undiluted exhaust gas from a single engine at different conditions, must differ by at least 0.5 odor or irritation unit before the intensities at the two conditions can be considered to differ significantly. When two different types of engines are compared at a single operating condition, the fact that there may be differences in the quality as well as the intensity of the exhaust odor or irritation introduces an added complication. Therefore, the range of uncertainty is somewhat greater than 0.5 odor or irritation unit. How-

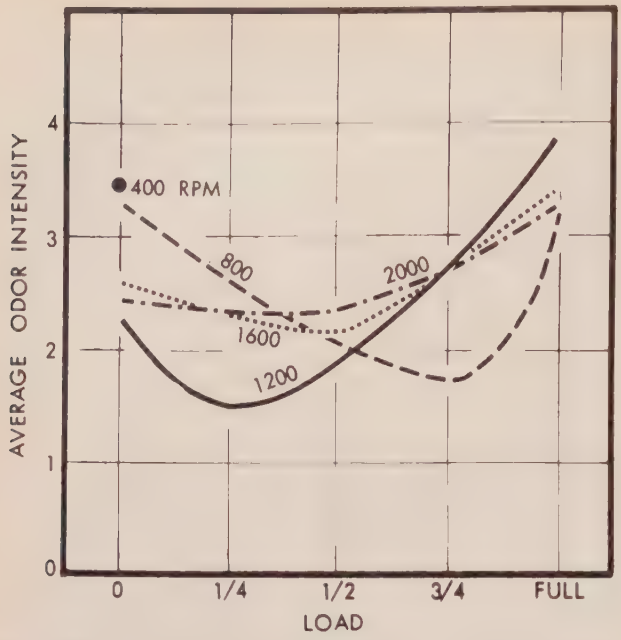

Fig. 5—Effect of engine speed and load on odor intensity of undiluted exhaust gases for Engine A

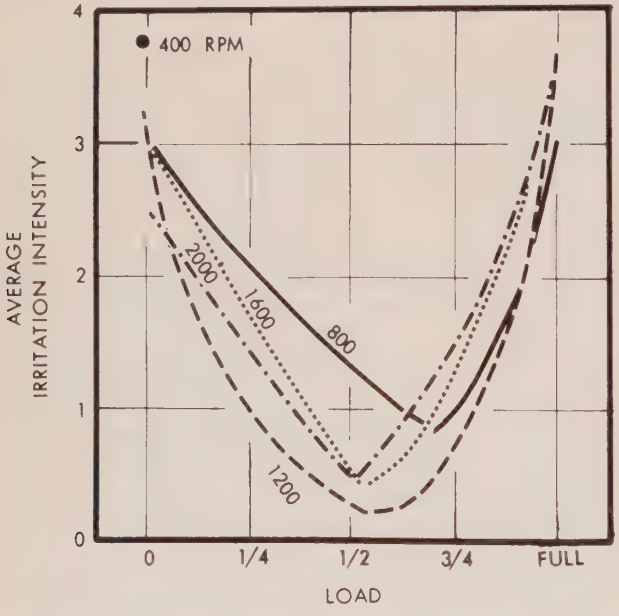

Fig. 6—Effect of engine speed and load on irritation intensity of undiluted exhaust gases for Engine A

ever, the magnitude of the effect in this case was not determined.

Figs. 5 and 6 illustrate typical results obtained by the direct technique. These figures show the effects of engine speed and load on the odor and irritation intensities, respectively, of the exhaust gases from Engine A. In Fig. 5, it is evident that load has a significant effect, the least odorous condition occurring in the mid-load region and the most odorous condition either at full load or for some speeds at no load. No consistent effect of engine speed was observed.

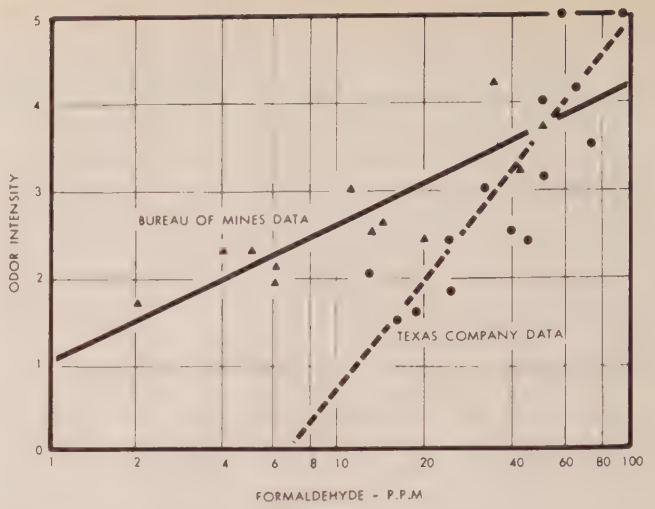

Fig. 7—Correlation between odor intensity and formaldehyde concentration

Similar trends are illustrated in Fig. 6 for the corresponding irritation intensity estimates. For this engine, irritation intensity was more affected by load than was the odor intensity. It will also be noted that the intensity estimates obtained during these studies differed from those obtained during the earlier dilution studies (Figs. 2 to 4) on the same engine. These differences are believed to arise from differences in panel membership and experimental arrangements.

In the work with the direct method, some inherent difficulties became evident:

1. Although the required test time is appreciably less than in the case of the dilution method, the method is still slow and tedious.

2. With undiluted exhaust gases the quality of the odor may influence the intensity estimates.

3. Since undiluted exhaust gases contain significant amounts of toxic materials at some engine conditions, long-continued testing might be undesirable. For example, oxides of nitrogen concentrations over 200 times the recommended 8-hr continuous exposure limit [11] are sometimes found in diesel exhaust gases. (Even higher concentrations of oxides of nitrogen are found in spark-ignition-engine exhaust gases, as illustrated in Table 1.) However, no ill effects on panel members were observed in our study which extended over a period of about a year.

Chemical Analysis Techniques—The third general approach to the problem of odor or irritation intensity evaluation is to find one or more constituents that vary in a systematic, known way with the odor or irritation intensities as determined by a panel. Once such a general relationship has been established, chemical analyses might be used to predict the odor or irritation intensity of the exhaust gas at a previously untested condition or from a previously untested engine. Other investigators using both dilution and

[11] In April, 1954, the American Conference of Governmental Industrial Hygienists adopted a limit of 5 ppm for continuous 8-hr exposure to nitrogen oxides.

direct odor measurements have stated that the odor intensity of exhaust gas is related to the aldehyde content.[12,13,14] A chart summarizing some of this previous work is given in Fig. 7, the published data being converted to the units used in this paper. The two investigating groups which evaluated diesel exhaust found positive correlations between exhaust odor intensity and the aldehyde concentration (reported as formaldehyde) but the relationships found by the two groups differed somewhat. Therefore, it was felt that more experimental work was needed to determine whether a measured constituent (such as aldehydes) or group of constituents could be found that would serve as a reliable index of odor or irritation for all types of diesel engines.

While the odor and irritation intensities of the exhaust gases were being estimated, exhaust gas samples were obtained for determination of total aldehydes, formaldehyde, and oxides of nitrogen. From the analyses for total aldehydes and formaldehyde, "higher aldehydes" were calculated by difference. All the compounds mentioned are highly odorous unless diluted sufficiently and were included in this study because they are present in exhaust gases in significant concentrations under some conditions. Other major constituents of exhaust gas such as nitrogen, hydrogen, oxygen, argon, water vapor, carbon monoxide, and carbon dioxide are odorless. Some other odorous minor constituents such as organic acids and peroxides were omitted from the study because their properties did not make them likely sources of the characteristic diesel exhaust odor.

An indication of the reproducibility of the analyses of exhaust gas samples from Engines A and C obtained over a period of several days is given in Table 5. The oxides of nitrogen analyses are seen to be very reproducible, the standard deviation being only 4.7 to 6.2% of the measured amounts of nitrogen oxides (see last column). The total aldehyde analyses were less reproducible, the standard deviation being 13.7% of the measured values. The formaldehyde analyses were still more variable, especially at the two lowest concentrations. The measured formaldehyde contents of samples drawn in rapid succession at a single engine condition varied less than did those shown in Table 5 for an engine setting established on different days. Consequently some of the variations in Table 5 may reflect real changes in formaldehyde content. Minor variations in experimental conditions such as exhaust system cleanliness and temperature are known to affect the formaldehyde content. For example, a 50% reduction in the formaldehyde content of exhaust gas upon passage through a muffler has been observed in our laboratories.

The effect of engine speed and load on the concentrations of various constituents are shown in Figs. 8 through 11 for the same engine and test conditions as for the odor and irritation intensity data in Figs. 5 and 6. At each speed, the concentrations of total aldehydes, shown in Fig. 8, were minimum at about one-quarter load and maximum at full load. Similar trends were noted for the formaldehyde data, shown in Fig. 9. It will be noted that the highest apparent concentration of formaldehdye, obtained at idle, was higher than for total aldehydes at that condition. This anomaly emphasizes the need for better analytical methods in this field.

Fig. 10 shows that higher aldehydes increased generally with load. When Figs. 9 and 10 are com-

Table 5—Reproducibility of Chemical Analyses

Engine	Load	Speed, rpm	Compounds Analyzed	Dates Sampled	Number of Determinations	Average Content, ppm	Standard Deviation, ppm	Coefficient of Variation, %
A	Full	2000	Oxides of nitrogen	April 12,13	6	954	44.7	4.7
C	Full	600	Oxides of nitrogen	July 20,25	10	630	39.0	6.2
A	Full	2000	Total aldehydes	April 11,13	8	95	13.0	13.7
A	Full	2000	Formaldehyde	April 11,13	9	43	16.6	38
C	Full	2000	Formaldehyde	June 6,8,10	18	18.3	4.2	23
C	Half	2000	Formaldehyde	June 6,8,10	19	8.2	4.0	49
C	Full	1000	Formaldehyde	June 6,8,10	15	26.3	7.7	29
C	Half	1000	Formaldehyde	June 6,8,10	10	5.7	4.0	70

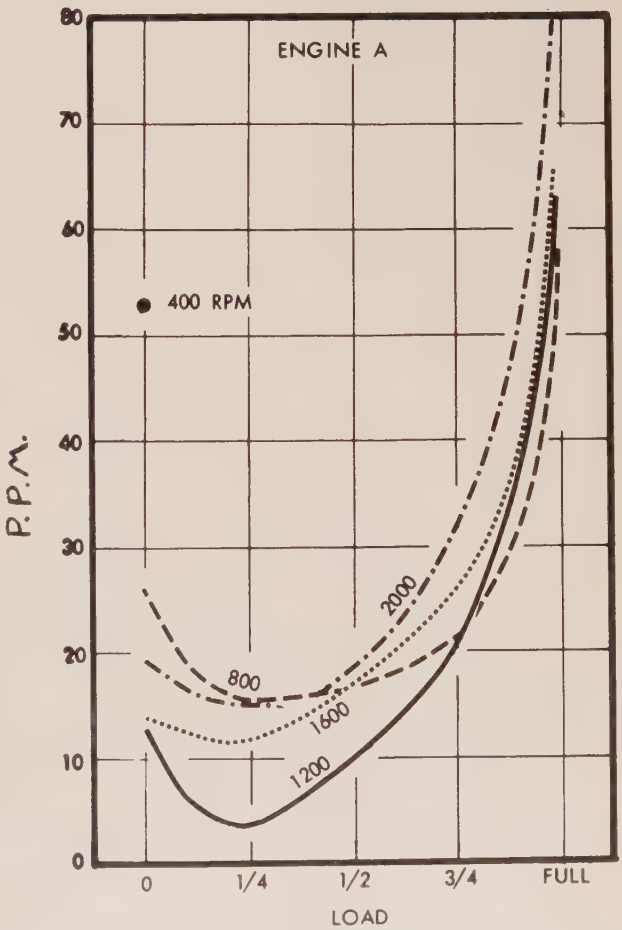

Fig. 8—Effect of engine speed and load on total aldehyde concentration

12 SAE Transactions, Vol. 50, December, 1942, pp. 509–520: "Effect of Diesel Fuel on Exhaust Smoke and Odor," by R. S. Wetmiller and L. E. Endsley, Jr.
13 SAE *Quarterly* Transactions, Vol. 4, July, 1950, pp. 330–346: "Composition of Diesel Exhaust Gas," by M. A. Elliott and R. F. Davis.
14 SAE Transactions, Vol. 51, 1943, pp. 12–19: "Gasoline Engine Exhaust Odors," by J. J. Mikita, H. Levin, and H. R. Kichline.

pared, it appears that formaldehyde was the predominant aldehyde at low loads and full load whereas other aldehydes apparently predominated in the part-load region. The oxides of nitrogen concentrations, shown in Fig. 11, increased almost linearly with load up to about three-quarters load. In summary, the concentrations of total aldehydes and formaldehyde tended to reach minima at part load whereas the concentrations of higher aldehydes and oxides of nitrogen tended to increase with load. At engine speeds above the idle setting, speed had little effect on the concentrations of the measured constituents.

Correlation Between Odor Or Irritation Intensity And Chemical Composition—To test the degree of correlation between concentration of the various measured constituents and odor or irritation intensity, plots like that shown in Fig. 12 were made. The figure shows all the available data from four different diesel engines (two 2-cycle and two 4-cycle engines) relating formaldehyde concentration with odor intensity. The dashed and dotted lines are the "best" straight lines through the experimental points for Engines A

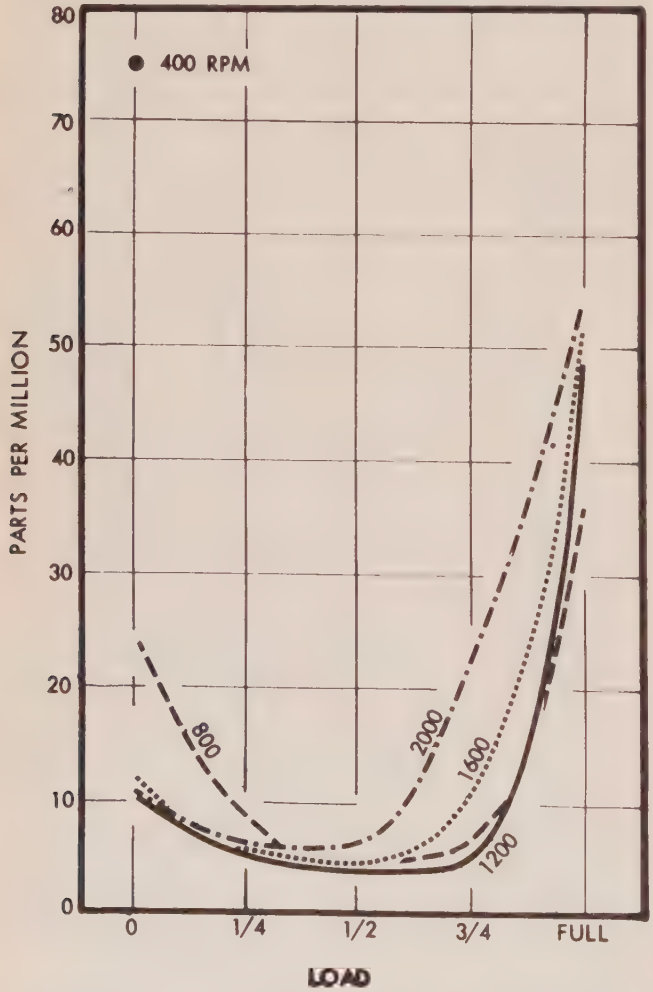

Fig. 9—Effect of engine speed and load on formaldehyde concentration for Engine A

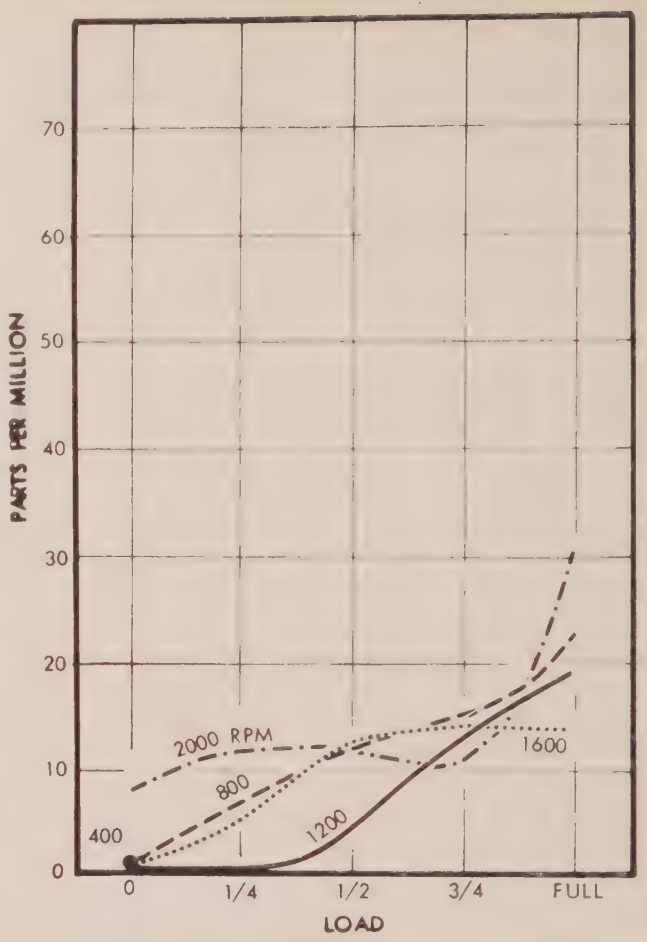

Fig. 10—Effect of engine speed and load on higher aldehyde concentration for Engine A

and C, respectively, and will be discussed later. The formaldehyde concentration is plotted on a logarithmic scale since such a relationship (Weber-Fechner Law) generally applies in sense perception studies such as these.[15] The Weber-Fechner Law states that the ratio of the intensity of the applied stimulus (such as an odorant material in exhaust gas) to the incremental change in intensity of the stimulus which is necessary to produce a "just noticeable difference" in response (such as odor detection) is independent of the original intensity of the stimulus. If there were a strong correlation between odor intensity and formaldehyde concentration, the data should fall in a line or band with a positive slope toward increasing formaldehyde concentrations. However, when an odorant (such as formaldehyde) becomes diluted beyond its odor threshold value, no correlation between its concentration and the odor intensity can be expected. The data in Fig. 12 fall in a band which rises only slightly with increasing formaldehyde concentration in the range shown. The results suggest, therefore, that although there may be some relationship between formaldehyde concentration and odor

[15] "Handbook of Experimental Psychology," by S. S. Stevens. Pub. by John Wiley and Sons, New York, 1951.

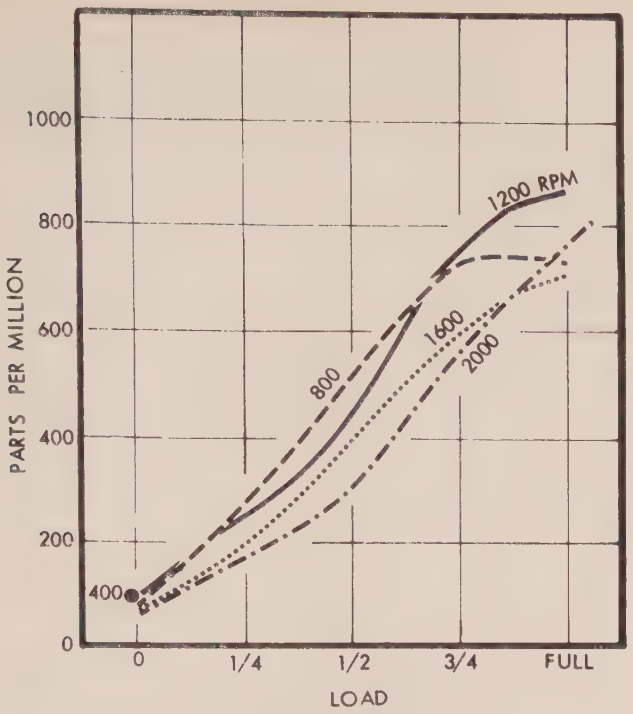

Fig. 11—Effect of engine speed and load on oxides of nitrogen concentration for Engine A

intensity in the range shown, the correlation is weak. Corresponding plots for the other measured constituents versus odor or irritation intensity show similar trends but correlations were even less evident.

Although none of the plots showed strong correlation, the possibility remained that statistically significant but weak correlations did in fact exist. Furthermore, plots such as Fig. 12 deal with only one constituent whereas some combination of analytical data for two constituents might give a stronger correlation. Therefore, the data were analyzed statistically on an electronic computer. The degree of correlation between odor or irritation intensity and the concentration of each constituent was determined. Then the calculations were extended to determine the degree of correlation between the optimum weighted combination of the concentrations of each pair of constituents and odor or irritation intensity. For the calculations it was assumed that the line of regression (best line through the experimental points) was a straight line on either a logarithmic plot (such as Fig. 12) or on a linear plot. Details on the statistical methods and calculations are given in the Appendix.

A summary of the calculations for odor intensity is given in Table 6 and for irritation intensity in Table 7. Examination of the odor intensity data indicates that only (a) total aldehydes and (b) higher aldehydes plus formaldehyde (with optimum weights being assigned to each of these) gave acceptable correlations that apply to both engines. The latter correlation is interesting in view of the fact that values for higher aldehydes are probably the least reliable of all the concentration data since they were obtained by difference between values which are themselves subject to considerable uncertainty. According to statistical theory, the square of the correlation coefficient (r^2) denotes the fraction of the dependent variable (odor or irritation intensity) that can be accounted for by the change in the independent variable (the concentration of the constituent or pair of constituents being considered). With this criterion and a logarithmic equation for the line of regression, almost 70% of the observed variation in odor intensity in Engine A can be accounted for by changes in the concentrations of higher aldehydes and formalde-

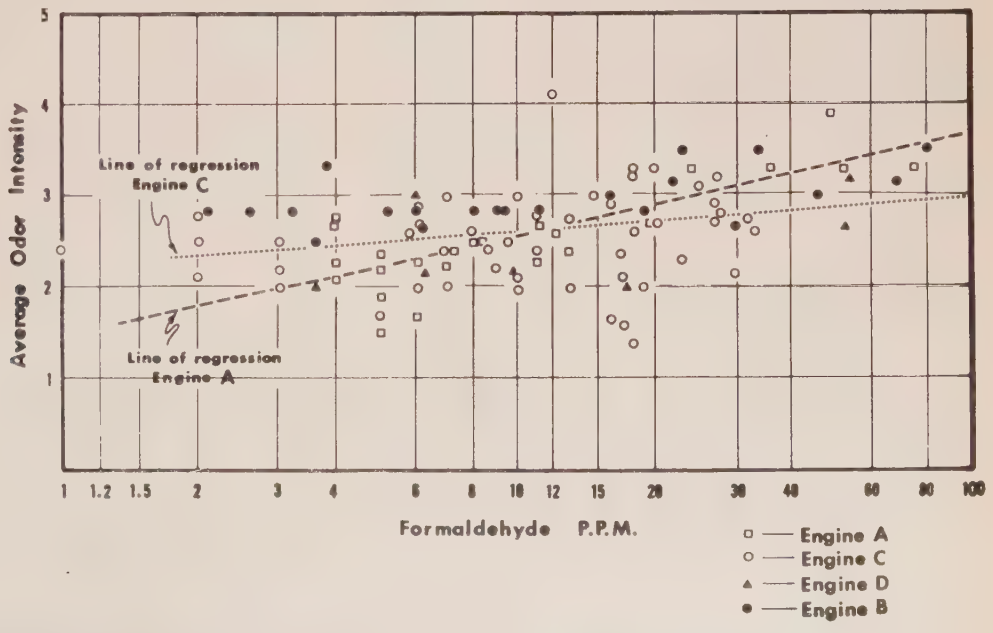

Fig. 12—Relation between odor intensity and formaldehyde content of diesel exhaust gas

Table 6—Correlation Between Concentrations of Constituents and Odor Intensity

Engine	Constituent 1	Constituent 2	Coefficients of Equation for the Line of Regression: a or d	b or e	c or f	Standard Error of Estimate, Odor Units	Correlation Coefficient r	r^2	Is Correlation Significant at 95% Confidence Level?	Are Correlation Coefficients Similar at 95% Confidence Level?	Are Regression Coefficients Similar at 95% Confidence Level?	Are Regression Intercepts Similar at 95% Confidence Level?
EQUATION (1)		Odor Intensity = $a + bC_1 + cC_2$										
A	Total aldehydes		2.069	0.0232		0.39	0.793	0.628	Yes	Yes	Yes	Yes
C	Total aldehydes		1.912	0.0178		0.45	0.437	0.191	Yes			
A	Formaldehyde		2.198	0.0222		0.35	0.778	0.605	Yes			
C	Formaldehyde		2.314	0.00441		0.56	0.0497	0.00247	No			
A	Higher aldehydes		2.465	0.0167		0.28	0.187	0.0350	No			
C	Higher aldehydes		1.936	0.0292		0.47	0.533	0.284	Yes			
A	Oxides of nitrogen		2.376	0.000480		0.52	0.312	0.0973	No			
C	Oxides of nitrogen		1.751	0.000584		0.42	0.651	0.423	Yes			
A	Total aldehydes	Oxides of nitrogen	2.133	0.0270	−0.000344	0.32	0.816	0.663	Yes	Yes	No	
C	Total aldehydes	Oxides of nitrogen	1.519	0.0121	0.000519	0.39	0.713	0.507	Yes			
A	Formaldehyde	Oxides of nitrogen	2.166	0.0216	0.000971	0.34	0.782	0.609	Yes	Yes	No	
C	Formaldehyde	Oxides of nitrogen	1.283	0.0313	0.000706	0.38	0.729	0.530	Yes			
A	Higher aldehydes	Oxides of nitrogen	2.591	−0.00156	+0.000497	0.52	0.313	0.0975	No			
C	Higher aldehydes	Oxides of nitrogen	1.690	0.0131	0.000461	0.43	0.681	0.462	Yes			
A	Higher aldehydes	Formaldehyde	2.105	0.0135	0.0220	0.34	0.793	0.628	Yes	Yes	Yes	Yes
C	Higher aldehydes	Formaldehyde	1.977	0.0297	−0.00444	0.47	0.536	0.286	Yes			
EQUATION (2)		Odor Intensity = $d + e \ln C_1 + f \ln C_2$										
A	Total aldehydes		1.305	0.463		0.38	0.726	0.525	Yes	Yes	Yes	Yes
C	Total aldehydes		1.523	0.253		0.56	0.381	0.145	Yes			
A	Formaldehyde		1.427	0.488		0.33	0.814	0.660	Yes			
C	Formaldehyde		2.264	0.153		0.57	0.0527	0.0028	No			
A	Higher aldehydes		2.522	0.042		0.55	0.0837	0.0070	No			
C	Higher aldehydes		1.733	0.239		0.54	0.537	0.287	Yes			
A	Oxides of nitrogen		2.211	0.068		0.55	0.161	0.0258	No			
C	Oxides of nitrogen		0.148	0.370		0.46	0.572	0.325	Yes			
A	Total aldehydes	Oxides of nitrogen	1.474	0.484	−0.0407	0.38	0.734	0.535	Yes	Yes	No	
C	Total aldehydes	Oxides of nitrogen	−1.065	0.343	0.359	0.40	0.704	0.493	Yes			
A	Formaldehyde	Oxides of nitrogen	1.494	0.492	−0.0144	0.32	0.817	0.664	Yes	Yes	No	
C	Formaldehyde	Oxides of nitrogen	−1.646	0.348	0.490	0.40	0.698	0.485	Yes			
A	Higher aldehydes	Oxides of nitrogen	2.206	0.0225	0.0629	0.55	0.167	0.027	No			
C	Higher aldehydes	Oxides of nitrogen	0.142	0.242	0.246	0.43	0.642	0.410	Yes			
A	Higher aldehydes	Formaldehyde	1.260	0.0932	0.500	0.36	0.836	0.697	Yes	Yes	Yes	Yes
C	Higher aldehydes	Formaldehyde	1.419	−0.00247	0.392	0.47	0.538	0.288	Yes			

hyde whereas only about 29% of the variation in Engine C can be so explained. It is of interest that the panel members reported a distinct difference in the odor quality of exhaust gases from the two engines.

Similarly, the irritation intensity data in Table 7 indicate that a combination of the data for higher aldehydes and formaldehyde gave the best correlations acceptable for both engines. The amount of the variation in irritation intensity accounted for by variations in the concentrations of these constituents is only about 52% and 25% for Engines A and C, respectively.

Almost equally satisfactory correlations were obtained by the use of either linear or logarithmic equations even though the usual finding in sense-perception studies is that logarithmic equations fit the data best (the Weber-Fechner Law). This suggests that the Weber-Fechner Law is not fully applicable in the range of concentrations covered in the present studies.

Even though statistically significant correlations were obtained between certain chemical analyses and the odor or irritation intensity, the question remains as to whether chemical analyses of an unknown sample of exhaust gas could be used to predict reliably the odor or irritation intensity of that sample. To answer this question, the standard error of estimate was calculated for each line of regression. These results are also shown in Tables 6 and 7. For the pair of constituents giving the best correlation, higher aldehydes plus formaldehyde, the observed odor intensity data would be expected, at the 95% confidence level, to fall in a band having a width equal to four times the standard error of estimate or 1.6 odor units. The band would center on the line of regression. Since the total spread of the observed odor intensity estimates was 2.4 and 2.7 odor units for Engines A and C, respectively, the expected variation in the odor intensity for a fixed exhaust gas composition is 60 to 70% of the total observed spread in odor intensity measured at different exhaust gas compositions. A similar analysis of the irritation intensity data indicates that the variation in the irritation intensity values at a fixed exhaust gas composition represents 70 to 90% of the total spread in the observed irritation intensity data. Consequently, the concentrations of the exhaust gas constituents measured in the present study cannot be used to predict reliably the changes in odor or irritation intensity which would accompany changes in factors such as the engine operating conditions, the engine design, the fuel, or

the lubricant. Further, the data suggest either that constituents other than those measured are contributing significantly to odor and irritation or that the chemical methods used are not measuring accurately the constituents they are intended to measure.

Comparison Of Evaluation Techniques—From the data presented on the three techniques for evaluating odor and irritation intensity, some conclusions as to their relative merit can be reached. All three were found to have important limitations, but the direct estimation of odor or irritation intensity on undiluted exhaust gas appeared to have the most merit for engine studies at present. It is less laborious than the diluted exhaust gas technique and more accurate than the chemical analysis technique. However, the latter may become more reliable if better means of analysis can be developed. In this connection, newer analytical methods such as partition chromatography may be of great service.

Engine Test Results

Effects Of Engine Variables—To obtain a better understanding of the factors influencing diesel odor, tests were made to determine the effects of engine operating conditions, fuel type, and lubricating oil type on the odor and irritant properties of diesel exhaust gas. For the engine variables study, both the dilution and direct techniques were employed. The results of these tests showed that the effects of the engine variables on odor intensity and on irritation intensity were generally similar, and for brevity only the odor data will be discussed.

The two variables most frequently changed in normal engine operation are the speed and load. The effects of these variables have been illustrated previously in Fig. 5. As was pointed out earlier, engine load had a relatively large effect, the least odorous condition being part load and the most odorous conditions being either full load or for some engine speeds, no load. Engine speed, on the other hand, had little effect on odor intensity. The data indicate that diesel exhaust gas is most odorous at either no load (very lean fuel-air mixtures) or at full load (relatively rich fuel-air mixtures). At part load (moderately lean fuel-air mixtures) the odor is measurably less intense.

A comparison of the odor intensities of exhaust gases from three different makes of open-combustion-

Table 7—Correlation Between Concentrations of Constituents and Irritation Intensity

Engine	Constituent 1	Constituent 2	Coefficients of Equation for the Line of Regression: g or j	h or k	i or l	Standard Error of Estimate, Irritation Units	Correlation Coefficient r	r^2	Is Correlation Significant at 95% Confidence Level?	Are Correlation Coefficients Similar at 95% Confidence Level?	Are Regression Coefficients Similar at 95% Confidence Level?	Are Regression Intercepts Similar at 95% Confidence Level?
EQUATION (3)		Irritation Intensity = $g + h\,C_1 + i\,C_2$										
A	Total aldehydes		1.384	0.0345		0.95	0.585	0.342	Yes	Yes	Yes	Yes
C	Total aldehydes		1.928	0.0171		0.62	0.384	0.148	Yes			
A	Formaldehyde		1.483	0.0384		0.75	0.668	0.447	Yes	Yes	Yes	Yes
C	Formaldehyde		1.848	0.0462		0.54	0.474	0.224	Yes			
A	Higher aldehydes		2.258	−0.0153		1.09	−0.0850	0.00722	No			
C	Higher aldehydes		2.172	0.0127		0.59	0.210	0.0442	No			
A	Oxides of nitrogen		2.354	−0.000473		1.05	−0.152	0.0231	No			
C	Oxides of nitrogen		2.432	−0.0000776		0.61	−0.0788	0.00622	No			
A	Total aldehydes	Oxides of nitrogen	1.815	0.0598	−0.00230	0.60	0.843	0.707	Yes			
C	Total aldehydes	Oxides of nitrogen	2.069	0.0191	−0.000180	0.56	0.424	0.179	No			
A	Formaldehyde	Oxides of nitrogen	1.905	0.0463	−0.00129	0.70	0.777	0.601	Yes	Yes	No	
C	Formaldehyde	Oxides of nitrogen	1.680	0.0508	0.000120	0.53	0.488	0.237	Yes			
A	Higher aldehydes	Oxides of nitrogen	2.346	0.00348	−0.000511	1.10	0.153	0.0234	No			
C	Higher aldehydes	Oxides of nitrogen	2.331	0.0228	−0.000291	0.58	0.322	0.103	No			
A	Higher aldehydes	Formaldehyde	1.625	−0.0207	0.0387	0.82	0.679	0.459	Yes	Yes	Yes	Yes
C	Higher aldehydes	Formaldehyde	1.769	0.0439	0.00767	0.53	0.491	0.240	Yes			
EQUATION (4)		Irritation Intensity = $j + k\,\ln C_1 + l\,\ln C_2$										
A	Total aldehydes		0.235	0.697		0.94	0.541	0.292	Yes			
C	Total aldehydes		1.907	0.110		0.68	0.288	0.0836	No			
A	Formaldehyde		0.077	0.876		0.77	0.723	0.520	Yes	Yes	No	
C	Formaldehyde		1.463	0.459		0.49	0.477	0.227	Yes			
A	Higher aldehydes		2.276	−0.087		1.10	−0.0867	0.0075	No			
C	Higher aldehydes		2.291	−0.017		0.68	0.204	0.0475	No			
A	Oxides of nitrogen		3.722	−0.270		1.05	−0.339	0.114	No			
C	Oxides of nitrogen		3.586	−0.191		0.69	−0.148	0.0218	No			
A	Total aldehydes	Oxides of nitrogen	2.268	0.954	−0.502	0.71	0.778	0.602	Yes			
C	Total aldehydes	Oxides of nitrogen	2.166	0.262	−0.0907	0.58	0.318	0.101	No			
A	Formaldehyde	Oxides of nitrogen	2.223	1.028	−0.460	0.50	0.897	0.800	Yes	No		
C	Formaldehyde	Oxides of nitrogen	0.826	0.449	0.0840	0.53	0.490	0.239	Yes			
A	Higher aldehydes	Oxides of nitrogen	3.723	0.000467	−0.288	1.05	0.340	0.115	No			
C	Higher aldehydes	Oxides of nitrogen	3.132	0.300	−0.225	0.48	0.356	0.126	No			
A	Higher aldehydes	Formaldehyde	0.074	0.00249	0.873	0.77	0.725	0.523	Yes	Yes	Yes	Yes
C	Higher aldehydes	Formaldehyde	1.227	0.383	0.125	0.53	0.502	0.252	Yes			

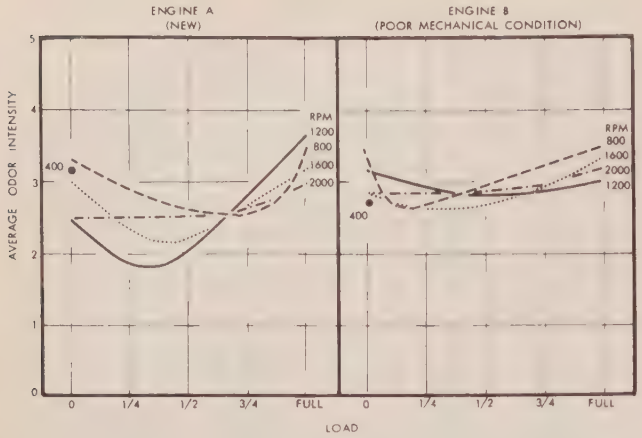

Fig. 13—Effect of mechanical condition on odor intensity of exhaust gases

chamber diesel engines is given in Table 8. Engine A used the 2-stroke cycle; Engines C and D used the 4-stroke cycle. In Engine C the fuel is sprayed across the air space whereas in Engine D the fuel is directed onto the piston. At 1200 rpm half load, the odor intensities of the exhaust gases from all three engines were practically the same. At full load and no load, however, there were differences among the three engines. Engine D gave relatively low odor intensities at full load, perhaps because of the fact that Engine D develops less power per cubic inch at full throttle than the other two. At no load, the observed differences in odor intensity may reflect differences in how completely the fuel burns in the three engines at very lean fuel-air ratios. On the basis of the data in Table 8 (as well as similar data obtained at other engine conditions), it appears that the differences between engine makes with respect to exhaust odor intensity are generally small in the normal operating range with the exception of the idle condition.

Another variable which might be expected to affect odor intensity is the mechanical condition of the engine. To evaluate this factor, tests were made on Engine B. This was of the same type as Engine A but had been removed from commercial service because of noise, high oil consumption, and excessive smoke. Odor intensity estimates were made on Engine B at a number of test conditions, and the results were compared with data obtained earlier with Engine A using the same panel and experimental arrangements. The results, summarized in Fig. 13, showed generally similar odor intensities for the two engines throughout the tests although the exhaust gases of Engine B may have had slightly higher odor intensities at the part-load conditions. These results indicate that exhaust smoke and odor intensity are not necessarily related, inasmuch as Engine B gave dense smoke at three-quarters and full load whereas Engine A gave practically no smoke under any conditions.

It should be pointed out that the injectors from

Table 8—Effect of Engine Make on Exhaust Odor Intensity

Speed, rpm	Load	Average Odor Intensity		
		Engine A	Engine C	Engine D
2000	Full load	3.0	2.9	2.0
1200	Full load	3.7	2.8	3.0
1200	Half load	2.0	2.0	2.2
1200	No load	2.5	2.1	3.2
400	No load	3.2	1.6	2.7

Engine B were removed after the test and found to be in good condition. Therefore, the data do not reflect the influence of injector condition. Workers at the Bureau of Mines have stated [16] that leaking injectors significantly increase the aldehyde content of exhaust gas, and it is definitely possible that injector condition might also affect odor intensity.

The type of fuel used is another variable which might be expected to affect exhaust gas odor intensity. Two fuel properties which are believed by some investigators to be related to exhaust odor are the ignition characteristics (as reflected by cetane number) and the volatility (as reflected by end point). Previous investigators [12,13] have reported that at idle the aldehyde content (and, by implication, the odor intensity) increases as the cetane number of the fuel decreases. To investigate these possibilities, the three fuels described in Table 9 were evaluated at several test conditions. For Engine A at idle and at 2000-rpm full load (two of the more troublesome conditions with respect to odor), no significant difference in exhaust odor intensity was observed when fuel X (low end point, intermediate cetane number) was replaced with fuel Z (high end point, low cetane number). However, the latter fuel gave significantly higher formaldehyde concentrations at idle. Other tests in which Engine C was operated at 1600 rpm with engine loads from full load to no load indicated that the exhaust gases were only slightly less odorous with fuel X than with fuel Y, which had a higher end point but almost the same cetane number.

Tests were also run comparing No. 2 diesel fuel, light crude oil, and Bunker C fuel in a free-piston engine, which burns the fuel in the same manner as a diesel engine. The sulfur contents of the three fuels were 0.72, 0.48, and 3.31%, respectively. No differences in odor or irritation intensities were observed between the first two fuels, but the Bunker C fuel gave considerably higher odor and irritation intensities. It seems probable that the high sulfur content of the Bunker C fuel was responsible for these increases. In short, the limited test results suggest that, as far as diesel fuels in the normal composition range are concerned, only small reductions in exhaust odor can be achieved through improved control of fuel properties.

[16] Bureau of Mines Report of Investigations No. 3700, 1943, "Diesel Engines Underground: IV—Effect on Composition of Exhaust Gas Variables Influencing Fuel Injection," by J. C. Holtz, L. B. Berger, M. A. Elliott, and H. H. Schrenk.

Another variable investigated was the effect of lubricating oil type on the exhaust odor. Table 10 shows the results of a study comparing a conventional fully compounded Mid-Continent mineral oil, a commercial polyalkylene glycol oil, and a commercial diester oil. The data indicate that the lubricating oil had little effect on the exhaust odor intensity. However, the odor quality differed considerably in the three series of tests. At low speeds and loads, the diester oil gave pleasanter smelling exhaust gases than did the mineral oil whereas at high speeds and loads the opposite was true. At all conditions, the exhaust gases when polyalkylene glycol oil was used were more offensive than when mineral oil was used. From these results it seems certain that diesel odor is affected by the type of oil used.

Other Test Results—It has been suggested in the past that diesel odor might be reduced by adding a combustible gas to the intake air of the engine with the hope of eliminating localized areas in the combustion chamber where the mixture is too lean to burn. To investigate this possibility propane was added to the intake air of Engine A, with the results shown in Table 11. The amount of propane added in these tests (0.45% by volume) was well below the 2.4% required to reach the lower flammability limit. The propane addition had little effect except to lower the irritation intensity at full load and to increase the irritation intensity and formaldehyde concentration at no load. In previous work at the Bureau of Mines,[13] the addition of methane was found to increase markedly the formaldehyde concentration in the exhaust gas. Thus the addition of a combustible gas to the intake air appears to offer little promise for reduction of odor.

Another possible odor control measure is to find a material which, when added to the intake air or exhaust gas of a diesel engine, will react with the odorous constituents in the exhaust gas and convert them to nonodorous reaction products. To test this approach, various amounts of gaseous ammonia were added. (Ammonia is known to react quickly with formaldehyde at somewhat elevated temperatures to form hexamethylene tetramine, a white solid having little odor.) The data in Table 11 summarize the results obtained for the ammonia additions that gave maximum odor reduction. The additions to the intake air almost completely eliminated the formaldehyde at full load and greatly reduced it at no load. However, no lowering of the odor or irritation intensities was obtained at either load condition. Addition of ammonia to the exhaust gases eliminated the formaldehyde and reduced the irritation intensity at full load and, in addition, slightly reduced the odor intensity at the two loads tested. It was also observed that a greasy, opaque deposit was formed on all areas exposed to the exhaust gases. As a practical odor control measure, the addition of ammonia did not offer much promise. Both the ammonia-addition and the propane-addition results suggest that formaldehyde is one of the less important contributors to odor, but that it does contribute importantly to irritation.

Another series of tests was run to determine the benefits that might be derived from filtering the exhaust gas. The tests were made with an engine of the same type as Engine A in which a large glass-wool filter (2 sq ft in area, 4 in. deep) was mounted in the exhaust line. In tests run at idle, the odor intensity immediately decreased from the original 3.0 odor units to 0.8 odor unit. However, the filter lost its odor-retaining ability in a few hours. Glass wool which had been wet initially with liquids such as lubricating oil or silicone oil had a somewhat longer effective life, but even in this case the filter soon became ineffective. When filtration tests were run at 2000-rpm full load or even 1200-rpm half load, oil and sooty material which had collected on the filter fumed and smoldered intolerably. Therefore, a means of cooling the exhaust gas before it reaches the filter apparently would be required for such a system to be practical. These filtration experiments indicated strongly that aerosols

Table 9—Properties of Fuels Tested

	Fuel X	Fuel Y	Fuel Z
Specific Gravity at 60F	0.814	0.853	0.897
ASTM Distillation, F			
Initial	355	354	392
10%	399	412	475
50%	436	500	532
90%	501	573	603
End point	548	603	648
Composition, %			
Paraffins	82.9	65.6	52.2
Olefins	3.7	4.0	10.6
Aromatics	13.4	50.4	37.2
Sulfur	0.11	0.46	0.37
Cetane Number	51.1	47.9	29.6

Table 10—Effect of Lubricating Oil on Exhaust Gas Odor Intensity for Engine A

Speed, rpm	Load	Dilution Ratio	Odor Intensity: Mineral Oil	Odor Intensity: Polyalkylene Glycol Oil	Odor Intensity: Diester Oil
700	1/3 load	0.20	2.9	3.2	2.7
1200	2/3 load	0.20	3.1	3.5	2.8
2000	Full load	0.20	3.1	3.9	3.6

Table 11—Effect of Added Gases on Diesel Exhaust Gas Odor Intensity (Engine A, 800 rpm)

Load	Added Gas	Concentration, vol%	Point of Addition	Formaldehyde, ppm	Odor Intensity, Odor Units	Irritation Intensity, Irritation Units
Propane Additions						
Full load	None			41[a]	2.8[a]	2.7[a]
Full load	Propane	0.45	Intake	41	2.7	1.7
No load	None			30[a]	2.8[a]	2.9[a]
No load	Propane	0.45	Intake	122	3.0	3.8
Ammonia Additions						
Full load	None			41[a]	2.8[a]	2.7[a]
Full load	Ammonia	0.12	Intake	0	3.0	2.5
Full load	Ammonia	0.05	Exhaust	1	2.3	0.5
No load	None			30[a]	2.8[a]	2.9[a]
No load	Ammonia	0.12	Intake	14	2.5	3.3
No load	Ammonia	0.05	Exhaust	29	2.2	3.3

[a] Average of two runs

Table 12—Minimum Significant Correlation Coefficient Values

Confidence Level	Correlation Coefficient For One Constituent	Correlation Coefficient For a Pair of Constituents
90%	0.323	0.374
95%	0.381	0.431
99%	0.487	0.467

Table 13—Minimum Values of K for Acceptable Confidence Levels

Confidence Level	K
90%	2.074
95%	2.508
99%	2.819

and/or particulate matter in exhaust gas contribute significantly to diesel odor and irritation.

Tests were also made to determine the effectiveness of an electrostatic precipitator and a cyclone separator for removing odorous materials from diesel exhaust gas. Both devices were almost totally ineffective.

What Can Be Done About Diesel Halitosis?—At this point the question, "What can be done practically to reduce diesel exhaust odor?" might well be asked. Possible solutions to the problem all seem to fall into four categories of prevention, physical removal, chemical destruction, and masking.

1. *Prevention*—the most desirable solution to the odor problem might be to prevent the odorous materials from forming. One conceivable approach to the problem is through control of engine load since part-load operation ordinarily gave less odorous exhaust gas than did full-load operation. Consequently, it might be more desirable, odorwise, to use a larger engine at part load than to use a smaller engine at full load.

Other potential improvements in the odor situation might arise through refinements in engines. Although our data did not show large differences among the engine makes studied, future changes in engines and/or components might be helpful. Similarly, our limited tests on the effects of lubricating oil type and fuel type indicated that there was little to be gained in this direction. However, radical changes in fuels and lubricating oils might prove beneficial. Attempts to prevent the formation of the odorants by the addition of a combustible gas were ineffective.

2. *Physical Removal*—Possible means of physically removing odorous materials from exhaust gas include passing the gas through a filter or an adsorbing medium such as activated charcoal. Tests have shown that a filter does reduce the odor of diesel exhaust gases under some conditions. However, the size and cost of an effective filter system appear to be prohibitive for commercial vehicles.

No tests were run to determine the effectiveness of adsorbents such as charcoal for removing odor from diesel exhaust gases. However, calculations showed that this approach is impractical because of the size and complexity of equipment required.

3. *Chemical Destruction*—Catalytic mufflers may have some promise for the control of diesel odor. These devices contain a catalyst that promotes full oxidation of small amounts of hydrocarbons and other materials which may ordinarily escape from the combustion chamber unburned. However, the catalyst is effective only at high temperatures and, consequently, the average diesel vehicle would require an auxiliary heater to keep the catalyst hot. Since engine fuel is used in the auxiliary heater, the resultant fuel expense might make the diesel engine uneconomic. A recent development, not yet fully evaluated for road vehicles, uses an exhaust heat exchanger to keep the catalyst hot, and thereby reduces the amount of auxiliary heat required.

Exhaust scrubbers have been used in mine operations to remove some of the objectionable constituents such as certain aldehydes. In these units, the exhaust gases are passed through water, limestone and water, or aqueous sodium sulfite solution.[17] In tests of a commercial unit towed behind a diesel coach, aldehydes were removed, but great clouds of steam were formed which obscured the vision of following motorists. Therefore, any practical device would require an exhaust cooler and radiator in addition to the scrubber. This fact along with the added expense, weight, and bulk make the use of scrubbers on highway vehicles appear impractical.

Although the addition of ammonia gave little odor benefit, the results do not preclude the possibility that some additive might be found which would appreciably reduce the odor and irritation intensity of diesel exhaust gases. However, in view of the possible health hazards of the reaction products as well as the problems of economics, rate control, and product disposal, a practical solution by this approach seems unlikely.

4. *Masking*—Another approach to the odor problem is to add odorant materials (or "masking agents") to the fuel or exhaust gas of a diesel engine. The object of masking agents is not to change an unpleasant odor but rather to cover up the odor with another which is more pleasant. Some work on masking agents for diesel exhaust gas is under way, but no results are yet available. Masking treatments of one kind or another have been applied successfully and economically in the food products, paper, petroleum, and rubber industries.

In this paper, three techniques for evaluating the odor and irritation intensities of diesel exhaust gases were investigated. These were (1) the evaluation of diluted exhaust gas by a panel, (2) the evaluation of undiluted exhaust gas by a panel, and (3) prediction of the odor or irritant properties from measured exhaust gas composition. The undiluted exhaust gas technique appeared to be the most suitable for engine studies since it did not require as much time or such precise metering of small gas flows as did the diluted exhaust gas technique, and it was much more reliable than predicting odor intensities by analyses of exhaust

[17] ASME Transactions, Vol. 70, 1948, pp. 745–750: "Removal of Aldehydes from Diesel Exhaust Gas," by R. F. Davis and M. A. Elliott.

Table 14—Analysis of Variance Between Lines of Regression for Higher Aldehydes plus Formaldehyde versus Odor Intensity by Equation (2)

Source of Variance	Degrees of Freedom	Sum of Squares	Estimated Mean Squares
Difference between regression coefficients	3	1.005	$0.335 = E_D$
Overall regression coefficient	1	4.448	
Sum of regression coefficients	4	5.453	
Residual	44	9.802	$0.223 = E_R$
Total	48	15.255	

gas. The latter approach appears unsuitable because changes in concentration of those constituents which gave the best correlation with odor and irritation intensity, formaldehyde plus higher aldehydes, accounted for only part of the observed changes in the odor and irritant properties of the exhaust gas samples tested. Further development of analytical methods might improve this situation.

Panel evaluation methods were used to determine the influence of several engine variables on the properties of exhaust gas. Limited studies described in the paper indicated that engine load has a pronounced effect on exhaust odor and irritation intensity. Engine make, engine speed, engine condition, fuel type and lubricating oil type had comparatively little effect.

Several special approaches to exhaust-gas-odor reduction were tried, but no panacea was found. For the present, close attention should be given to factors such as improved engine and injector design, proper fuel and oil, good maintenance, and avoidance of overloading. For the future, catalytic mufflers and masking agents may offer some promise.

Acknowledgment

The authors wish to acknowledge the help of many individuals of the General Motors Research Staff, the General Motors Truck and Coach Division, and the Detroit Diesel Division who assisted in running the tests and preparing the manuscript. Particular appreciation goes to the heroes of science who evaluated a multitude of unsavory odors. Thanks are also due Dr. M. Elliott of the Institute of Gas Technology for guidance in planning and running the tests, and J. Dallemand of General Motors Research Staff for the statistical calculations.

APPENDIX

Before a correlation between some exhaust gas constituent or pair of constituents and odor or irritation intensity can be considered for general use in diesel-engine studies, it must be shown that the correlation applies to exhaust gases from "all" different types of diesel engines, or at least from widely differing types such as Engines A and C. Consequently exhaust-gas data for Engines A and C were analyzed statistically. For the calculations it was assumed that the line of regression (best line through the experimental points) was in each instance a straight line describable by one of the following equations:

$$\text{Odor intensity} = a + bC_1 + cC_2 \quad (1)$$

$$\text{Odor intensity} = d + e \ln C_1 + f \ln C_2 \quad (2)$$

$$\text{Irritation intensity} = g + hC_1 + iC_2 \quad (3)$$

$$\text{Irritation intensity} = j + k \ln C_1 + l \ln C_2 \quad (4)$$

where:

$a, b \ldots k, l$ are constants

C_1 = Concentration of one measured constituent

C_2 = Concentration of second measured constituent

The lines of regression were obtained by the method of least squares. Four successive questions were then asked regarding each combination of exhaust gas constituent or pair of constituents, engine, and equation for the line of regression. Failure to answer any question affirmatively at the 95% confidence level eliminated that particulaar combination from further consideration. The questions asked were: [18]

1. For either engine, is the correlation coefficient large enough to be statistically significant? The answer to this question was yes if the calculated correlation coefficient exceeding the values in Table 12.

2. Is the correlation coefficient similar in magnitude for the two engines? The answer to this question was yes if K, defined by the following formula, exceeded the values given in Table 13.

$$K = \frac{R_1 - R_2}{\sqrt{2(n-s)\left[1 - \left(\frac{R_1 - R_2}{2}\right)\right]}}$$

where:

R_1 = Correlation coefficient for Engine A

R_2 = Correlation coefficient for Engine C

n = Number of observations

s = Number of variables

3. Are the coefficients of the lines of regression (b and c, e and f, h and i, or k and l) statistically similar for the two engines? To answer this question, it was necessary to determine the variance between the lines of regression for the two engines and the variance of these lines of regression from an overall line of regression. A typical analysis is given in Table 14.

In the above table the sums of the squares and the estimated mean squares are determined by usual statistical methods.

If F_c, defined by the equation below, is greater than F'_c (1,46) or F'_c (3,44), the difference between the slopes of the lines of regression for the two engines is significant. These values are given in Table 15.

$$F_c = \frac{E_D}{E_R}$$

4. Are the intercepts of the lines of regression

18 "Associated Measurements," by M. H. Quenouille. Pub. by Academic Press, Inc., New York, 1952.

Table 15—Significant Values Affecting F_c

Confidence Level	F'_c (1,46) for One Constituent	F'_c (3,44) for a Pair of Constituents
95%	4.05	2.82
99%	7.2	4.29

Table 16—Significant Values of t'_c (44)

Confidence Level	t'_c (44)
95%	2.02
99%	2.69

(*a*, *d*, *g*, or *j*) similar for the two engines? The test for this question can be made only if the answer to question 3 was affirmative, since the equation below is valid only for parallel lines of regression. The following formula is used to answer question 4:

$$t_c = \frac{\overline{X} - \overline{Y}}{\sqrt{E_R}}$$

where:

$\overline{X}$ = Mean odor intensity for Engine A
$\overline{Y}$ = Mean odor intensity for Engine C
E_R = Same as in Question 3.

If t_c is greater than the values of t'_c (44) given in Table 16, the difference between the intercepts of the lines of regression for the two engines is significant.

DISCUSSION

Observes Conclusions from Diesel Exhaust Odor Studies

—P. V. Garin
Southern Pacific Co.

AS major operators of diesel engines, the railroads are vitally interested in any research work relating to the characteristics and/or performance of this type of engine. We are particularly gratified to see such excellent basic research efforts as exemplified by this report on odors from diesel exhaust gas. As stated in the opening paragraph and as confirmed by Table 1 of the report, the diesel exhaust can be considered the safest of the exhausts issuing from the various types of internal-combustion engines in general use, although its halitosis, as it is called, has been objected to in some quarters.

There are a number of interesting observations and conclusions that can be drawn from this study which are of particular significance to the operation of railroad diesel engines.

1. Engine load has a pronounced effect on exhaust odor and irritation intensity. Part-load operation gives less gas than full-load operation.

2. Engine make, engine speed, engine condition, fuel type, and lubricating oil type have comparatively little effect, although the type of lubrication oil might affect the characteristics of the odor.

3. No significant difference was found in exhaust odor intensity with Fuel X having low end point (548 F) and intermediate cetane number (29.6). This fact is of particular interest to the railroads where the use of "economy" fuels with broadened specification requirements is being widely extended.

4. For a given design engine, odor is not necessarily related to exhaust smoke level provided injectors are in good condition. This fact was established by comparison of Engine A and Engine B, both of identical 2-cycle design. Engine B had been removed from commercial service because of noise, high oil consumption, and excessive smoke with about 300,000-miles total service and 100,000 miles from last overhaul. Engine A had only 300-hr service on the dynamometer stand.

Basic studies of this type will go far to alleviate and dispel the doubts that exist in the minds of many on the characteristics of diesel exhaust gases. The authors are to be commended on their excellent report on the techniques for evaluating diesel exhaust gas odor and on the interrelationships of odor with factors involved in diesel-engine operation and maintenance.

Wants Further Work On Correlation of Smoke, Odor Data

—P. H. Schweitzer
Pennsylvania State University

EVERYONE who has a concern for diesel engines or for air cleanliness on streets and highways—and this includes all of us—must welcome this report on the work done by two able investigators on diesel exhaust odor. Even if diesel exhaust is nontoxic, as we believe, people are so annoyed by it that it is beginning to take the fun out of motoring and the profit out of the automobile and oil companies.

Not having conducted tests on diesel exhaust odor, I cannot make very constructive comments. The absence of a proper instrument makes such investigation awkward. The authors have explored various avenues from the sniff box to chemical tests which promise enlightenment, and they registered some progress.

What I miss in this report is an attempt to correlate smoke and odor. I do not believe that the investigators failed to record smoke in their tests, but they perhaps did not find significant correlations to report.

Many of you have, no doubt, noticed that "cold" smoke is frequently accompanied by unpleasant acrid odor.[a] Cold smoke has light color and originates from fuel particles ignited too late or not at all. This frequently occurs during idling. Hot smoke, on the other hand, has dark color and originates from thermal decomposition of the fuel into carbon (soot) and the failure of the engine to burn the carbon before exhaust opens. Overload generally causes hot smoke.

It is hard to tell whether hot smoke is accompanied by odor or not. The reason it is hard to tell is because the exhaust smoke is mostly a mixture of hot smoke and cold smoke. It is easy to see if some dark smoke is mixed into white smoke, but less easy if white smoke is mixed into dark smoke. The authors found at full load odor intensities equal to or higher than those at idle. But how much "cold" smoke was present during full load, they, of course, don't know.

Filter-type smokemeters record the solid content of the smoke while the light extinction types, like the CRC-Photovolt smokemeter, respond also to the liquid fog which is the main constituent of the cold smoke. Therefore, a high reading on the CRC smokemeter together with a low reading on the Von Brand-type smokemeter would indicate the presence of cold smoke which may correlate with exhaust odor.

What we need is more complete analysis of the exhaust gases by physical and chemical means. Such a project is sponsored by CRC and is now under way.

After we have discovered means to measure exhaust odor it will be easier to get rid of it. Until then we are groping in the dark.

The authors found that exhaust odor intensity was lowest between ¼ and ¾ load. If you have an oversize engine in your bus you perhaps can avoid loading it higher than ¾, but this does not help you get rid of idling odor. On the other hand, if you had at idle or during deceleration fuel/air ratios, charge temperatures, and spray atomization equal to those at half load, you should not have smoke and odor any worse than at half load.

At idle, a diesel engine operates with about 600% excess air, and the charge temperature is pretty low. Spray atomization is also poor under idling conditions. Overlean mixture could be remedied by "recirculation" of the intake air as mentioned by Dr. List[b] or by the use of well matched turbochargers. Atomization

[a] SAE *Quarterly* Transactions, Vol. 1, 1947, pp. 476–489: "Must Diesel Engines Smoke?" by P. H. Schweitzer.

[b] See pp. 780–809 of this issue: "High-Speed, High-Output, Loop-Scavenged Two-Cycle Diesel Engines," by H. List.

could be improved by higher nozzle valve opening pressures. "Fumigation" of high cetane fuel or amyl nitrate into the intake air has relieved cold smoke and may relieve odor. These are all wildcat suggestions. We need more research of the type conducted by the authors to make the concentrated attack on exhaust odor; that is long overdue.

Discusses Toxic Effects Of Nitric Oxide in Diesel Exhaust Gas

—J. P. Ruth
The Ruth Co.

WHILE the title of this paper is restricted to "odor" the authors gratuitously drag in the statement in respect to diesel exhaust that it is "the safest exhaust gas from any internal-combustion engine then in existence." It is this statement that I shall comment on as, obviously, if the exhaust is dangerous to health, the removal of the "odor" may be undesirable, as the "odor" is a warning to those exposed to the exhaust.

The statement that diesel exhaust is the "safest exhaust gas from any internal-combustion engine then in existence" warrants an examination into the toxicity of both carbon monoxide and nitric oxide.

Diesel engines and spark-ignition engines both have exhausts which contain carbon monoxide and nitric oxide.

Diesel-engine exhaust contains practically no carbon monoxide but does contain relatively large amounts of nitric oxide. Gasoline or propane engines operating below the preignition level produce exhaust that contains practically no nitric oxide but substantial amounts of carbon monoxide. The contrast suggests that an investigation into the relative toxicity and toxic characteristics of the two gases is necessary to determine their relative "safety."

Carbon monoxide inhaled in high concentrations is deadly in a short space of time. Nitric oxide inhaled in lethal concentrations may not cause death for several days after the exposure. This is why nitric oxide is considered insidious.

Carbon monoxide is absorbed into the blood through the alveoli of the lungs 300 times faster than is oxygen. However, it does not injure the alveoli of the lungs when passing in and out. The alveoli of the lungs are that semipermeable membrane which permits the oxygen to enter the blood and which also permits the carbon dioxide to discharge from the blood into the lungs and thence exhaled into the atmosphere. Cigarette smokers who inhale have from 3 to 9% carbon monoxide in their blood, which shows that small quantities of carbon monoxide in the blood are not particularly injurious. Carbon monoxide causes death when the quantity in the blood increases the viscosity of the blood to a point where the heart cannot pump it through the arteries. Rigor mortis sets in belatedly, if at all, on carbon monoxide victims. This characteristic of carbon monoxide provides a positive means of pinpointing deaths caused by carbon monoxide.

To prove that carbon monoxide does not injure or destroy the tissue of the lungs, an English professor had an airtight cubicle built with a glass observation door and sat in this cubicle breathing concentrations of carbon monoxide gas until he was overcome. When he slumped unconscious the attendants outside withdrew him to fresh air. He recovered without noticeable injury or after effects, thus demonstrating to his own satisfaction and to others that carbon monoxide is not injurious to lung tissue. He repeated this experiment several times. This might be called a glorified snifter box, and perchance Mr. Rounds or Mr. Pearsall would like to make the same experiment with nitric oxide, breathing concentrations of air and nitric oxide in the manner of the English professor to back up their statements that the composition of diesel exhaust is the "safest exhaust from any internal-combustion engine then in existence."

M. B. Jacobs in his book, "The Analytical Chemistry of Industrial Poisons, Hazards, and Solvents" has this to say about nitric oxide:

"The fumes of the nitrogen oxides are extremely dangerous because of their insidious character. The bad feature of this type of poisoning is that little warning is given to the worker, for the oxides of nitrogen fail to set up defense respiratory reflexes. Thus a worker may inhale quantities of these gases that will affect him seriously and even cause death, without knowing it. Cases have been recorded where the workman felt entirely well throughout the working day after the inhalation of these fumes, only to die the following day because of pulmonary edema. Symptoms after exposure are restlessness with a dry cough and shortness of breath. These symptoms increase and are accompanied by a frothy sputum tinged with blood. Death may follow from pulmonary edema, even after several days.

"As little as 0.01% (100 ppm) may cause illness if breathed for a short time and 0.07%, or 700 ppm, is fatal if breathed for 30 min, or even a lesser time. The maximum concentration allowable for an exposure of several hours' duration is of the order of 10 ppm. The generally accepted maximum allowable concentration for daily 8-hr exposures is 25 parts of nitrogen oxides, other than nitrous oxide, per million parts of air. When the nitrogen oxides are calculated as NO_2, this concentration is equivalent to 0.047 mg per liter at 25 C and 760 mm."

Five years later in Chicago, Ill. on April 24–27, 1954, at the Transactions of the 16th Annual Meeting, American Conference of Governmental Industrial Hygienists, they had this to say about nitric oxide:

"It is recommended that this designation for mixtures of nitrogen oxides with a limit of 25 ppm be deleted from the present listing, and nitrogen dioxide (NO_2) be added with a value of 5 ppm.

"Nitrogen oxides should be revised downward from the past value of 25 to 5 ppm on the basis of the work reported by the Army Chemical Center [c] performed in 1951. Errors in appraising the true toxicity of nitrogen oxides appear to have occurred in the past because of failure to standardize on the most toxic component of nitrogen oxides, nitrogen dioxide. Nitrogen dioxide now appears to be a major contributor to the toxicity under most conditions where these gases are produced. The problem of setting the limits of this group of gases is a most difficult one because of the variation in amount, and types of nitrogen oxide gases produced. The M. A. C. Committee is currently submitting information urging a limit lowered to 5 ppm from the former value of 25 ppm for nitrogen dioxide with the further suggestion that future values be assigned to specific nitrogen oxides."

This body of experts sets the maximum allowable concentration of carbon monoxide that a workman may breathe during an 8-hr exposure at 100 ppm and recommends that the M.A.C. of nitrogen dioxide be set at 5 ppm for an 8-hour exposure. From this you will see that they consider nitric oxide to be 20 times more toxic than carbon monoxide. Of course the physiological effect of breathing nitrogen dioxide from diesel-bus exhaust is the same as from diesel exhaust in industrial plants, allowance being made for variations of intensity.

Many state authorities have accepted the 5 ppm figure as the M.A.C. for nitrogen dioxide as a standard where diesel power is used underground.

In many states where diesel power is used underground the mining authorities hold to the 5 ppm M.A.C. nitrogen dioxide in the mine air and permit 100 ppm of carbon monoxide. This shows that they consider nitric oxide to be 20 times more toxic than carbon monoxide.

Nitric oxide and nitrogen dioxide are called insidious because they injure the lungs by destroying the tissue without causing pain. Either of these two gases readily combine with the moisture of the lungs, forming nitric acid. Nitric acid burns, or destroys, the lung tissues in *any* concentration. The lung tissue, however, has the limited ability to neutralize these acid gases by forming nitrate salts from the body tissue; the body also has the ability to heal the slight burns caused by the acid gas on the alveoli. The question, of course, is, what is the M.A.C. of nitric oxide which the body can be depended upon to neutralize or heal, the burns of which result from breathing low concentration of this toxic gas over a 24-hr period 365 days per year, which is what the city dweller is confronted with?

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[c] Medical Division Research Report No. 52, Chemical Corps, Army Chemical Center, 1951, "Summary Report on Toxicity of Oxides from Red Fuming Nitric Acid," by E. LeB. Gray, J. K. MacNamee, and S. G. Goldberg.

(According to definition: nitrous oxide is N_2O, nitric oxide is NO, nitrous anhydride or nitrogen trioxide is N_2O_3, nitrogen tetroxide or dioxide is NO_2 or N_2O_4, and nitric anhydride or nitrogen pentoxide is N_2O_5.

Only the first four are gases, and of these only the NO_2 form of nitrogen tetroxide is colored the familiar brown-red. The other gases, including the N_2O_4 form of nitrogen tetroxide, is colorless or practically colorless.)

While death from carbon monoxide can be pinpointed, death from nitric oxides cannot. Nitric oxide acts on the lungs as does the pneumonia virus. Antibiotics which cure pneumonia, however, have no effect on nitric oxide victims, because their pneumonia has been caused by inorganic acid over a period of time and not by a virus that can be destroyed by antibiotics.

It is obvious, therefore, that breathing air containing carbon monoxide in concentrations up to and including 100 ppm will not injure an individual or the lung tissue any more than does breathing oxygen, carbon dioxide, or nitrogen.

Nitric oxide in diesel exhaust is oxidized by air into the anhydride, which forms nitric acid in the lungs. Whether just one molecule is inhaled, or whether it is breathed in concentration of 5 ppm, nitric oxide will destroy tissue, or neutralize the alkalinity of the body. If nitric oxide were a product of combustion as is carbon monoxide, all cigarette smokers would be dead in a short time.

Table 1 gives many analyses but without the variables or methods used to produce the test exhaust gases which were analyzed; they form no valid criterion by which fairly to judge the amount of toxic nitric oxide produced.

For example, if the material of which the exhaust valves were made in the engines tested, contains calcine oxides of such metals as cobalt, chrome, nickel, barium, magnesium, or lead, large quantities of nitric oxide would be found in the gasoline engine exhaust. Since these oxides act as catalytic agents and readily form nitric oxide at much lower temperatures than would ordinarily be required, the exhaust valves were made of simple steel.

The temperature created by compression ignition promotes the endothermal reaction which combines the free oxygen and free nitrogen after substantially all of the hydrogen and carbon of the fuel have been converted to carbon dioxide and water.

A 0.10% or 1000 ppm of nitrogen and oxygen combine in air at a temperature of 1227 C or 2246 F. At a temperature of 2827 C or 5126 F, 4.4% of nitric oxide is produced as a result of the heat. Temperatures as high as 2887 C are created in diesel combustion, but the opportunity for the excess oxygen to combine with the nitrogen comes only after the oxygen demand of the carbon and hydrogen in the fuel have been satisfied. As a result, therefore, it must combine in a highly diluted gas.

If the stoichometric fuel/air ratio of 14 to 1 could be used no nitric oxide would be formed. The excess of 20 to 1 fuel/air ratio necessary to oxidize all of the liberated carbon to a clear CO_2 exhaust provides the free oxygen that is later available to combine endothermally with the nitrogen.

If it were possible to use only the stoichometric fuel/air ratio and convert all of the carbon to carbon dioxide and the hydrogen to water, nitric oxide could not be formed, as both the CO_2 and H_2O reactions are exothermal, combining avidly the carbon and hydrogen with the oxygen, and there would be no remaining oxygen to combine endothermally with the nitrogen.

The temperature of diesel combustion would, with air alone, combine from 4% to 5% of the oxygen and nitrogen as nitric oxide. The gas products of combustion, CO_2 and H_2O, not only use up a substantial part of the original oxygen but they also dilute the remaining free oxygen and nitrogen to a point where only around 1000 ppm or 0.10% of nitric oxide forms in a temperature that is otherwise capable of combining around 4%.

Regarding nitric oxide in gasoline exhaust, the automobile passenger car or truck used and operated in cities produces practically no nitric oxide. Nitric oxide up to 5000 ppm can be produced in gasoline engine exhaust in a Delco knock-rating unit, but if a car engine were operated with such a knock, it would destroy itself in a short time. From the above, it is clear that while an ordinary automobile engine does not produce any nitric oxide, tests can be run on a gasoline engine and produce nitric oxide in the exhaust, which would be impossible in a practical operating engine.

Since diesel engines will burn and get the same high thermal efficiency from propane, gasoline, or heavy diesel fuel, it is important to know whether the tests mentioned in Table 1 cover analysis of diesel-engine exhaust burning the above-mentioned fuels. Obviously, the low carbon to hydrogen ratio in propane would greatly augment the formation of nitric oxide in diesel exhaust from engines using propane as fuel. The figures shown in Table 1 of the paper for oxides of nitrogen in gasoline and propane indicate that they are not the product of a normally operated passenger car fueled by ordinary gasoline or propane.

The problem of safety will not be solved by removing the odor, and leaving the toxic nitric oxide in diesel exhaust.

The solution lies in either of two directions: find a way to burn the hydrocarbon diesel fuel to a clear exhaust in a diesel engine with the exact amount of air to burn the carbon to CO_2 and the hydrogen to H_2O with no excess oxygen left over; or remove substantially all of the toxic nitric oxides from diesel exhaust before discharging it into city atmosphere and deionizing it so that it does not make aerosols from all solid particles that otherwise get into city atmosphere.

I have found that diesel engines put out around 10 cu ft of exhaust for each horsepower of output. The 200-hp engines used on most of our city buses will, therefore, put out under load around 2000 cu ft of exhaust per min. This exhaust will contain around 500 ppm of nitric oxide. In a city such as Los Angeles where 2000 buses operate and where they have huge stationary diesel engines within the city limits, to say nothing of the trucks and diesel locomotives, there is not enough air to dilute the nitric oxide produced by the intercity diesel power. *Smog News* of July 31, 1956, states that 9000 tons per day of nitric oxide pour into Los Angeles city atmosphere every 24 hr. The figures are given by the Territorial Department of Health of the Los Angeles County Air Pollution District.

In conclusion, it is submitted that the inclusion of the phrase, "safest exhaust gas" in the paper is unwarranted and could lead to serious misunderstandings by the uninitiated and that in any publication by our organization on the question of "odor of exhaust" the reader should be warned that "odor" and "safety" are, in this field, unrelated factors which should not be confused.

Wants Study of Aerosol Content in Exhaust Gases

—M. A. Elliott
Institute of Gas Technology

PROBLEMS connected with the odor of diesel exhaust gases have been recognized for some time, but there has been comparatively little work done in this field, and until the present paper was published there had never been a thorough and comprehensive study of the subject. Unless one has been directly connected with studies of the odor of diesel exhaust gases, it is difficult to appreciate the tremendous amount of effort that the present paper represents. Unfortunately, studies involving odor or irritation cannot be made without the expenditure of such effort because the only satisfactory measurement for these qualities is the human nose. As the authors have pointed out, in order to obtain definitive results a comparatively large number of individuals must be used as observers, and a comparatively large number of tests are required. Unfortunately there appears to be no alternative because the paper clearly shows that it is not possible to use chemical analysis as a means for uniquely characterizing odor. However, in demonstrating this point the authors have contributed much provocative information which leads to a much better understanding of the odor problem.

When we are faced with assigning a quantitative value to such an intangible thing as odor or irritation, it is difficult for us sometimes to realize that such measurements can be made with any degree of reproducibility. The authors have clearly demonstrated that this is possible. Furthermore, it is of considerable interest to me to compare some of their findings with the findings of other investigators. For example, their data show that the threshold of irritation is at a concentration seven times that of the threshold for odor. A comparable value from studies made by the Bureau of Mines was four times. For work of this type in two widely different laboratories under widely different conditions and with different observers, this is certainly a reasonable check. It is of interest to note also that even though no strong correlation was found between concentrations of aldehydes and odor intensity, the results obained by General Motors, the Texas Co., and the Bureau of Mines indicated the same order of magnitude of odor in the range of 1 to 100 parts formaldehyde.

One of the most significant findings in the paper is the removal of an appreciable fraction of the odor by filtration. The authors observe, "these filtration experiments indicate strongly that aerosols and/or particulate matter in exhaust gas contribute significantly to diesel odor and irritation." The importance of this observation cannot be too strongly emphasized. Aerosols are known to adsorb gases on their surface. When these aerosols impinge on membranes in the respiratory system, comparatively high local concentrations of the adsorbed materials will result. Thus it is conceivable that a low concentration of an odorant or irritant in combination with a high-concentration aerosol might be significantly worse than a high concentration of odorant or irritant in the absence of aerosol. It would certainly seem that this effect of aerosol content might account for some of the failure to observe a strong correlation between odor and any of the chemical constituents. This suggests a whole new field of very difficult research and one in which comparatively little information has been obtained, namely, a study of the aerosol content in exhaust gases. Such studies are assuming increasing importance in connection with air pollution problems.

The thoroughness and objectiveness with which the authors have approached this problem along with the information made available constitute a major contribution to the knowledge in this field, and they have opened up additional possibilities for future research to obtain a better understanding of the phenomena involved.

Wants Further Checks On Noxious Exhaust Gases

—W. M. Keller
Association of American Railroads

AS this paper shows, the use of sensory organs as a measuring device has a tendency to widen the band of variations and increase the scatter on curve evaluations of diesel exhaust gases. There are standards of length, mass and various other comparisons, but there are no standards on sensory perception of odor which can be kept in a glass case and used to check odors. The use of human senses as measuring devices is subject to error. We may some day have a standard nose, but in its absence it appears that the authors of the paper have provided a fairly acceptable stopgap. Indeed, it would appear that the paper offers a good foundation upon which to build future technology on this subject.

The diesel engine needs a press agent to extol its virtues, as the popular conception of its exhaust gases is not in agreement with the facts. The "halitosis of the diesel" may cloud its real virtues as one of the best engines in existence, in so far as its sparing output of noxious gases is concerned.

Carbon monoxide in diesel gases is a rare constituent in any appreciable quantity. The steam locomotive was a much more prolific producer of carbon monoxide than the diesel. The private dwelling house was a still worse offender, and because there were so many houses, the real atmospheric pollution in most towns and cities could be accounted for more from private dwellings than from steam locomotives. But because odor was less distinctive in the gas discharge from houses the question of its contents received less attention.

With the use of diesel locomotives, the improvement, so far as carbon monoxide is concerned, was spectacular. Other noxious gases were, of course, at an acceptably low level. It appears that the authors have stressed vital points in comparing odor with the chemical composition of gases. It is interesting to note that the formaldehyde contributes more to irritation than to odor. Reference to Fig. 7 does show that odor intensity increases with formaldehyde percentage. This situation, however, can be incidental and does not necessarily indicate that the formaldehyde content is responsible for the increase in odor intensity.

It is logical to assume that the rate of engine power output has a pronounced effect on odor and irritation intensity. Since the characteristics of engines are such that the amount of air used per pound of fuel diminishes with increased engine load, it would be expected that the undesirable effects of odor and irritation would increase, first because there is less air for combustion and secondly, because the combustion gases receive less dilution. Reference to Table 1 shows that, almost without exception, the diesel engine is vastly superior to gasoline- and propane-fueled engines since the characteristics of the exhaust show the undesirable gases are greatly reduced.

The need for keeping the sampling apparatus clean, including the piping to and from the sniff box, has been stressed in the paper. This is important since odor-producing particles have a tendency to cling to surfaces, and unless clean surfaces are present a large degree of error could be introduced. As an example of the tendency of odors to become semipermanently resident, it is interesting that piping used for public utility gas will retain the odor of the material used as a warning for escaping gas for many months after the piping is removed from its position as a gas conductor. So much so, that it is difficult to wash out the odor unless something other than water is used.

The use of filters for removing odorous materials from exhaust gases appears to be completely impractical, even for a vehicle such as a bus. Certainly the size of the filter required for a 2000-hp diesel unit would be such that there would be no place to put it on diesel locomotives. Moreover, much maintenance expense would be involved.

The conclusion of the authors that a practical solution of the chemical destruction approach seems unlikely, appears to be completely justified. The chemicals used would require frequent renewal and the expense incident to this work would reduce the economic advantage of the diesel. In the railroad industry means of reducing diesel operating costs are under investigation, and the addition of expensive equipment that serves little or no purpose would not be welcome. A railroad diesel locomotive discharges its exhaust gases upward from a point about 15 ft from the ground, whereas most buses discharge the exhaust gases in a horizontal plane a foot or two from the road or street. It is therefore evident that this type of vehicle is more vulnerable to criticism as a result of gas odor.

There is a phase of this subject, which may require further study to evaluate its significance, if any, in its importance to the entire parameters under discussion. It is a common human experience to enter areas having odors and find that after remaining in the area over an extended period of time perception of the odor disappears. This phenomenon may be described as sensory fatigue. While the use of the sniff box prevents this factor from introducing serious error into laboratory data, it should be remembered that if field tests were conducted in the open atmosphere this factor should be analyzed. Perhaps in this case the sniff box procedure should be reversed and fresh air introduced in the sniff box from a source such as compressed air cylinders. This problem is more relevant when considering the odor evaluation by dilution method.

Authors' Closure To Discussion

WE would like to thank the discussors for their remarks. We should probably have asked Mr. Garin to write the summary for our paper, since he did an excellent job in his remarks.

With respect to Prof. Schweitzer's comments concerning the relationship between smoke and odor, no measurement of exhaust smoke other than visual observation of the exhaust stack was made. Based on observations of exhaust gases from Engine A (no visible exhaust smoke except at full load) and Engine B (easily visible exhaust smoke at all loads greater than half load), we concluded that there was no correlation between visible smoke and odor. However, Prof. Schweitzer's suggestion that relatively invisible "cold" smoke may be related to odor is interesting and certainly warrants investigation. Such a study should include as an essential part an investigation of the role of aerosols in diesel exhaust odor as recommended by Dr. Elliott.

We have made limited field evaluations of diesel odor as suggested by Mr. Keller but have found such tests to be less reliable than laboratory measurements. The greater variability results not only from sensory fatigue as mentioned by Mr. Keller, but also inability to supply uniform samples to all observers due to variabilities in such things as wind velocity and wind direction. For dilution measurements, reasonably odor-free air was obtainable by passing air through a small activated charcoal column and a fine paper filter.

THIS paper discusses the free-piston engine which is being investigated by Ford.

The analytic method for calculating free-piston-gasifier thermodynamic performance over the complete operating range is discussed for the given limitations and conditions.

Using these analytical methods, an automotive-size free-piston and matched turbine was designed, built, and tested.

THE AUTOMOTIVE

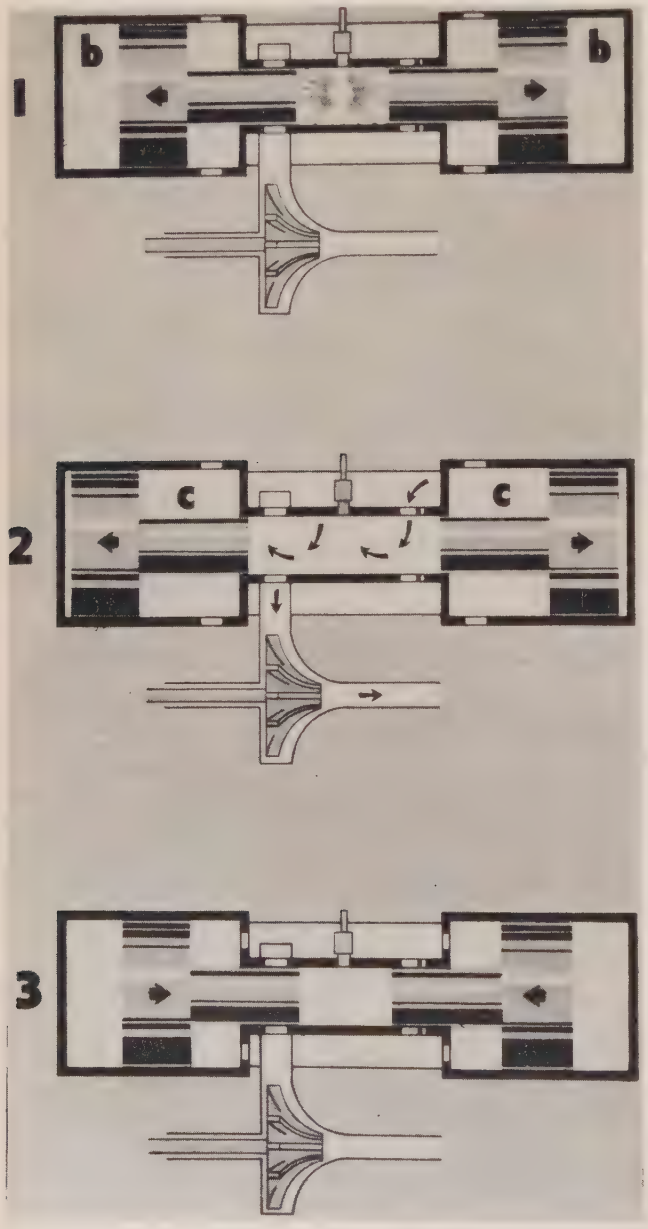

Fig. 1—Schematic diagram of operation of inboard compression, free-piston gasifier

THE free-piston engine is one of the many kinds of advanced engines being investigated today as part of the important and continuing search for better automotive engines. We believe that for any new kind of automotive engine to supplant in significant measure the present piston engine it must

1. Weigh less.
2. Occupy less space.
3. Cost less.
4. Operate more economically.
5. Give better vehicle performance.

Not one of these qualitative factors can be ignored, in our opinion, when judging the possible success of a new kind of engine. In addition, the present-day piston engine will be constantly improved in regard to all of the above factors, and so it presents a moving target that is always moving in such a way as to make it more difficult for engine substitution.

All this means that the search for a new kind of automotive engine is not easy, and extravagant claims of substitution coming tomorrow can probably be discounted.

Ford Motor Co. has been seriously investigating the regenerative gas turbine for some time. We believe that this kind of gas turbine has a future provided certain rather fundamental problems can be overcome. At the present time, we have found that the engine has no outstanding weight advantage over present-day piston engines, has borderline fuel economy particularly at part load, and has very bad throttle or acceleration response in the sizes needed for present-day passenger cars. This poor acceleration response arises from the compressor compressor-turbine inertia.

FREE-PISTON-TURBINE ENGINE

Donald N. Frey, Paul Klotsch, and Adolph Egli, Ford Motor Co.

This paper was presented at a Detroit Section Meeting, Detroit, Oct. 8, 1956.

It is not my intention to discount the regenerative gas turbine. Rather it is to portray some of the reasons why we felt that investigation of the free-piston-turbine engine was in order. In fact, I am sure that many kinds of engines will be considered before deciding upon which horse to back in the future engine race.

General Characteristics

Fig. 1 shows schematically the operation of an inboard compression, free-piston gasifier.

The substitution of a free-piston gasifier for the rotary compressor and compressor turbine of a gas-turbine powerplant has several immediately known advantages. These follow from the following general characteristics: less alloy content than a gas turbine; fuel economy slightly better than a conventional 9.5/1 piston engine, and much better than a regenerative gas turbine, particularly at part load; throttle response about the same as a piston engine equipped with a torque converter. Also, free-piston engines can be adapted to split installations in passenger cars with the gasifier in front and the power turbine in the rear. This eliminates the driveline tunnel so prominent in today's low cars, yet gives a good weight distribution. The free-piston-turbine, however, weighs something more than 3 lb per hp at the present time, more than a comparable piston engine. Further, the full-load life of the free-piston gasifier at design weights of 3 lb per hp remains uncertain.

From these characteristics, one can see that the free-piston-turbine engine overcomes some of the objections to today's regenerative gas turbine but leaves unsolved the problem of weight.

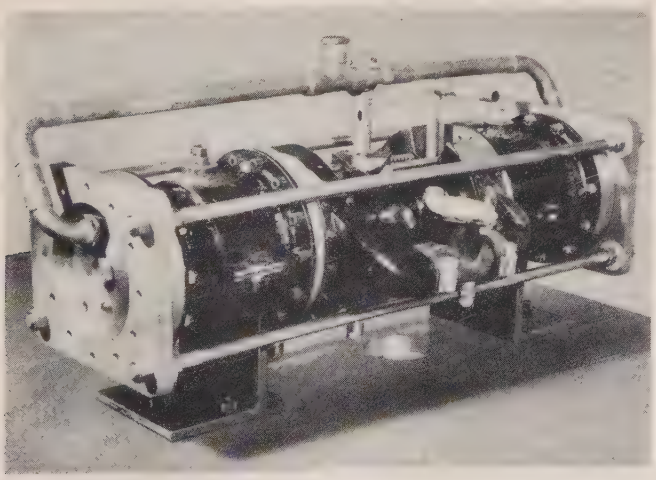

Fig. 2—Small inboard compression gasifier

Analysis of Gasifier Performance

Several phases have been accomplished in our work on the free-piston-turbine engine. We have already reported briefly on some of the results of working with a very small (10 gas hp) inboard-compression gasifier as shown on Fig. 2. Another important phase has been the establishment of an analytical method for calculating free-piston-gasifier thermodynamic performance over the complete operating range.

In order to do this, certain assumptions must be made regarding the thermodynamic processes involved. Typical assumptions used for the work done in each section of an inboard compression machine

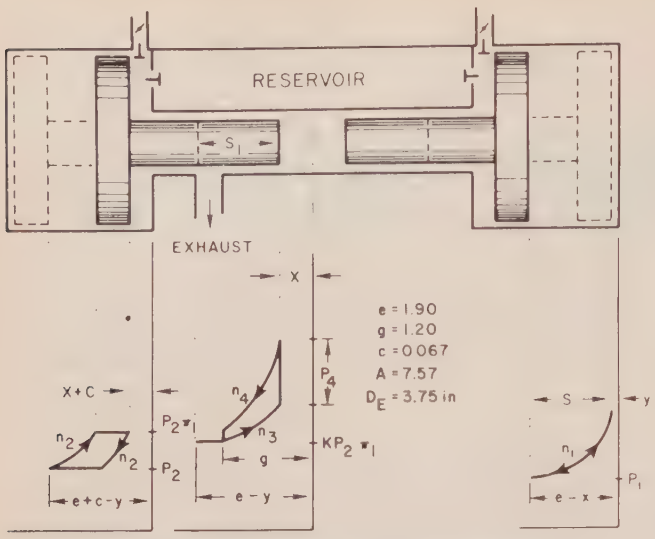

Fig. 3—Free-piston-gasifier cycle diagram

Table 1—Inboard Equation

$$\frac{1}{n_1-1}P_1(A+1)(e-x)\left[\left(\frac{e-x}{y}\right)^{n_1-1}-1\right]$$

$$-P_2A\left[\frac{(e+c-y)}{n_2-1}\left\{n_2\pi_1^{\frac{n_2-1}{n_2}}-1\right\}-\pi_1(x+c)\right]$$

$$-\pi_2\left[e-y-g+\frac{g}{n_3-1}\left\{\left(\frac{g}{x}\right)^{n_3-1}-1\right\}\right]-P_3(e-x-y)=0$$

Table 2—Outboard Equation

$$\frac{1}{n_1-1}P_1(A+1)(e-x)\left[\left(\frac{e-x}{y}\right)^{n_1-1}-1\right]$$

$$-P_2A\left[\frac{\pi_1(x+c)}{n_2-1}\left\{1-n_2\pi_1^{\frac{1-n_2}{n_2}}\right\}+(e+c-y)\right]$$

$$-\pi_2\left[e-y-g-\frac{1}{n_1-1}\left\{g\pi_2\left(\frac{g}{x}\right)^{n_3-1}+\frac{R}{C_v}F\right\}\left\{1-\left(\frac{g}{x}\right)^{1-n_1}\right\}\right.$$

$$+P_3(e-x-y)=0$$

with corresponding cycle diagrams (Fig. 3) are as follows:

1. The compression and expansion processes are assumed to follow the adiabatic law, PV^n = constant, and no energy loss is considered.

2. The reversible cycle of the compressor is assumed with polytropic compression and expansion; constant temperature and pressure intake and exhaust processes are used for simplicity.

3. Polytropic compressions are again used for the engine cycles, and it is assumed that the combustion and exhaust pressure changes occur at constant volume. Only a fraction, α, of the total available heat is assumed introduced at constant volume; the remainder is assumed burned during expansion. The scavenging process is assumed to take place at constant temperature and pressure.

Friction is assumed a constant coulomb-type force, always opposing the piston motion. It is expressed as a piston pressure.

With these assumptions, the energy balance equation for inboard and outboard strokes of a free-piston gasifier can be written as shown in Tables 1 and 2. Dimensionless form is used for generality, and the notation is as seen in Table 3.

One of the unique features of a free-piston machine is the absence of any fixed stroke or piston position at inner and outer dead-center. The position of these end-points is determined by the two energy balances. The piston will stop, for instance, on its outboard stroke only when the energy stored in the bounce is equal to the work delivered by the engine expansion minus both the compressor work and the friction work during that stroke.

The energy equations shown are nonlinear in the two basic unknowns x and y and, therefore, an iterative solution is required.

To determine the complete operating characteristics of a gasifier by the method outlined, a large amount of computation is required. Further, it is frequently necessary to compare the operating characteristics of gasifiers with different geometrical proportions to arrive at an optimum design. For these reasons and the fact that the solutions of the energy equations cannot be written explicitly, all the computations have been programmed on a high-speed digital computer. The results discussed here are all from machine computation.

Certain further limitations must also be imposed on the solutions to insure that the operating point is physically realizable. For instance:

1. On each stroke, the inlet ports must open and admit a sufficient quantity of air to secure scavenging and reasonable fuel/air ratio.

Table 3—Notation Equations (1) and (2)

Cycle Variables

Polytropic Exponents
- Bounce, n_1
- Compressor, n_2
- Engine compression, n_3
- Engine expansion, n_4

Pressures (in Atmosphere)
- Minimum bounce pressure, P_1
- Compressor inlet pressure, P_2
- Equivalent friction pressure, P_3
- Pressure rise due to heat addition at constant volume, P_4

Pressure Ratios
- Inside compressor, π_1
- Available to turbine, $\pi_2 = k\,P_2\,\pi_1$
  k = Product of exhaust valve and engine port percentage flow losses

Piston End Points

$$\frac{\text{Engine clearance at inner dead-center}}{\text{Engine bore}}=\frac{x_1}{D_E}=x$$

$$\frac{\text{Bounce clearance at outer dead-center}}{\text{Engine bore}}=\frac{y_1}{D_E}=y$$

$$\text{Volumetric engine compression ratio}=\frac{g}{x}$$

Fuel Addition
- Heat added by fuel, Q
- Nondimensional fuel quantity per stroke, $F=\dfrac{F_1}{D_E}=\dfrac{\alpha Q}{P_{atm}A_E D_E}=x\dfrac{C_v}{R}P_4$

$$\frac{\text{Compressor cross-sectional area}}{\text{Engine cross-sectional area}}=\frac{A_c}{A_E}=A$$

$$\frac{\text{Gasifier length}-\text{piston length}}{\text{Engine bore}}=\frac{e_1}{D_E}=e$$

$$\frac{\text{Distance from center line to exhaust port edge}}{\text{Engine bore}}=\frac{g_1}{D_E}=g$$

$$\frac{\text{Compressor clearance}-\text{engine clearance}}{\text{Engine bore}}=\frac{C_1}{D_E}=C$$

2. Maximum pressures and minimum clearances must be set to insure a reasonable design.

3. The operating point must be dynamically stable; that is, if a small short-duration disturbance should occur, the gasifier must return to its original operating point.

Particular attention has to be paid to stability criteria. More than one combination of x and y is frequently a solution for the two energy equations with a given set of the independent variables. The single dynamically stable solution can be determined from an examination of tangents at all the intersections of the inboard and outboard energy equations in the xy field. One can show that a dynamically stable point is the one for which

$$\left(\frac{dx}{dy}\right)_{\text{inboard}} < \left(\frac{dx}{dy}\right)_{\text{outboard}}$$

Once the simultaneous solution of the inboard and outboard energy equation has been found, subject to limitations covered above, it is possible to compute any desired engine characteristic, such as airflow, exhaust temperature, gas horsepower, specific fuel consumption, or cyclic frequency. For calculation of frequency, it was found that a simple equation

$$f = K \frac{V_{\max}}{\text{stroke}}$$

was an accurate representation of piston frequency f, in terms of maximum piston velocity $V_{\max}$, piston stroke, and an engine constant K.

Another useful interpretation of the energy equation solutions is to extract the basic engine control quantities for each desirable engine operating point. These basic control quantities are:

1. Quantity of fuel injected per stroke.
2. Minimum pressure level in bounce cylinder.
3. Compressor inlet pressure.
4. Compressor pressure ratio.

From the analytical program, several general rules emerge. For example, frequency and gas horsepower increase, and specific fuel consumption decreases when the level of any of the control quantities is raised. Also, the airflow rate increases and the turbine inlet temperature decreases whenever the fuel quantity or bounce pressure is raised or the inlet pressure is lowered.

Table 4 shows a set of coefficients and geometrical quantities for a particular free-piston gasifier whose design will be discussed shortly. With these quantities, complete operational maps have been computed.

Fig. 4 shows turbine inlet temperature versus airflow for various turbine pressure ratios. Each point represents a solution of the two energy balance equations for a given value of each of the control quantities. Lines of increasing fuel quantity are drawn connecting the points. The figure also illlustrates another unique feature of the free-piston machine: the same operating point may be reached with quite different values of the control variables.

The points on Fig. 4, then, define a region within which the gasifier may operate. The boundaries of this region are determined by the physical limitations mentioned earlier and by the fact that solutions to the two energy balance equations exist only in certain regions. By combining the several such regions, a complete operational field may be determined for a gasifier.

Fig. 5 shows this field presented in terms most

Table 4—Numerical Example of Free-Piston-Gasifier Design

Polytropic Exponents
- Bounce, $n_1 = 1.40$
- Compressor, $n_2 = 1.35$
- Engine compression, $n_3 = 1.30$
- Engine expansion, $n_4 = 1.24$

Equivalent Friction Pressure, $P_3 = 0.6$

Flow Loss Constant, $K = 0.855$

Geometry
- $A = 7.57$
- $e = 1.90$
- $g = 1.20$
- $C = 0.067$

Engine Bore, $D_E = 3.75$ in.

Total Weight of Oscillating Parts is 100 lb

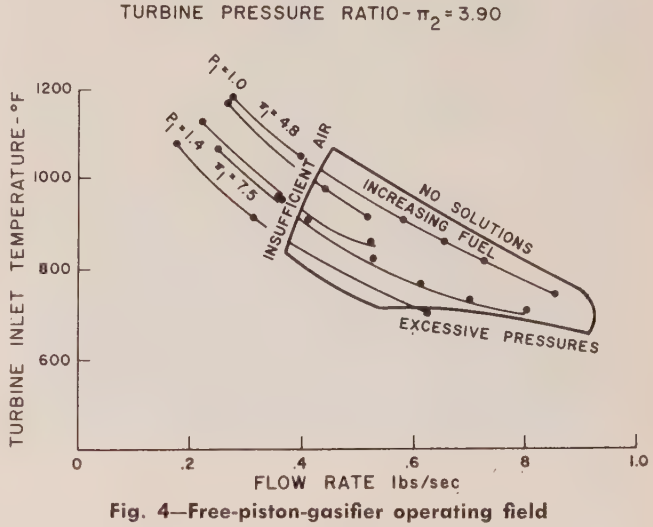

Fig. 4—Free-piston-gasifier operating field

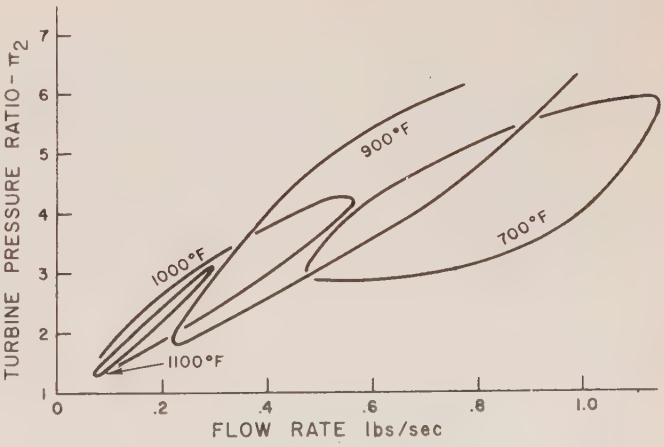

Fig. 5—Best free-piston-gasifier operating field for matching turbine and gasifier

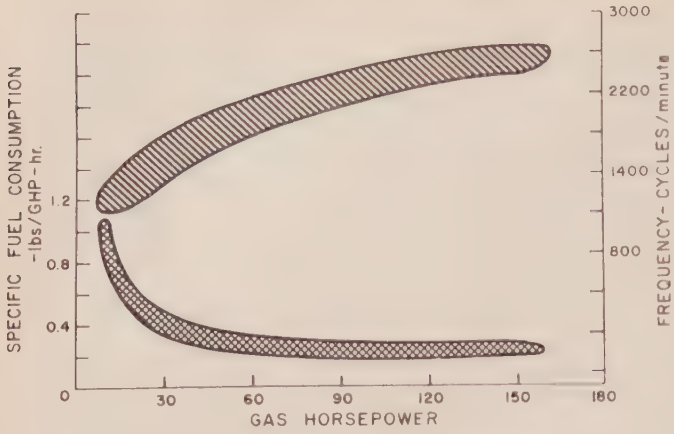

Fig. 6—Free-piston-gasifier performance

useful for matching turbine and gasifier. Each temperature line encloses a region within which the gasifier may operate with suitable choice of control variables and yields an exhaust gas of the temperature indicated.

Fig. 6 shows in still another fashion, output quantities as a band defining regions within which the gasifier may operate. Specific fuel consumption and frequency are shown as functions of gas horsepower output. Note also the wide gas power output range. Until quite recently, it was not thought possible to design free-piston gasifiers to operate stably over the wide output range shown.

The analytical methods discussed lead directly to an optimum design of a free-piston gasifier. They also portray, as in Fig. 5, the predicted output quantities upon which a suitably matched turbine design may be predicated.

Design and Operation of Free-Piston Gasifier

As the next phase in our work on free-piston-turbine engines, we designed and built an automotive-size free-piston gasifier and matched turbine. Performance prediction, optimization, and matching were done using the analytical methods outlined above.

The gasifier is an inboard compression machine using a 2-stroke uniflow-scavenged, compression-ignition power cylinder. The bore of the power cylinder is 3.75 in., minimum effective piston stroke (one side) is 4.2 in., and maximum piston stroke is 6.8 in. Compressor bore is 11.0 in. This gasifier is capable of about 150 gas hp at a discharge pressure ratio of 3.6 and a cyclic speed of 2400 cpm.

Rather conventional reed-type valves are used on the compressor intakes and exhausts, and a rack and pinion piston synchronizing system is used. The fuel pump initially installed was driven from a translating cam mounted on the top side of one of the piston synchronizing racks. The pump formed part of a jerk-type fuel injection system. Fig. 7 shows the partially assembled engine and illustrates these features. The pistons are supported internally in their reciprocating motion on brass guides which are press-fitted into the bounce cylinder heads.

An air starting system was incorporated initially, using air discharged rapidly through two rapid opening valves into the bounce chamber with the pistons in the outboard position. The pistons are moved to their outboard positions preparatory to starting, with a small vacuum pump evacuating the bounce cylinder. Fig. 8 shows the assembled gasifier on the test stand.

Gasifier Development Difficulties

A brief description of some of the development problems of this gasifier is worth while.

Difficulties were encountered with the following items:

1. Starting.
2. Power cylinder piston rings.
3. Injection pump cam failures.
4. Poor combustion.

Initial starting difficulties were traced to the entrapment of too much starting air in the bounce cylinder. An initial compression ignition could be obtained, but with the excess starting air still in the bounce, the piston outboard stroke was too short, and the scavenging ports could not be opened.

The first change made was to add ports to the hollow bronze piston guide. As the piston moved inboard, these ports uncovered and allowed air to flow from the bounce chambers. These ports subsequently became the bounce pressure control ports. It was also determined from simultaneous oscillographic recording of the bounce pressure time curves

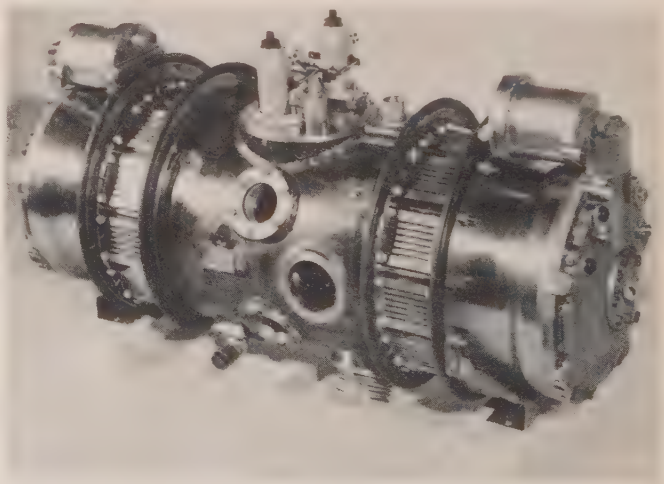

Fig. 7—Partially assembled gasifier

during starting that the bounce pressure rise at start was not very steep, and the two bounce pressure curves were not in phase. New starting valves were designed which gave an improved bounce pressure rise rate, and by triggering the starting valves with a pair of pilot valves synchronism was obtained.

These changes solved the starting problem. The engine starts very reliably within the normal ambient range.

Power-cylinder piston-ring failures by both breaking and burning were traced to extremely high combustion pressures of over 4000 psi and inadequate ring materials. Substitution of steel for the initial cast-iron rings resulted in considerable improvement in ring life. Chromium-plated power-cylinder bores are currently used. High combustion pressures were believed due to early injection, and a delay line was added between the pump and injector. This led to an immediate reduction in combustion pressures and consequent improvement in ring life. Subsequent recording simultaneously of the fuel injection pressure pulse, correlated with the fuel spray, and the combustion pressure time indicates near-optimum delay and consequent optimum timing as shown in Fig. 9.

A further word on the use of a delay line is in order. A peculiar problem of the free-piston engine is that, since there is no continuously rotating crankshaft, there is no means for actuating an injection delivery pump through the inner dead point of the piston motion. In fact, the piston motion slows down and stops at about the same time the fuel injection stroke should start. Use of delay lines enables this problem to be overcome without too early combustion.

Injection pump cam failures have been a problem, although progress in terms of increasing cam life is being made. The problem is now one of materials development.

It is easy to see why cam failures occur. It is characteristic of jerk-pump injection systems that rate of

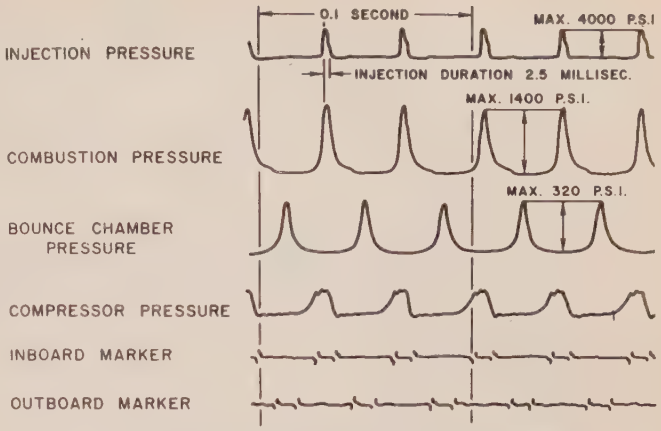

Fig. 9—Oscillograph recording of free-piston-gasifier cycle events

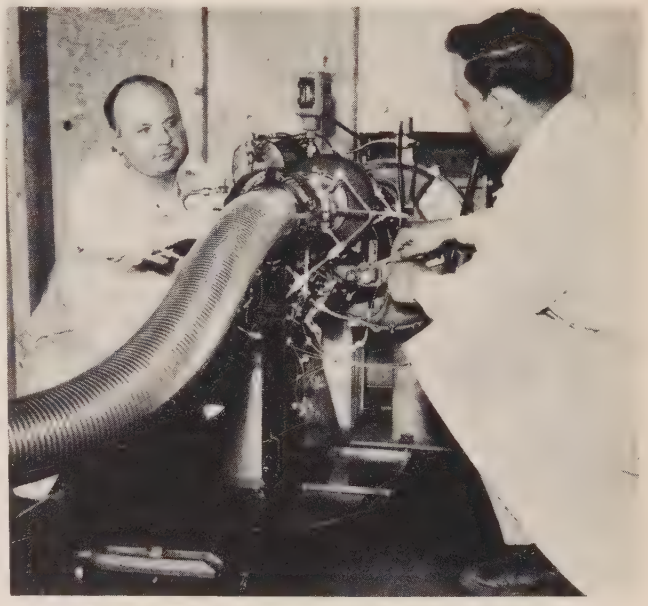

Fig. 10—Turbine with 160-gas-hp gasifier added

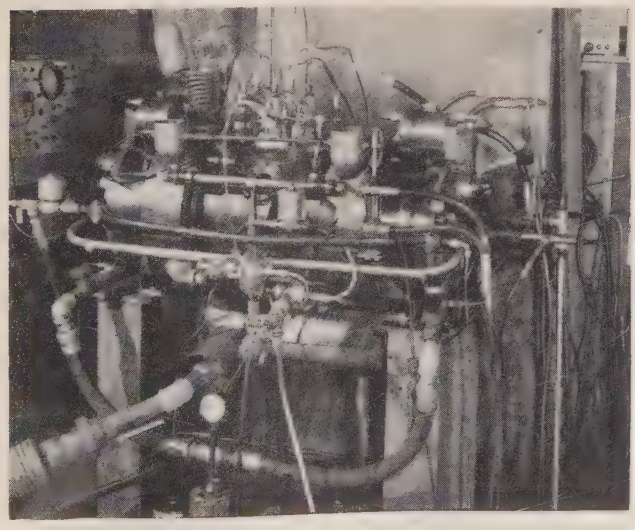

Fig. 8—Assembled gasifier

cam lift is high. The cams used in the injection system described here provide 0.315-in. lift in 2.40 in. of travel. For the cyclic speeds used (up to 2400 cpm), this gives for any usable cam profile accelerations of the order of 2000 *g*.

The original free-piston-gasifier design used only one cam and associated pump. In an effort to reduce the cam loading and to improve combustion efficiency, two somewhat smaller pumps were installed with a cam on each synchronizing rod.

Poor combustion efficiency was evident when the gasifier first became operable. Some of this was due to excess lubricating oil in the combustion cylinder, but a great deal of partially burned fuel was running through the engine. The latter also caused fires to start in the surge tank ahead of the turbine.

Upon installation of the two injector pump systems,

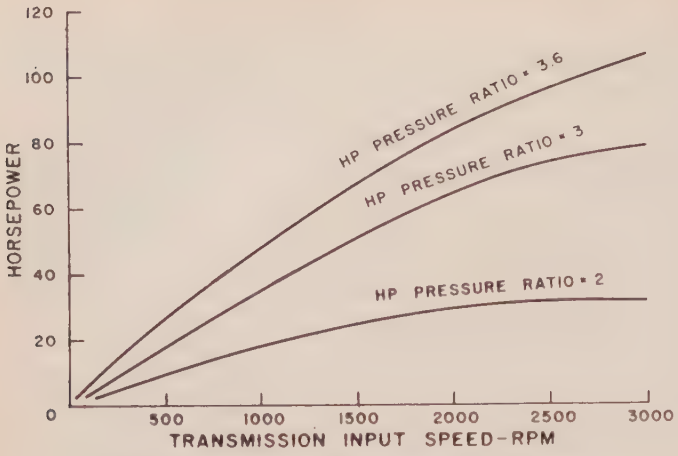

Fig. 11—Horsepower performance of turbine

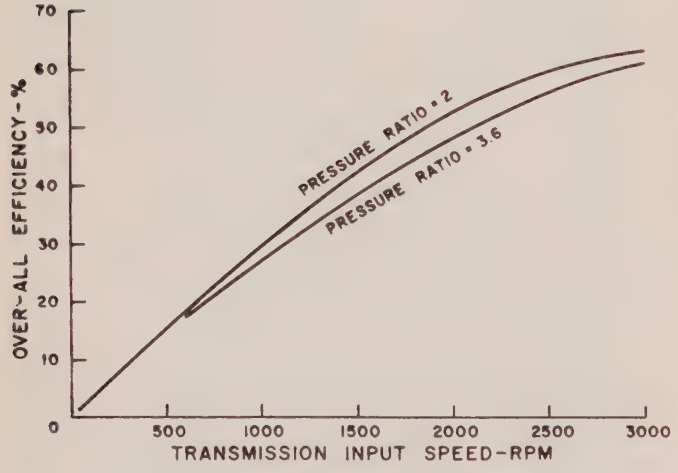

Fig. 12—Overall efficiency of turbine

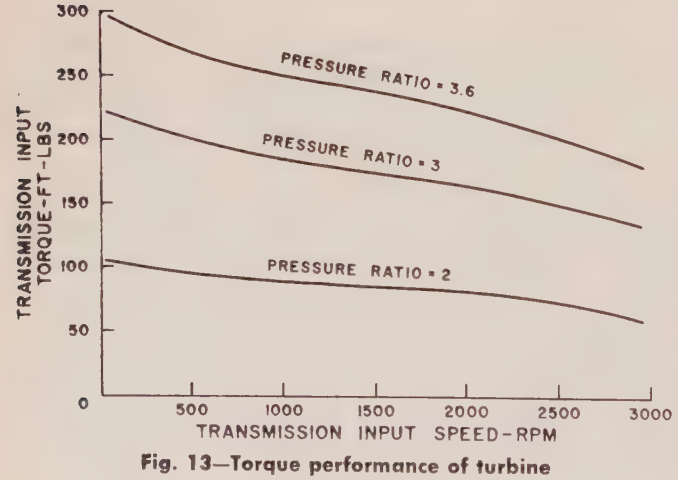

Fig. 13—Torque performance of turbine

substantial improvement in combustion was obtained. The exhaust was cleaned up to the point of equivalence with any diesel engine running in good trim.

The use of two injectors per cylinder in the bore size considered here is not unusual. We hope, however, eventually to develop a satisfactory single injector for its obvious cost advantages.

The turbine used with the gasifier is of the radial-inflow type. At present this type of turbine has a poorer efficiency range than the multistage axial-flow type, but it has the advantages of simplicity, small size, and relatively low manufacturing cost.

Fig. 10 shows the turbine used with the 160-gas-hp gasifier described in this paper. Figs. 11, 12, and 13 show the performance of this turbine. Note particularly the favorable torque curve of Fig. 13 which is characteristic of gas-turbine drives.

Earlier, an analytical method for performance prediction was presented. The free-piston gasifier subsequently built is being used to check the accuracy of this analytical method. To date, the points experimentally determined verify very well the cyclic rate versus power predictions found on Fig. 6. Low end fuel economy values have been roughly verified, but high end data are incomplete. Also, the wide range of gas horsepower output predicted as being stable has been well verified. This latter finding is particularly pleasing since it was unknown before the present work.

Conclusions

1. The continuing search for an eventual successor to the automotive piston engine will cover many kinds of engines besides the regenerative gas turbine.

2. To be successful the regenerative gas turbine must overcome several major disadvantages. It has, at present, no weight advantage over present-day 9.5/1 piston engines. It has a much poorer throttle response and a borderline fuel consumption. The free-piston gasifier appears to be a possible alternative to the compressor and compressor turbine of the gas-turbine engine.

3. The free-piston-turbine engine is a possible contender for the engine of the future because it does not suffer by comparison to the piston engine in regard to throttle response, fuel economy, or cost of manufacture. It has a superior torque curve and allows for split passenger vehicle installation with important vehicle advantages. Currently, it has a serious weight disadvantage.

4. The free-piston-turbine engine must develop, if successful, into a machine of much higher gasifier cyclic speeds than the 2400 cpm presently attainable.

5. An analytical method for predicting gasifier performance has been established, and its accuracy to date appears good. However, further correlation is necessary.

Radioactive Tracer Measurements of Engine Bearing Wear

M. W. Savage and L. O. Bowman,
California Research Corp.

This paper was presented at the SAE National Fuels and Lubricants Meeting, Tulsa, Nov. 8, 1956.

CONNECTING-ROD bearing wear occurs to some extent even under ideal lubrication conditions. The purpose of this paper is to show and explain some of the effects of operating variables and lubricating oils on connecting-rod bearing wear.

Bearing failures in the field are due mostly to dirt, misassembly, misalignment, and fatigue. Normal rubbing wear, defined as the wear ocurring when the oil film is not adequate to prevent bearing contact, also contributes to bearing failures but to a lesser degree. This paper deals generally with rubbing wear which bearing and lubricating oil designers are constantly striving to reduce.

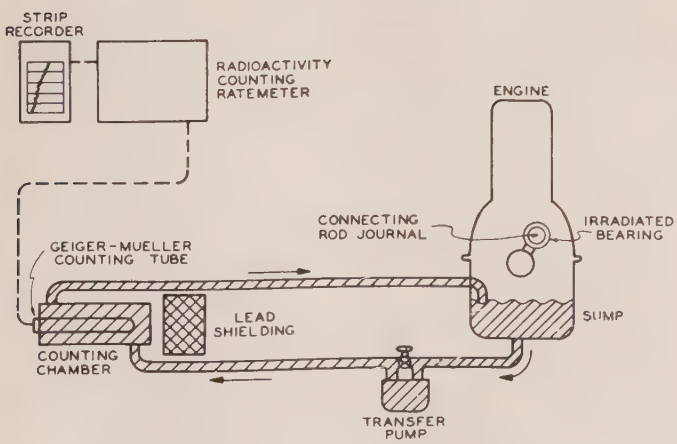

Fig. 1—Flow diagram for bearing wear test apparatus

A method has been developed by the authors using a radioactive tracer technique to measure bearing wear as it occurs. Either babbitt bearings or hard alloy bearings overlayed with babbitt are generally used in automobiles, and both have been utilized in our test program. Babbitt compositions are predominantly tin or lead, with small amounts of antimony and occasionally copper added. This investigation has been confined to the lead-base babbitt material because that type appears to be in most common usage at the present time. The uniformly dispersed antimony is the only element in babbitt alloys that retains radiation energy long enough to permit engine testing, and bearings containing 2–8% antimony have been successfully used.

Bearings are placed in an Atomic Energy Commission reactor at Oak Ridge, whereupon the radioactive isotope antimony-124 is produced. This isotope is excellent for wear testing because of its long half-life of 60 days and its high radiation energy.

Fig. 1 shows schematically a flow diagram of the test apparatus. An irradiated bearing is installed on a connecting-rod journal near the center of a multicylinder engine. A middle journal is used in order to reduce radiation hazard to the front and rear of the engine. The wear particles from the irradiated bearing accumulate in the lubricating oil which is continuously circulated by the transfer pump through

A RADIOACTIVE tracer method for measuring connecting-rod bearing wear as it occurs is presented.

A bearing wear mechanism is proposed which is substantiated by wear measurements using the above technique.

Effects of operating variables and lubricating oils are discussed.

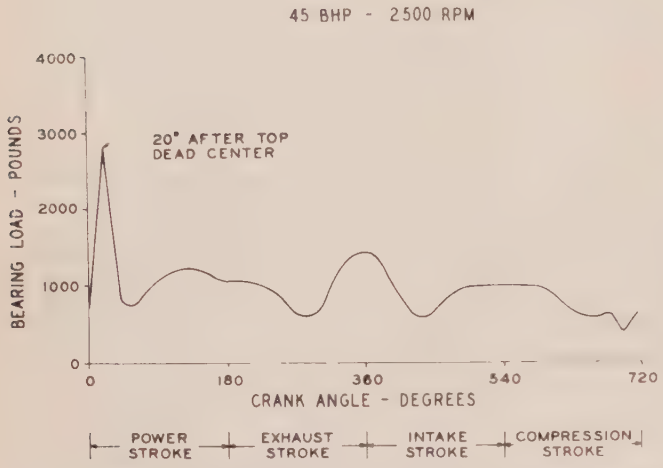

Fig. 2—Connecting-rod bearing load diagram for high engine load and low engine speed

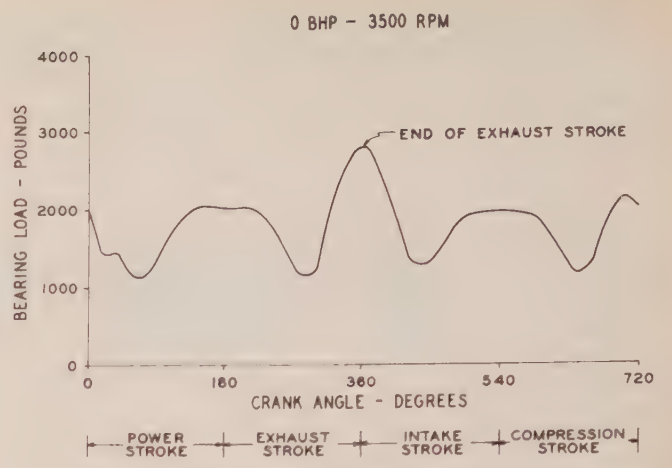

Fig. 3—Connecting-rod bearing load diagram for low engine load and high engine speed

the counting circuit. The radioactivity level is counted and recorded using a Geiger tube, counting ratemeter, and recorder, respectively. Lead shielding is placed between the counting chamber and the test bearing which are approximately 15 ft apart, in order to insure a low background radiation level at the counting chamber.

A typical test is conducted as follows: charge the engine with test oil, start the engine and transfer pump, and flush the engine and counting system for 15 min. Drain the flush oil and recharge with fresh test oil. Turn on the transfer pump and radioactivity counting apparatus, then start the engine. Warm up at low speed and load until the oil temperature reaches 150 F, then set the desired test speed and load. When the operating conditions and bearing wear rate have reached equilibrium, the test is started. The test is run for 12 hr, and the increase in radioactivity level during the test period is noted. The observed radioactivity increase is then expressed as bearing-weight loss by referring to the radioactivity of a standard solution which contains a known weight of irradiated bearing material.

An explanation of the probable bearing wear mechanism will aid in a better understanding of the results to be discussed. Visualize a connecting-rod journal and a bearing with clearance space completely surrounding the journal. If a load is applied at any point on the bearing, the bearing will move towards the journal surface. When there is relative motion between the two surfaces, wear will occur if the surfaces contact each other. The load being applied to the bearing must be transmitted to the journal. During the major portion of engine operation, the load is transmitted by an oil film of sufficient thickness to prevent wear. The oil film is of inadequate thickness at certain times during an engine cycle however, and the two bearing surfaces contact each other briefly, resulting in wear.

The oil film's ability to carry load in a connecting-rod bearing depends on several factors. One primary factor is the complex pattern of bearing loading. An analysis of the pattern of bearing loading and its effect on oil film load-carrying capacity follows.

The load on a connecting-rod bearing at any point during an engine cycle is approximated very closely by a summation of three forces.[1] These forces are: (1) the gas force acting on the piston, (2) the inertia force due to the reciprocating-piston assembly and small end of the connecting rod, and (3) the centrifugal force due to the rotating mass of the large end of the connecting rod and the bearing. In calculating the inertia forces, the common method is to consider the rod as having two component masses, concentrated at either end of the rod.

Figs. 2 and 3 show bearing load diagrams typical of two extremes of engine operation. These were calculated for the engine equipped with the irradiated bearing. Fig. 2 shows that when the engine is operated at high-engine-load low-speed conditions the maximum connecting-rod bearing load occurs at about 20-deg atdc on the power stroke. This is due to high gas pressure, which is a function of engine torque. In contrast, Fig. 3 shows that when the engine

[1] "Analysis and Lubrication of Bearings," by M. C. Shaw and E. F. Macks. Pub. by McGraw-Hill Book Co., Inc., New York, 1949.

is operated at low-engine-load high-speed conditions the maximum bearing load occurs at the end of the exhaust stroke. This is due to high inertia forces, which are a function of engine speed.

As shown in Figs. 2 and 3, the connecting-rod bearing loads change continuously throughout the engine cycle. In addition to the continuous variation in load magnitude, there is also a continuous change in the direction in which the load is acting. At any given crankangle, a specific bearing load is applied radially to a specific portion of the bearing surface. This direction of load application is referred to as bearing-load position. As an example, calculated load positions at several points in an engine cycle are shown in Fig. 4. The bearing load diagram shown is the same as in Fig. 2 for high engine load and low engine speed. The load is applied to the bearing in the directions shown by the arrows at various crankangles. Thus, in addition to the continuous changes in bearing load, continual changes also occur in the bearing-load position during each cycle. This motion in bearing load position has a pronounced effect on the bearing load-carrying capacity of the oil film within the bearing. This was described by J. M. Stone and A. F. Underwood in their paper "Load-Carrying Capacity of Journal Bearings." Load-carrying capacity can be defined as equal to the bearing load required to produce the minimum oil film thickness which will prevent contact between the bearing and journal surfaces. The load-carrying capacity depends on several factors including bearing and journal dimensions, viscosity of the lubricant, minimum oil film thickness to prevent wear, rate of journal rotation, and rate of change in bearing load position. Using bench-test apparatus, Stone and Underwood demonstrated that the effects of journal rotation and load motion could be expressed by the term $(2N_L - N_J)$, where N_L is the rate of change in bearing load position and N_J is the rate of rotation of the journal with respect to the bearing. Assuming that the other factors mentioned above remain constant and are represented by K, load-carrying capacity can be expressed by the equation:

$$\text{Load Carrying Capacity} = K(2N_L - N_J)$$

Their finding was extremely important because it showed that the rate of change of bearing load position has a major effect on the hydrodynamic oil film and that the rate of rotation of the journal with respect to the bearing was not the only speed effect involved, as was generally believed.

One of the more interesting conclusions that can be drawn is that if the load position happens to be moving in the same direction and at one-half the speed of journal rotation, the oil film has no load-carrying capacity. This is commonly referred to as the "critical speed" ratio.

Calculations were made of the load-carrying capacities under the conditions shown in Figs. 2 and 3. Figs. 5 and 6 show these same bearing load diagrams

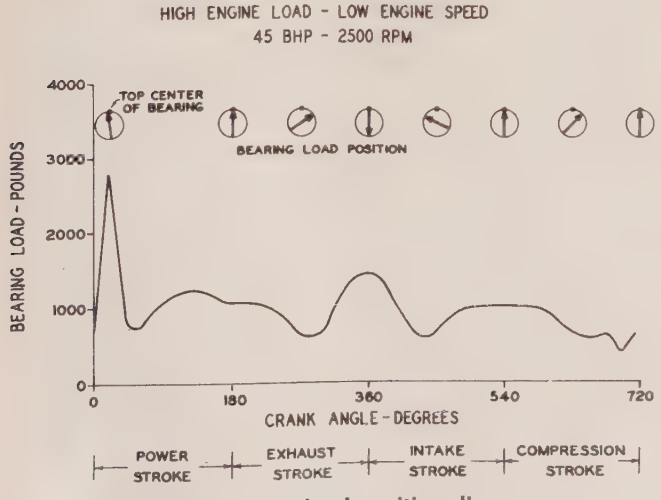

Fig. 4—Bearing load position diagram

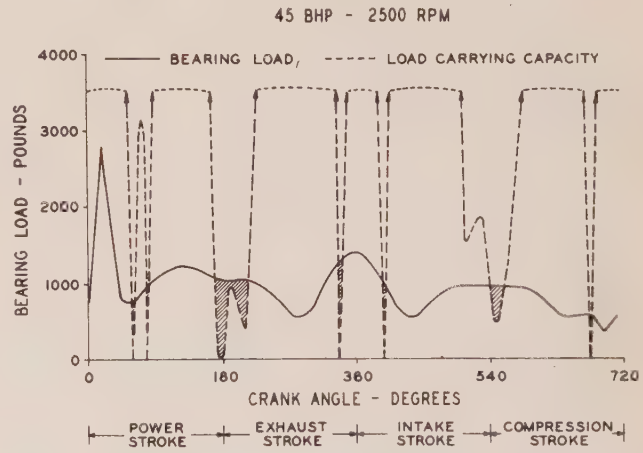

Fig. 5—Theoretical bearing wear diagram for high engine load and low engine speed

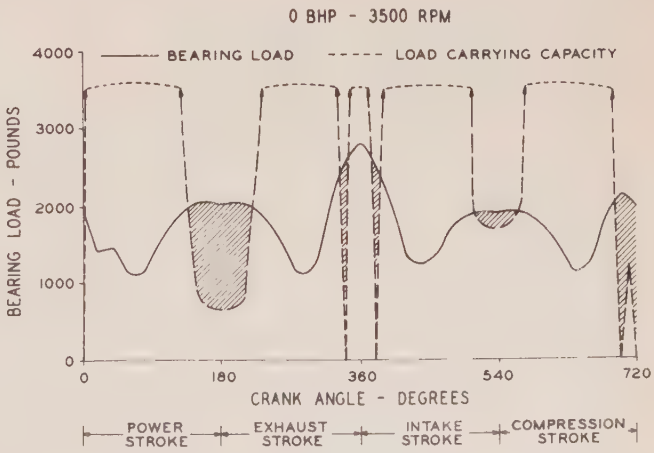

Fig. 6—Theoretical bearing wear diagram for low engine load and high engine speed

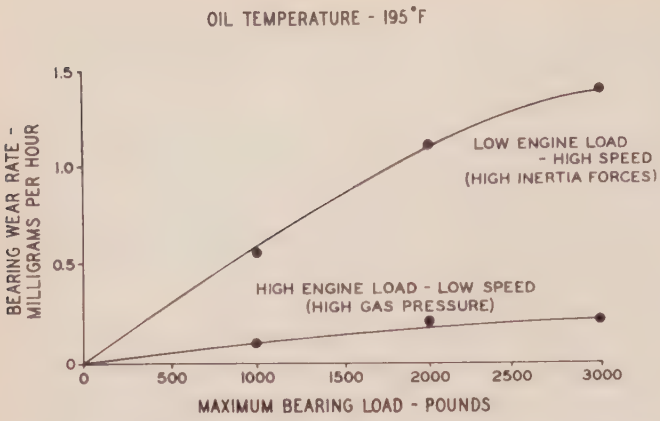

Fig. 7—Effect of maximum bearing load and its source on connecting-rod bearing wear

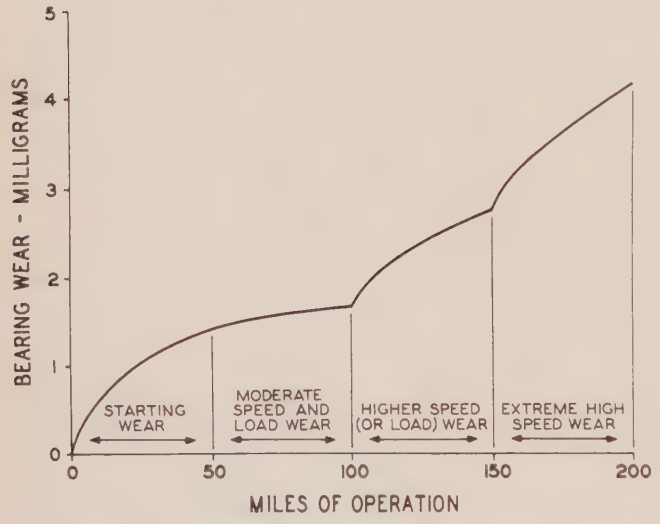

Fig. 8—Effect of changes of engine load and speed on bearing wear

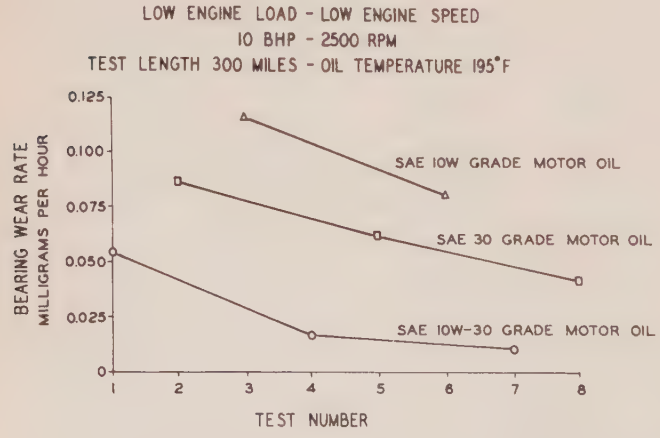

Fig. 9—Effect of oil viscosity on bearing wear

with the load-carrying capacity curves superimposed. In these cases the load-carrying capacities were determined by assuming a value for K, the constant term in the load-carrying-capacity equation. This assumption was necessary due to the difficulty in assigning exact values for the viscosity of the lubricant in the bearing and the minimum oil film thickness to prevent wear. These curves, therefore, do not show exact values, but they do demonstrate the pattern of load-carrying capacity. It is shown that the load-carrying capacity varies over a wide range and in fact equals zero several times during an engine cycle. These curves define the positions where the bearing load will probably exceed the load-carrying capacity of the oil film. The shaded areas denote these positions, and these are the portions of the cycle during which wear is most likely to occur. Further, the size of the shaded areas indicates the relative severity of wear.

Comparison of Figs. 5 and 6 shows that a larger amount of wear would be expected to occur during the high-speed, low-load operating conditions shown in Fig. 6, because the shaded areas are three to four times as large as those in Fig. 5. This same relationship would hold regardless of the value assumed for K. The operating conditions for which these curves were calculated were selected to give equal maximum bearing load.

The authors conducted engine tests employing the radioactive tracer wear technique to investigate this point. The results presented in Fig. 7 show that a considerably higher wear rate did occur under low-engine-load high-speed conditions than under high-engine-load low-speed conditions for given values of maximum bearing load. The observed differences in wear agreed with the calculated differences in wear, as shown by the relative amounts of shaded areas in Figs. 5 and 6, thus lending further support to the findings made by Stone and Underwood. It can also be concluded from Fig. 7 that bearing wear is a function of both the magnitude and source of maximum bearing load.

Fig. 8 is an idealized chart showing the effects of changes in engine load and speed on bearing wear. Starting wear is defined as the wear occurring during start and warmup until an equilibrium wear rate is established. Starting wear after several hours of engine shutdown is in the same order of magnitude as the wear caused by the extreme high-speed operation and is observed for as long as 1 hr of operation. The starting wear is significantly less, however, when the engine has been inoperative for only 10 to 20 min, probably due to the lower viscosity of the comparatively warm oil and perhaps residual oil in the bearing clearance space. Both factors promote faster establishment of a load-supporting oil film and tend to reduce roughening of the bearing during engine startup. These results indicate that starting wear is due to break-in of contact areas which become roughened during each engine startup.

Changing engine load or speed after an equilibrium

wear rate has been established causes an initial "jump" in wear followed by a gradual return to a steady wear rate. This is believed due to a shift in the positions where the bearing load exceeds the load-carrying capacity of the oil film, and a break-in of the new contact areas takes place before an equilibrium wear rate is established.

Connecting-rod bearing wear should be inversely proportional to the lubricant's viscosity according to the hydrodynamic load-carrying-capacity theory. The performance of oils over a range of viscosities is illustrated in Fig. 9. The wear results of a series of eight tests conducted with three oils are plotted against the test number.

These data show that engine usage significantly affects the bearing wear performance of oils, and satisfactory comparisons between oils can be made only on a relative basis. Although the wear rate for any given oil decreased in this series of tests, sufficient data have not been obtained to conclude that bearing wear generally decreases with engine usage. These tests were conducted shortly after installation of a new bearing, and the mating surfaces were probably still wearing in.

Fig. 9 also shows that the SAE 30 grade motor oil protects against wear better than the SAE 10W grade motor oil. These data support the load-carrying-capacity theory, because the 30 grade oil is more viscous than the 10W grade oil.

The SAE 10W-30 grade motor oil inhibited wear significantly better than the 30 grade oil, although the measured viscosities of the two oils are very similar at the 195 F gallery oil temperature. The effective viscosity of the 10W-30 oil in the bearing is not known. The temperature rise in the bearing would tend to give a higher relative viscosity for the multigrade oil, but, on the other hand, the high shear rate in the bearing would tend to result in a lower effective viscosity for the V.I. improved oil. In any event, the observed differences in wear appear considerably larger than can be explained by simple viscosity effects. Whatever the inhibiting mechanism, five SAE 10W-30 grade oils, each containing a different type of V.I. improver, have exhibited similar performance relative to an SAE 30 grade oil.

The wear rates shown in Fig. 9 were observed in tests corresponding to 300 miles of operation. Actually, all of the tests were run for about 550 miles. Fig. 10 shows typical results of a series of extended tests with the 10W-30 grade oil and the SAE 30 grade oil. The wear measurements were adjusted to eliminate the effect of engine usage. A pronounced change in wear rate at about 400 miles of operation was exhibited by the multigraded oil. After this inflection, the wear rate was nearly identical with that shown by the 30 grade oil. The inflection in wear with the multigraded oil has been observed consistently but as yet has not been satisfactorily explained. The wear advantage of the 10W-30 oil is distinct, but extended

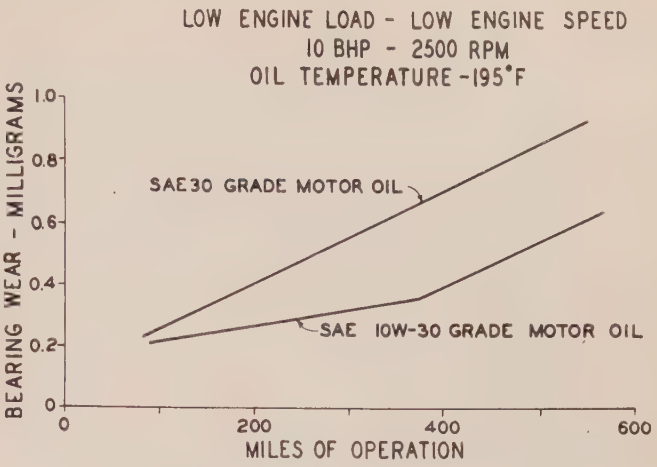

Fig. 10—Effect of oil usage on oil performance

oil-change periods would reduce the percentage improvement.

The following conclusions have been formed from this work:

1. Connecting-rod bearing wear using babbitt-surfaced bearings can be determined rapidly by using a radioactive tracer technique.
2. The data indicate that relative connecting rod-bearing wear at different operating conditions can be predicted using the load-carrying-capacity concept.
3. Bearing wear is a function of the magnitude and source of the maximum bearing load.
4. In this engine, the most severe bearing wear occurs during engine starts or at extreme high-speed operation.
5. Bearing wear is lower with an SAE 30 grade motor oil than with an SAE 10W grade motor oil.
6. Bearing wear is lower with an SAE 10W-30 grade motor oil than with an SAE 30 grade motor oil.

DISCUSSION

Shows Effect of Lubricants on Bearing Wear

—R. F. Gasvoda
Ford Motor Co.

THE radioactive tracer technique applied to measurement of engine bearings presented here offers another means for the peaceful use of atomic energy in advancing the knowledge of the engineer and scientist. We at Ford Research are also employing the methods of radioactive tracers in the study of wear problems of rings, gears, and roller bearings. We have found this technique to be an effective method for determining wear and wear rates in a short period of time.

Several years ago we initiated a program to study the effects of bearing loads and load concentration on the wear of shell-type bearings. At the time we used physical measurements to

determine the wear. We found that the wear, even under severe conditions, could not be accurately measured. We also observed that the starting wear was an important variable in determining wear at different bearing loads.

The authors have developed here the fact that high engine loads or high engine speeds promote a higher rate of bearing wear than low loads or low speeds. After establishing this fact, they proceed to show the results of tests on oils of different SAE viscosities. However, we note that these tests have been run under conditions of low load and low speed. According to the authors these conditions are not conducive to the highest rates of bearing wear.

In Fig. 9, we saw data indicating that lower bearing wear rates result as the viscosity of the oil is increased. This is in agreement with the generally accepted theories on the viscosity effect in lubrication of a journal bearing. However, next we saw data indicating that 10W-30 oils possess some property not generally recognized in journal-bearing calculations, and the wear rates are extremely low. Fig. 10 showed results of extended bearing wear tests at low load and low engine speeds. In these tests the 10W-30 oil shows lower bearing wear than the SAE 30 over the test period, and a lower rate of wear for the first 350 plus miles of operation. According to the classical bearing lubrication theories, we might suspect that the 10W-30 oil has a plus value giving it an effective viscosity comparable to something like an SAE 50. The authors do say that they consistently observed an inflection point in the wear rate when 10W-30 oils were used. This suggests that something in the composition of this oil may change and that after a short operating period this plus value is missing. Now we ask, would the same results be obtained at higher loads or higher speeds and greater rates of shear? Would the inflection point occur earlier if at all? Is the lower wear rate experienced due to the large molecular structure of the V.I. improver? Could this condition also be produced by employing an SAE 30 oil composed of 10 to 20% of an extremely heavy bright stock?

Before too much emphasis is placed on the merits of 10W-30 oils in reducing bearing wear, Fig. A shows the results obtained by the Ford Motor Company in 20,000-mile field tests on a variety of crankcase oils. These lubricants are marketed, branded oils of a number of petroleum companies. The bearing weight losses shown represent the average of approximately 300 individual bearings. The data presented include losses attributed to mechanical functions such as dirt scratches and some slight fatigue areas in the bearings.

We consider these weight losses to be satisfactory for 20,000 miles of operation. Four of the oils represented are of the high V.I., multiple viscosity, branded oils. Observation of these test data shows that good SAE 10W oils are effectively lubricating the bearings of our engines in this type of field test.

Discusses Wear Rate Under Different Operating Conditions

—F. C. Burk
Atlantic Refining Co.

WE wish to thank the authors for a very informative paper on their bearing wear studies. It is another excellent example of extending the tracer technique in investigating engine wear phenomena in that it permits data to be obtained quickly which could not be determined as precisely by conventional means.

The authors have shown that relative bearing wear at different operating conditions can be predicted with this technique by using the load-carrying-capacity concept. In this connection it should be pointed out that the wear differences shown in Fig. 7 for the two kinds of engine operation would not be as great as pictured if expressed in wear per mile rather than wear per hour. In the case of low-engine-load high-speed conditions we might expect greater wear due to the additional rubbing revolutions since the engine is running 1000 rpm faster. Compensation for this difference in speed reduces the wear differences to 5/7 of its original value. In spite of this correction, however, there still remains a substantial difference in wear between the two operating conditions.

We were particularly interested in the data in regard to short periods of high wear at the start of their tests, followed by steady wear for an extended period of time. These results have been confirmed in our laboratory on piston-ring wear and merely reflect the break-in of new contact areas. The wear carry-over effect from a high-wear to a low-wear condition was previously presented at an SAE meeting by H. R. Jackson of our laboratory although it was specific to piston-ring wear. We would be interested in hearing if the authors encountered this same phenomenon of wear carry-over in their bearing wear studies.

The authors show, in Fig. 10, that SAE 10W-30 grade motor oils allowed less wear than SAE 30 grade oils. On extended tests, however, this was only true for about the first 400 miles of operation at which point the wear rate of the SAE 10W-30 grade became nearly identical with that of the 30 grade. We have also experienced this "jump" in wear rate of some multigrade oils after extended use, particularly in radioactive valve lifter wear studies in which friction wear was predominant. The amount of usage which the oil could tolerate without a marked change in wear rate varied with the oil composition and is thought to be due to the depletion or effectiveness of some additives.

Authors' Closure To Discussion

IN answer to the question raised by Mr. Gasvoda regarding the effects of higher loads, higher speeds, and greater rates of shear on the bearing wear results, we found that such changes in engine operating conditions did not significantly affect the relative performance of the various types of lubricating oils on bearing wear. The wear rates were increased for all oils as test severity was increased but the relative order of performance of the various types of oils did not change. Oils showing the least wear under low-load, low-speed test conditions also showed the least wear under higher-speed, higher-load conditions.

We investigated the effect of bright stock on bearing wear with SAE 30 grade oils having bright stock content ranging from 0% to greater than 20% and saw no indication of an inflection point in the bearing wear curve. This effect was also sought with SAE 20, 40, and 50 grade oils but was not obtained.

In answer to Mr. H. R. Jackson's question regarding wear carryover effect from a high-wear to a low-wear condition, we found wear carryover of this type to occur. Our bearing wear carryover was similar in pattern to the wear carryover effect observed by Mr. Jackson in piston-ring wear studies.

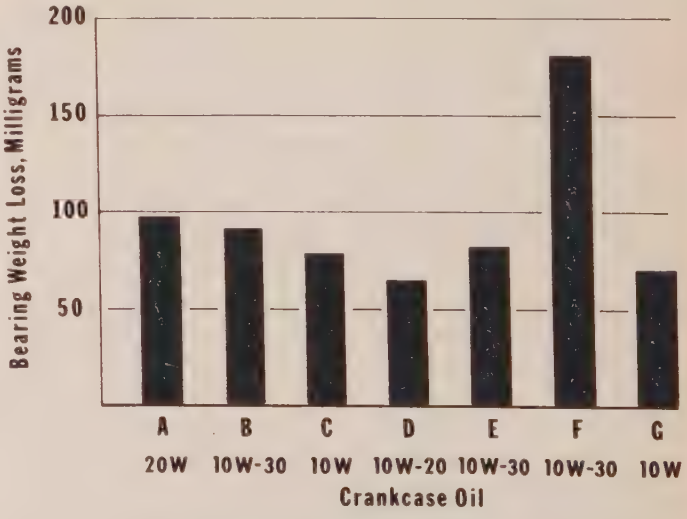

Fig. A—Connecting-rod bearing weight loss for 20,000-mile field test with different oils

The Continental 750-Horsepower Aircooled Diesel Engine

Herbert H. Haas and Earl R. Klinge,
Continental Aviation and Engineering Corp.

This paper was presented at the SAE Annual Meeting, Detroit, Jan. 17, 1957.

A 750-HP aircooled diesel engine has been successfully developed with the primary purpose of reducing the fuel consumption of gasoline-engined-powered combat vehicles by approximately 40%. The new powerplant is characterized by low fuel consumption, light weight, and high compactness.

The basic features of the engine are: 5.75-in. bore x 5.75-in. stroke, 1790-cu in. displacement, compression ignition, 4-stroke cycle, direct-injection system, 90-deg Vee, 12-cyl, aircooled, exhaust turbocharged. Rated output is 750 hp at 2400 rpm with a potential output increase up to 850 hp.

Engine oil coolers transmission oil coolers, and cooling fans are mounted on the engine and form an integrally cooled powerplant.

THE Ordnance Department of the Army and Continental Motors have cooperatively accumulated an experience of many years in solving the peculiar problems of armored-vehicle powerplants. Many outstanding engines have been developed especially for this purpose, and a number of them are in volume production. The most important of these engines is the AV-1790 aircooled gasoline engine installed in the M-48 medium tank. This engine installation typifies the special requirements of combat vehicles. These requirements might be stated basically as:

1. Minimum bulk for the total power package including all cooling of engine and transmission.
2. Maximum power within the limits of the military fuel specifications.
3. Adequate cooling of the power package buried in an engine compartment with all openings on top and with considerable airflow restrictions from armored bullet-proof airflow grilles.
4. Aircooling to minimize the problems in the temperature extremes of arctic and desert.
5. Light-weight power package.
6. Serviceability of all items from the top or front end of the engine.
7. Ability to function under extreme dirt conditions and complete submersion in water.

Diesel Versus Spark Ignition—Although satisfactory gasoline-engine powerplants resulted from the development work of the past, there exists a strong desire to reduce fuel consumption and increase vehicle range because of the logistic fuel problem. Various avenues of development are being pursued to achieve this objective. One of the most obvious matters to consider is whether or not the fuel economy of the diesel engine could be utilized in solving this problem without serious penalties in other areas. The possible gain in fuel consumption is very large

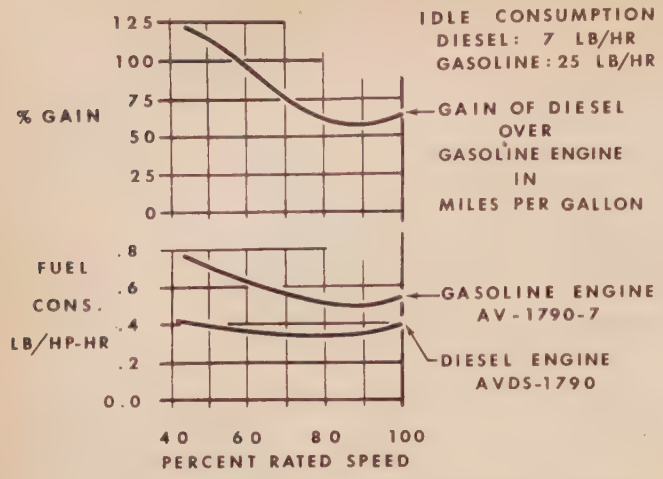

Fig. 1—Fuel consumption of diesel versus gasoline engine for propeller load curve

as shown in Fig. 1. From this curve as well as from experience, it can be concluded that the overall fuel consumption of the combat vehicle will drop as much as 40%.

Additionally, the turbocharged diesel engine has two other advantages:

1. High overload capacity.
2. Turbocharger compensates automatically for the losses due to high altitude and high ambient temperature.

However, to achieve this goal, there are several pertinent points to be considered:

1. Build the diesel engine into the same space as the gasoline engine.
2. Maintain the same net output.
3. Aircool the diesel engine and satisfy the stringent cooling requirements.
4. Cold start down to −25 F without external aid.
5. Make provisions to build a family of engines of different sizes using basically interchangeable major components as has been done on the gasoline series of engines in production.

All these requirements could be successfully met with the development of the aircooled Continental AVDS-1790 diesel engine. This engine can replace the present gasoline engine without any major penalty. A moderate weight increase and some small additions to the bulk are offset, when the total bulk of powerplant plus fuel tank is considered. If desired, the turbocharged diesel engine also can be overloaded 10 to 15% compared to the gasoline engine.

Aircooling—The cooling problems discussed in this paper are the result of specific requirements of a combat vehicle, such as bullet-proof air entrance and exit grilles, limited space for ducts, torque converter transmission cooling, and others. These problems are naturally the same for liquid-cooled and aircooled engine installations. Because a great deal of experience is now available with many thousands of aircooled engines in the Army, it is possible to conclude from this experience and from comparable liquid-cooled powerplants on the market, that the aircooled engine offers these advantages:

1. Power package bulk is minimum especially when designed for desert conditions.
2. Power package weight is minimum.
3. Cooling power is minimum especially in the extremely confined space of an armored vehicle.
4. Cooling trouble and cooling system maintenance are eliminated.
5. The problem of supplying coolants in the desert and in the arctic is eliminated.

It is upon this basis that it was determined that this new engine would be aircooled.

General Description

History—The predecessor of the AVDS-1790 aircooled diesel engine is the AV-1790 gasoline engine which, in its various models, is currently standard for combat vehicles. Some dieselization efforts on this engine in the past have shown that a redesign is necessary to obtain a diesel engine of satisfactory life at an output equal to or greater than the gasoline engine. However, it was desirable to maintain the basic configuration of the gasoline engine which represents a very suitable solution of the problems associated with aircooling and power package installation in armored vehicles. Furthermore, it reduces vehicle changes to a minimum and makes the diesel engine suitable for retrofit in existing installations. Although the dimensions of the major parts like the crankshaft could not be the same on the diesel engine as they were on the gasoline engine, the background of the gasoline engine is very apparent in the diesel engine.

Basic Configurations—Following the basic configuration of the gasoline engine, the new AVDS-1790 diesel is again a 12-cyl, 90-deg Vee engine with overhead camshafts, the cooling fans located on

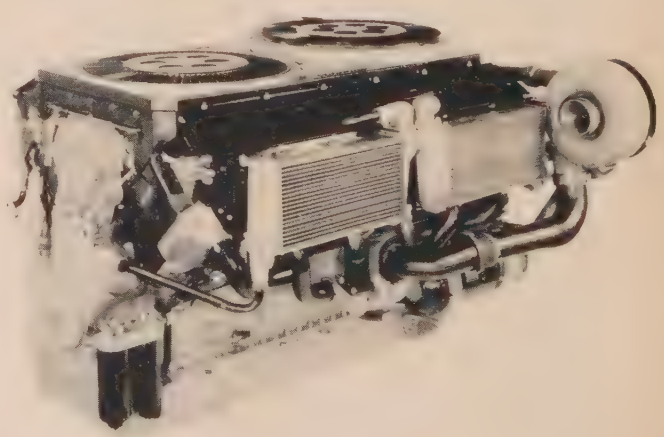

Fig. 2—Three-quarter right-front-end view of 1790 diesel

the top of the engine, and the engine oil and transmission oil coolers both mounted on the engine and cooled by the engine fans (unit cooling). The engine is attached directly to the transmission. Photographs of the complete engine package are shown in Figs. 2 and 3, basic installation dimensions in Figs. 4 and 5, and cross-sectioned views in Figs. 6 and 7.

Design Topics—With the features mentioned in the foregoing paragraphs, the similarity between the gasoline and diesel engine ends. The high combustion pressure, the injection system, and other characteristic items of the compression ignition engine required a new design concept. Five major objectives have been set for the design and development of the new engine.

1. Provide a rigid basic engine structure which permits high unit bearing loads and reduces stress concentration and engine friction. This has been accomplished by a single-piece crankcase of high rigidity and short stress path, by a crankshaft of generous dimensions, and by a carefully designed bearing construction.

2. Insure adequate cooling for cylinder and piston. This has been accomplished by careful finning of the aircooled cylinder unit, by ample oil cooling, and by adequate cylinder spacing and at the same time, minimizing the length of the accessory drive to keep the engine length within the given limits.

3. Drive the accessories in accordance to the individual requirements of each. This has been accomplished by driving all the accessories from the flywheel end and splitting the drive into two, a stiff

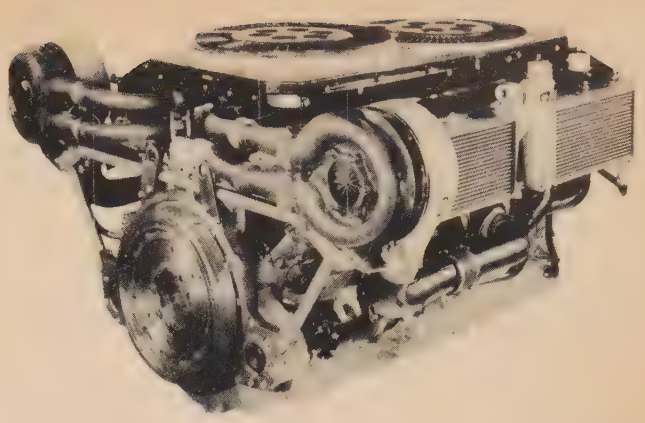

Fig. 3—Three-quarter left flywheel end view of 1790 diesel

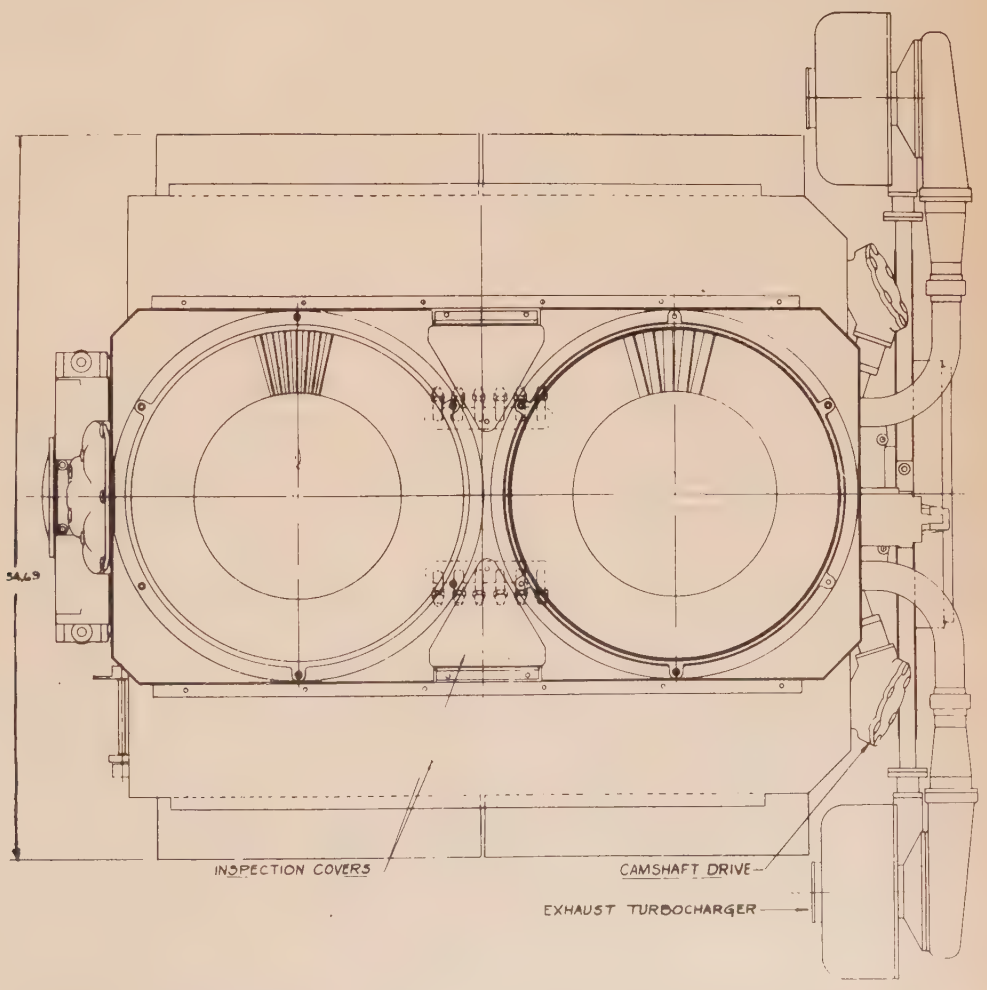

Fig. 4—Plan view of 1790 diesel installation

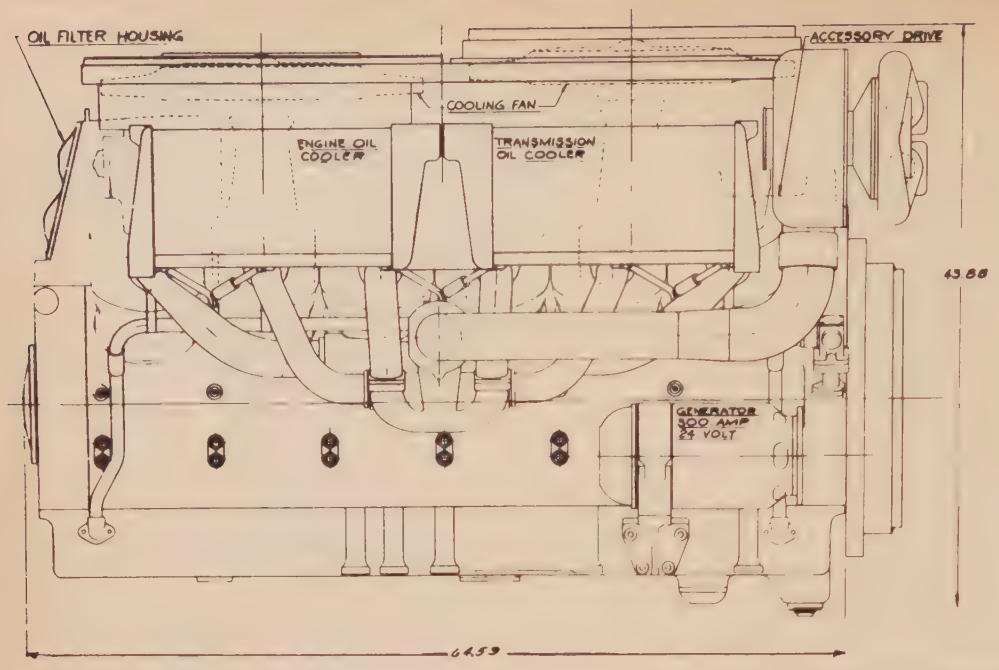

Fig. 5—Side view of 1790 diesel installation

train for camshafts and fuel injection pumps, and a flexible train for the cooling fans.

4. Simplify machining, tooling, assembly, and maintenance. This has been done by dividing the engine in several major assembly units:

a. Crankcase, a comparatively simple casting incorporating the bearing construction.
b. Accessory drive assembly incorporating the accessory drives, the bearings, shafts and gears, as well as the automatic timing device and one fan unit.
c. The damper and filter housing assembly incorporating the oil filters, control valves, and internal passages.
d. The triple oil pump including the drive gears and control valves.
e. The oil pan including the oil reservoir for the dry sump system.

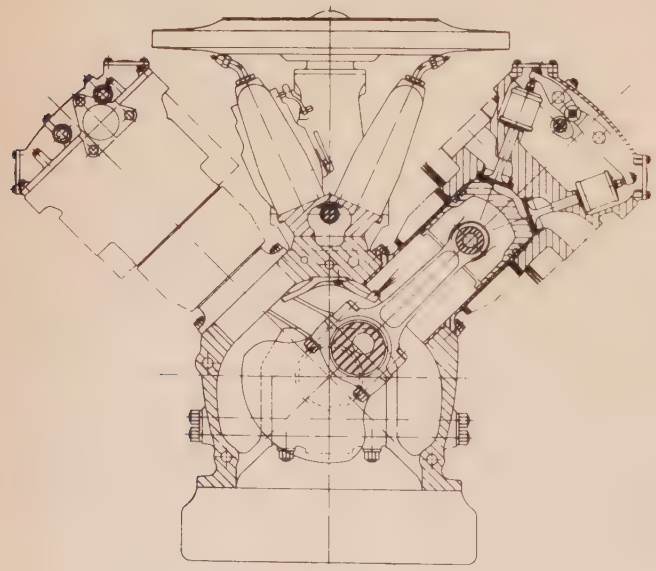

Fig. 6—Transverse section of 1790 diesel

5. Provide accessibility to oil and fuel filters and to the injection system (injection pumps, injection nozzles, and injection timing) from the top and the front end of the engine.

Major Dimensions—The bore and stroke of the engine are 5¾ in. x 5¾ in. and the displacement 1790 cu in. These dimensions are the same as the gasoline engine primarily because of the given bulk limitations. The cylinder spacing is 9.00 in. or 1.56 times the bore. This ratio, which is a critical dimension in both aircooled and liquid-cooled engines, is dictated by the required uniformity of the cylinder-dome wall structure and by the necessary cooling fin area in between the cylinders. Simultaneously, this spacing permits large counterweights on the crankshaft so that the reactions of the heavy reciprocating parts were reduced to a negligible size.

Combustion System—The combustion system is of the 4-cycle direct-injection type with an open quiescent combusion chamber. It has the so-called "Mexican hat" shape and is formed by the piston crown. The advantages of this type combustion chamber are:

1. High performance potential.
2. Minimum fuel consumption.
3. Suitability for exhaust turbocharging.
4. Good cold start.
5. Low heat rejection to cylinder and piston.
6. Uniform thermal loading.
7. Good breathing capacity particularly if combined with a dome cylinder head and angle valves.

However, it is to be pointed out that these qualities can be used to full extent only when particular attention is given to an adequate engine structure and matching performance of the entire injection system. On the combustion chamber chosen, considerable experience has been gained on previous developments and resulted in satisfactory combustion efficiency up to 2600 rpm.

Aircooling System—The cooling arrangement of Fig. 8 deserves some explanation. The basic reason for suction cooling is to surround the engine and the fuel tanks with the incoming cold air. This permits confining the hot discharge air to a relatively small duct in its path toward discharge from the vehicle. A slight penalty on cooling fan size is endured for this arrangement as the cooling fans must handle the less dense hot air. On the other hand, it can be demonstrated that the circumferential temperature distribution around the cylinder is more uniform with suction cooling than with pressure cooling. Besides the oil coolers for the engine, the torque converter transmission coolers are mounted on the engine and cooled by the engine fans. This results in considerable space saving compared to having separate fans for each cooling function. It should be pointed out that the torque converter transmission cooling constitutes a problem in off-road operation. A large portion of the engine horsepower might have to be dissipated through the transmission oil coolers if the vehicle is operated at full throttle but nearly stalled in the mud. On the AVDS-1790, transmission oil cooling is provided for a heat rejection equivalent of one-third of the net engine output.

Another problem is the airflow restriction presented by the bullet-proof grilles for admission and discharge of the cooling air. These restrictions add considerably to the required cooling fan power.

As a result, the total fan power amounts to nearly twice the fan power needed for the engine cooling alone. Two 26-in. axial fans are provided to satisfy all cooling requirements. They are mounted on top of the engine and driven at twice the engine speed.

Detail Description—Crankcase: The crankcase is a simple and rugged one-piece cast aluminum structure of Alcoa 355 with forged aluminum main bearing blocks as shown in Fig. 9. Special attention has been given to the shape of the various crankcase sections to minimize stress concentration and deformation under load. The contours of the crankcase are smooth and without ribs of any kind. Short and straight stress paths are provided. The introduction of the combustion force into the crankcase is accomplished, rather than by a few stiff bolts, by many highly flexible bolts to assure a good stress distribution over the entire crankcase structure. All highly stressed bolts are of the necked type.

Uniform load distribution in the bearing area is considered indispensable for utilizing the bearing capacity to its maximum. This is obtained by a heavy diaphragm support structure with four cap bolts and two through bolts for each bearing, clamping the main bearing caps in the tunnel slot of the crankcase. The bearing caps thus function as an integral part of the crankcase, tying the two crankcase side walls together. No motion of the bearing has been experi-

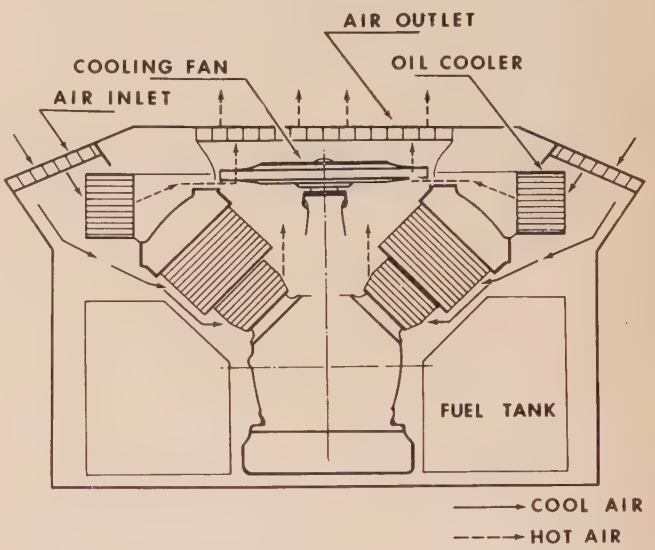

Fig. 8—Airflow over engine in vehicle hull

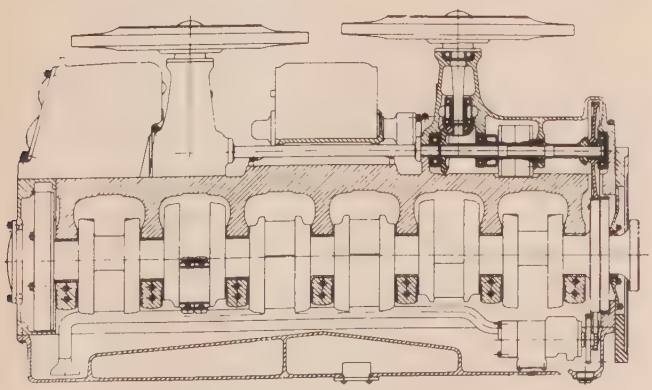

Fig. 7—Longitudinal section of 1790 diesel

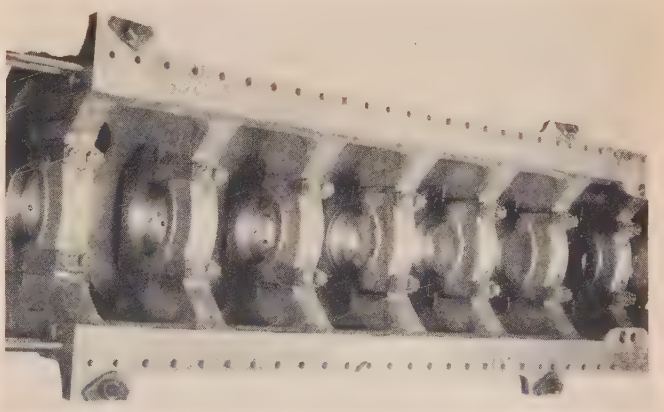

Fig. 9—Bottom view of 1790 diesel crankcase

Fig. 10—Crankshaft and connecting-rod assembly of 1790 diesel

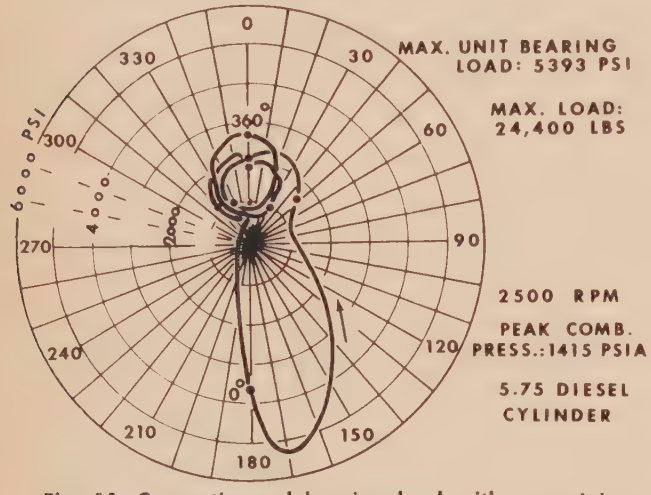

Fig. 11—Connecting-rod bearing load with respect to crankthrow

enced in spite of the very high total loads (20,000-lb peak) being handled.

The crankcase extends far below the crankshaft centerline, and the cast aluminum oil pan is made a part of the structure. Thus, a stiff beam member is obtained which compensates for the small total crankcase height, which is characteristic for aircooled engines.

Crankshaft and Bearings: The crankshaft received considerable attention because of the anticipated high output ratings demanding the use of peak combustion pressure between 1600 and 1800 psi. In addition, and perhaps even more important, was the need for torsional rigidity and bending rigidity of the crankshaft. Thus, the main journal size was chosen as 4.25 in. and the crankpin size as 3.75 in. The width of the cheek is 1.6 in. Crankshaft counterweights integrally forged were provided for approximately 85% balance. This was done to reduce the internal forces trying to break the engine in the middle. The crankshaft of Fig. 10 is forged from SAE 4340 steel (Ni, Cr, Mo, Mn) and nitrided. The bearings are copper-lead in both main and connecting rod and are backed with rather heavy steel shells, 0.250 in. thick in the case of the main bearings. The main bearings have a width of 2 in., the connecting-rod bearing of 1.35 in. Connecting-rod and main bearing polar load diagrams are shown in Figs. 11 and 12. The results to date with these bearings have been excellent, and no difficulty of any kind has been encountered.

Connecting Rod: The connecting rod is of conventional design. The length from center-to-center is 11 in., the piston-pin diameter is 2 in. The piston-pin bushing at the small end of the connecting rod is a rolled, steel-backed bronze bushing pressed into place and diamond-bored.

The rod is forged from SAE 4140H steel (Cr, Mo, Mn) and fully shotpeened. Considerable effort has been directed towards maximum load capacity at minimum weight by extensive stresscoat, strain gage, and fatigue tests. Also, it may be mentioned that the roundness of the big end bearing is maintained over a wide variation of load and of the torque applied to the connecting-rod bolt, thus permitting the full utilization of the bearing capacity. A view of the connecting rod is shown along with other related components in Fig. 13.

Piston and Rings: The piston is an aluminum forging of Alcoa 32ST-6 in view of the very high loads encountered, particularly in the piston-pin boss area. The piston is oil cooled by an oil jet mounted in the crankcase. A separate oil system is built into the engine to handle the piston cooling.

Piston-ring selection presented very little difficulty, the 4-ring setup being characterized by all chrome-plated rings for maximum dirt wear resistance. A 15-deg included-angle keystone-top ring is used, along with two taper-faced compression rings and a dual-edge conformable oil scraper ring. Basic ring thickness is 0.090 in. which provides maximum bore conformability and minimum ring groove wear without any evidence of ring flutter at high speed. The

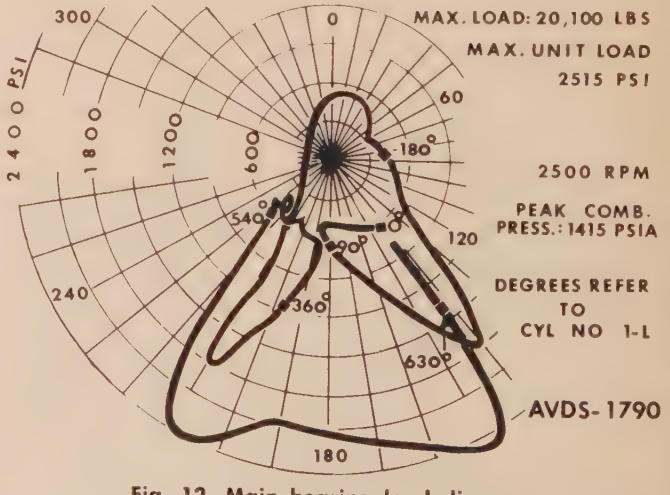

Fig. 12—Main bearing load diagram

piston assembly is shown in Fig. 13 with other related components.

Cylinder Unit: The cylinder head is an aluminum casting of Alcoa 142-T75 Cu Ni alloy with high-temperature strength characteristics. The main features of the cylinder-head design are: overhead camshaft, two angled valves, a simple cooling fin design, and an injection nozzle installation with a high degree of accessibility. The valve plane is shifted slightly off-center to permit the nozzle assembly to be removed directly from the top.

The combustion-chamber dome shape was the subject of a careful investigation with models and cylinder heads. A compromise was necessary between the conflicting desires of a large-radius dome for minimum stress, and a sharp-cornered dome which would permit an increase in piston length and reduce the piston-crown area exposed to combustion.

The cooling fins are set on the cylinder head at an angle of 30 deg and reach from the intake to the exhaust side of the head. Ample cooling fin area was developed by this method, although some foundry trouble occurred in casting these long deep fins. The cooling performance of the cylinder is outstanding with a maximum head temperature of 350 F at 750-hp output at 6 in. of water pressure drop across the cylinder. It is felt that the success of the cylinder and the satisfactory functioning of the injection nozzle is largely due to this cooling condition.

The cylinder barrel is from 8740H steel (Cr, Mo, Mn) and is screwed and shrunk into the cylinder head while the head is heated. The valve seats and valve guides are shrunk into place at the same time. This is a procedure of long standing in the aircraft industry and caused no difficulty of any consequence. Also routine is the use of Al-Fin bonded aluminum cooling fin muff on the barrel.

The intake and exhaust valves and seats are all fabricated from a common silicon-chrome valve steel, and due to the low exhaust temperature of the diesel cycle there is no need for expensive alloys. However, it has been found desirable to stellite-face the intake valve seat to prevent wear. Apparently seat wear occurs because of the completely dry inlet valve operation in a supercharged engine. The exhaust valve never showed any sign of difficulty.

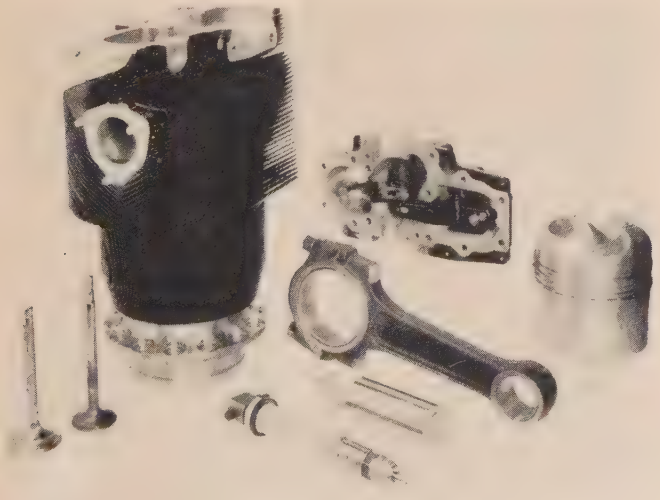

Fig. 13—Aircooled cylinder and major components of 1790 diesel

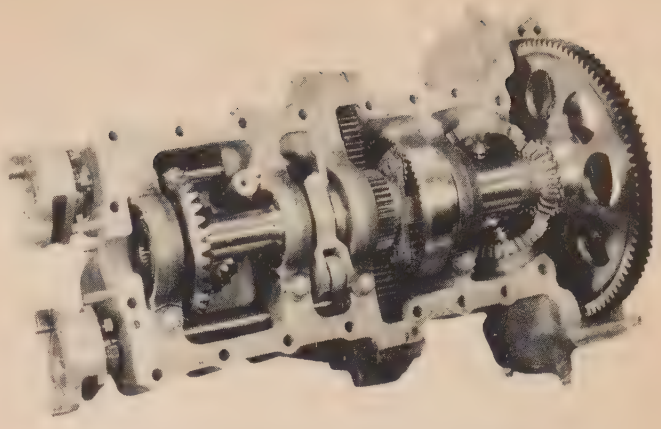

Fig. 14—Accessory gears of 1790 diesel

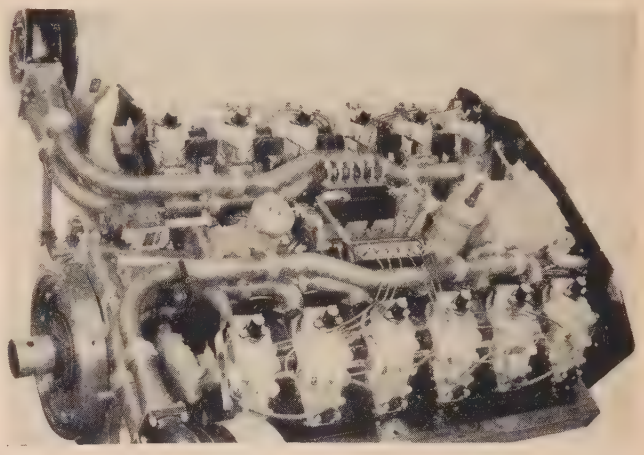

Fig. 15—Top view of engine showing exhaust manifolds, fuel injection system, and cooling fan towers of 1790 diesel

The injection nozzle assembly is extended through the rocker box for easy removal without touching any other parts of the cylinder assembly. A hold-down spring is used to avoid overloading the nozzle gasket and to compensate for differential expansion between aluminum and steel as shown in Fig. 13.

On the lower end of the injection nozzle assembly, cooling fins are provided over which air is bled from the supercharger through a hole drilled from the inlet port. Additionally, the nozzle holder is cooled by the oil in the rocker box, and a seal on the upper end of the nozzle holder prevents oil from leaking to the outside.

Accessory Drive: The entire accessory drive system is enclosed in one housing assembly on top of the crankcase (Figs. 14 and 15). It includes the bevel gear drive for the two overhead camshafts, the automatic injection timing unit, the drive for the two injection pumps, and the drive for both cooling fans.

The drive is taken from a gear next to the flywheel, thus assuring a nearly constant velocity which is considered an important provision in the basic engine design. Both camshafts and injection pumps are driven by a hollow shaft of high torsional stiffness whereas the fan drive is taken through flexible quills to avoid overstress during the acceleration period. All shafts, gears, bearings, and seals are integral with the accessory drive housing assembly, which forms a completely independent unit for machining and assembly.

Lubrication System: Considerable attention has been given to the lubrication system with regard to the following requirements:

1. Absolute minimum engine height.
2. Operation at 30-deg angle lengthwise and crosswise with a minimum oil quantity.
3. Dry sump condition when engine operates at high speeds to prevent oil foaming and power loss by crank dipping.
4. Minimum unusable oil quantity.
5. Ample piston oil cooling.
6. Proper feeding of the main oil pump pickup under all conditions.
7. Storage of the lube oil reserve within the engine.

The lubrication system shown in Fig. 16 developed for this engine satisfies these requirements. It consists of two circuits, the main lubricating circuit and the piston cooling circuit, which are operating independently and are fed by one multiple oil pump. The main pressure pump draws its oil from a compartment in the oil pan, which is fed by a twin scavenge pump from both ends of the engine. The oil is forced at a maximum quantity of 50 gpm through the engine oil coolers, then through a full-flow filter, and thence through the engine oil galleries to the bearings. A pressure regulator is influenced by the pressure in the main bearing gallery and dumps the surplus oil unfiltered back in the pan. A small partial flow is split off after the main filter and fed through a second full-flow fine filter to the exhaust turbochargers and the fuel injection pumps.

The piston cooling oil circuit is separated from the main lubricating system for three reasons:

1. To reduce the oil filter size by preventing cooling oil from flowing through the filters.
2. To reduce undue power loss by means of bypassing the cooling oil, if cold.
3. To divide the oil flow in view of the high total of nearly 70 gpm.

The piston cooling pump picks the oil up from the open end of the pan at an appropriate level and delivers the oil at a rate of 20 gpm to oil jets. These are located on the bottom of each cylinder and provide a continuous oil spray to the inside of the piston. A thermostat-controlled bypass valve opens at an oil temperature below 140 F and dumps the oil flow of the piston cooling pump directly back to the sump, thus preventing unnecessary power loss at other than high-load condition.

The oil pump is mounted on the flywheel end of the crankcase and is driven by spur gears from the accessory drive gear on the crankshaft. The oil-pump assembly is formed by the main pressure pump, the piston cooling pump, and the scavenge pump. The scavenge pump has three gears and operates as a twin pump. A safety valve attached to the main pressure pump and the thermostat bypass valve for the piston cooling circuit are both part of the oil pump assembly and prevent damage from overload during cold start. Also the idler gear which transmits the drive from the crankshaft to the oil pump shaft is mounted on the oil pump.

The oil pan is divided in the pressure pump compartment, the reserve compartment, and the two open end compartments. The oil flowing back from the engine is picked up on both ends of the pan by the twin scavenge pumps and fed to the pressure pump compartment. After this compartment is filled, the oil overflows to the reserve compartment, where it is stored. Thus, a constant level of 5 in. is maintained above the pressure pump pickup, which is an integral part of the pressure compartment. Small trickle holes on both sides of the reserve compartment feed oil back at a small rate into the circulation. The capacity of the reserve compartment is 10 gal. The capacity of the pressure pump compartment is 7 gal and represents the minimum oil quantity at which the engine can be safely operated. During the overflow of the surplus oil and the stagnating period in the reserve compartment, the oil foam generated during the circulation can disintegrate, thus preserving the load-carrying capacity of the oil. The air is discharged through a vent on top of the reserve compartment.

The lube oil filters and control valves are located in the filter housing which again forms an independent unit mounted in the engine "V" on the damper end. The filters and valves are all accessible from the front end of the engine. A wire-mesh filter of 80 micromm and a paper-fine filter of 25 micromm are arranged in series, thus preventing the paper-fine filter from being loaded with particles of a size greater than 80 micromm. The wire-mesh filter feeds the main oil galleries while the paper filter feeds the turbochargers and injection pumps. Both filters are supplemented by bypass valves for appropriate pressure levels. With this arrangement, the filter size and maintenance requirements were minimized. In case that replacement paper filters are not available, the engine is still protected by the wire-mesh filter, which can be easily cleaned.

Fuel Injection System: The two fuel injection pumps are located in the engine "V" between the fan towers (Fig. 15). This arrangement results in easy accessibility from the top as well as in engine compactness.

Conventional in-line, 6-cyl American Bosch APE 6BB injection pumps with 12-mm plungers were chosen and produce desirable injection characteristics from 600 to 2600 engine rpm. Uniform and short in-

jection lines were possible as a result of the central location of the injection pumps, a considerable asset on a high-speed engine.

A special nozzle holder has been developed, but contains a standard American Bosch ADB type nozzle with eight holes of 0.3-mm holes.

The fuel delivery pump is of the positive displacement type and is mounted on top of the gear drive for the generator. It delivers fuel at a pressure of 25 psi and at a rate of 200 gal per hr to the injection pumps. This rather high pressure and delivery rate has been chosen in view of preventing vapor formation when the engine is operated on JP-4 or kerosene.

The automatic timing unit is located ahead of the two injection pumps. A hydraulic vane type is under development. However, where the torque varies as the square of the speed as in a torque converter transmission, a timing change with load will probably satisfy the engine. This can be obtained with a upper helix plunger.

Exhaust Turbochargers: Dual turbochargers are provided because of space and package limitations. They are located outboard of the engine at the flywheel end, one on each side. This is appropriate for simple duct connections in all directions in the military vehicle, but other arrangements can be readily provided. The turbochargers are aircooled and specially developed for the purpose. Low rotating inertia of the rotating assembly, good component efficiency, light weight, and simple low-cost design were the considerations used in the selection.

Manifolds: The exhaust manifold system is divided in four groups of 3 cyl each leading to corresponding nozzle divisions in the turbines. By this arrangement the scavenge process is improved and interference between the cylinders prevented.

The intake manifold is of the spider type with individual manifolds of nearly the same length and of appropriate diameter, leading from a junction box to each cylinder. This arrangement assures an even distribution to all cylinders, thus permitting maximum utilization of the combustion air. It may be mentioned that by this arrangement together with the constant-velocity stiff injection pump drive already described, the exhaust temperatures of the 12 cyl can be maintained within a tolerance of 50 F.

Accessories: Starting is performed with an electric starter of 18 hp at 24 v. The ratio between engine and starter is 1/12, and the obtained starting speed at room temperature is 150 rpm. The starter pinion is manually shifted to assure trouble-free cold start.

A standard generator of 300-amp capacity at 24 v is driven at a ratio of 1/2.6 from the engine.

Fuel filters are of the conventional type.

Development and Performance

Development Plan—The multitude of new design features and uncertainties made it necessary to follow a development policy of separation of problems from the whole in order to obtain certain critical experiences early enough to influence the end product. The salient objectives were:

1. Anticipate probable troubles by separating items which have little or no background of knowl-

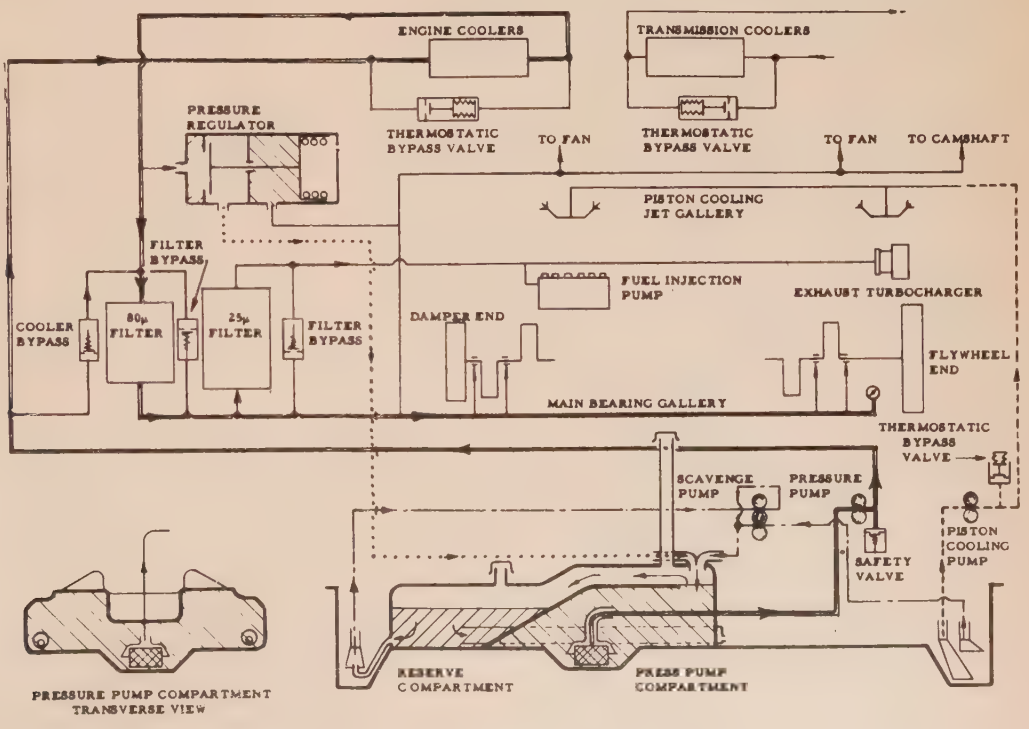

Fig. 16—Lubricating oil system

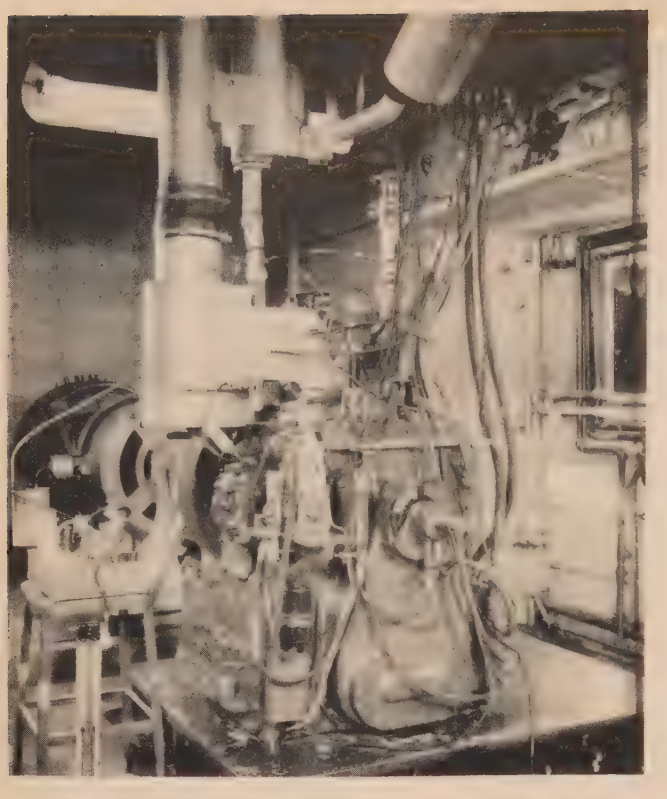

Fig. 17—CUE single-cylinder test installation

Table 1—Engine Development Problems

Single-Cylinder Test Engines	Combustion, injection system, aircooling, piston oil cooling, piston and rings, breathing, cylinder structure, connecting rod and bearings, valves and valve gear, basic durability, basic performance, high output exploration
V-Twin-Cylinder Test Engine	Main bearings, crankcase structure, camshaft drive, injection system drive, timing device, endurance qualification
Components on Bench Test	Injection system performance and endurance, oil pump performance, oil system at high tilt angles, cooling fans, cylinder-head models and actual heads, stress analysis of crankshaft, crankcase, connecting rod, pistons, piston fatigue testing
12-Cylinder Engine	Overall performance and endurance, manifolds, exhaust turbocharger adjustment, cooling airflow distribution, crankshaft torsional vibration, heat rejection, oil system functioning, injection distribution, accessory drives

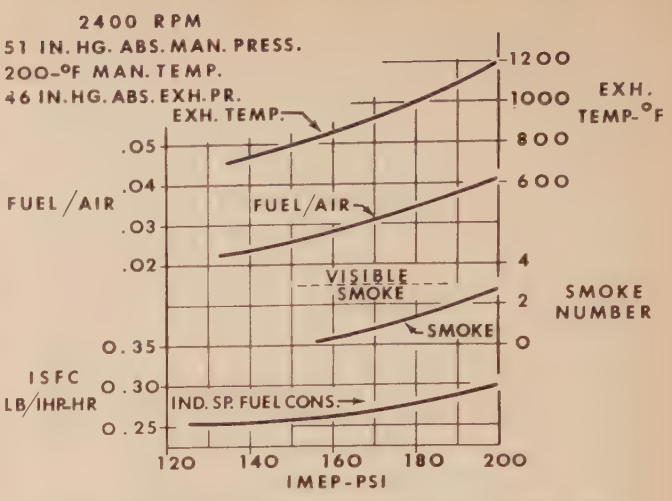

Fig. 18—Single-cylinder performance at 2400 rpm

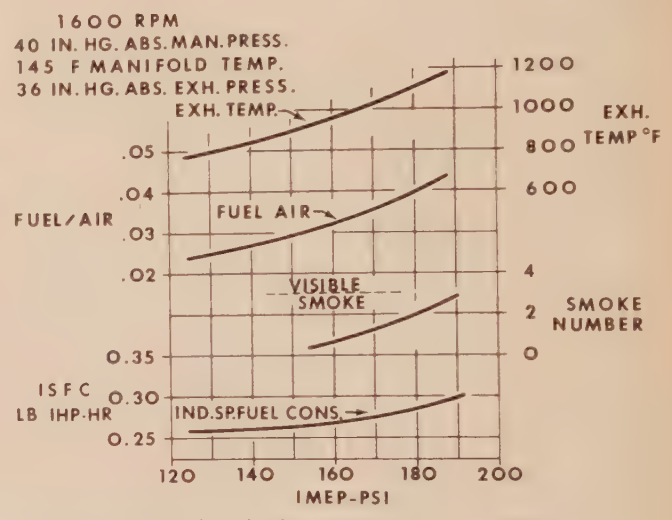

Fig. 19—Single-cylinder performance at 1600 rpm

edge from ones for which a background of knowledge exists.

2. Isolate problems from the whole by running components independently of and prior to testing of the whole engine.

3. Concentrate the multicylinder engine development only upon problems which are peculiar to it and cannot be separately investigated.

To accomplish the development in this manner, provision was made for two single-cylinder engines, a V-twin test engine, numerous bench-test setups, static and dynamic strain measurements, and fatigue test apparatus.

Important problems to be solved were divided as shown in Table 1.

Single-Cylinder Performance—Cooperative Universal Engines (CUE) are used for single-cylinder testing as shown in Fig. 17. It was possible to duplicate multicylinder test conditions to within a few per cent by estimates of friction and by prior knowledge of usual turbosupercharger performance on an engine. Testing is based upon indicated performance, thus eliminating the peculiarities of the test engine. Typical performance curves versus load at the constant speed of 2400 rpm are shown in Fig. 18 and at 1600 rpm in Fig. 19. An abbreviated plan is used wherein the full-load manifold pressure and temperature and back pressure for a given speed are used for all testing at that speed. Strict conformity with the multicylinder operation at part load was found unnecessary. A paper-filter-type smokemeter was used throughout the program and is considered essential for performance evaluation.

Few if any injection system components or systems were run on the test engine without prior test-bench research. The extreme dependence of the engine upon detail injection system performance is

well-known and cannot be overemphasized. Fig. 20 shows the performance of the injection system now in use on the V-12 engine.

Adequate cooling is high on the list of important problems solved on the test engine. Fig. 21 shows typical temperatures measured on the cylinder assembly at full power. The result is notable for the low temperatures as well as for the uniformity of the temperature distribution over the cylinder head. Another important item to be investigated was the piston cooling. The piston cooling jet flowing 1.5 gpm per cyl reduced the temperature at the top ring over 100 F from an uncooled piston. The full power temperatures with cooled pistons are shown in Fig. 22. These were measured by means of an aluminum hardness survey of the piston after a 50-hr steady full-load test run. Fig. 23 shows a photograph of a piston after over 500 hr of cyclical endurance testing.

Routine testing is conducted at 30 to 40% in excess of current rated power. This not only permits accelerated component evaluation but also proved the already mentioned overload capacity of the turbocharged diesel engine.

Component Testing—Extensive component development successfully shortened the testing program on the V-12 engine. Some examples of this work may be mentioned as a matter of interest.

The V-twin test engine of Fig. 24 was used to evaluate main bearing conditions and accessory drives.

The complicated lubricating oil system was set up on a test bench and tilted 30 deg in all directions in full operation. Many potential V-12 oil system faults were corrected in advance.

A careful evaluation of the cylinder dome was made in view of the high loads to be carried. Models

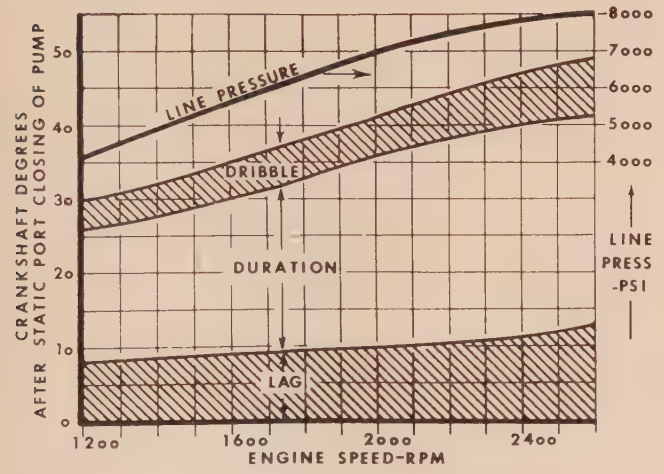

Fig. 20—Injection system characteristics

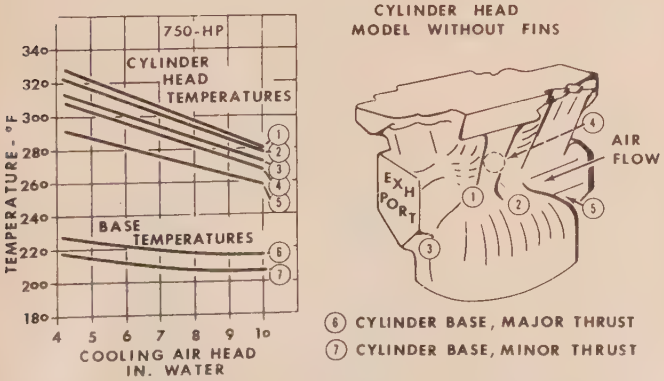

Fig. 21—Cylinder cooling of 1790 diesel

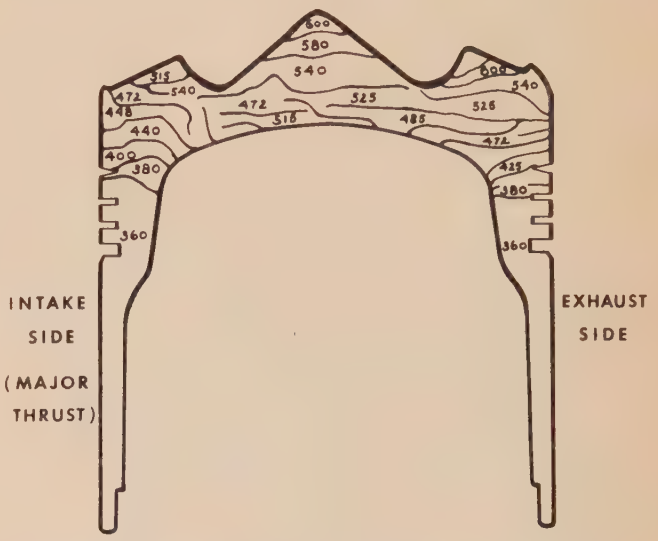

Fig. 22—Piston temperature at 750 hp of 1790 diesel

Fig. 23—Piston after 500-hr operation

as well as cylinders were used in stress-coat and strain gage analysis (Fig. 25).

Similar development was performed on the connecting rod which must handle heavy reciprocating loads at high speed (Fig. 26). The question of connecting-rod bolt strength was resolved by means of a strain gage installation. Fig. 27 shows how optimum bolt torque was established by plotting bolt stress range on a modified Goodman diagram to compare with experimentally determined allowable stress ranges. Optimum bolt stretch was determined and adequate margin below failure stress was shown. For a final check, the connecting rod was run at 40% overload in tension on a fatigue machine without encountering failure.

Multicylinder Performance—The first V-12 engine produced its rated performance within a week after completion of its green run. All performance expectations have been met or exceeded. A full-load power curve is shown in Fig. 28, and a fuel consumption map in Fig. 29.

The well-known "lugging" problem on turbocharged diesel engines could be satisfactorily solved on the AVDS-1790 diesel. A bmep of 145 can be obtained at 1600 rpm (65% of rated speed) without smoke, which meets the requirements of the lock-up transmission of the planned installation.

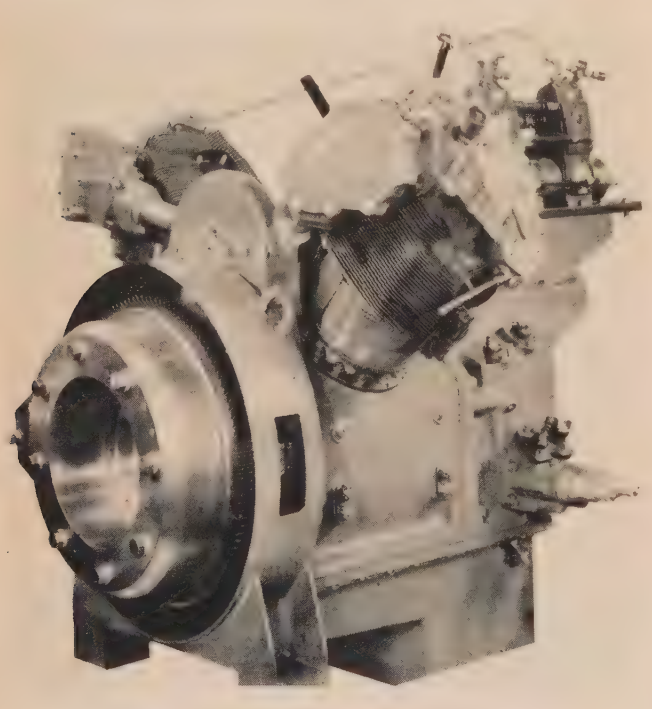

Fig. 24—Twin-cylinder test engine

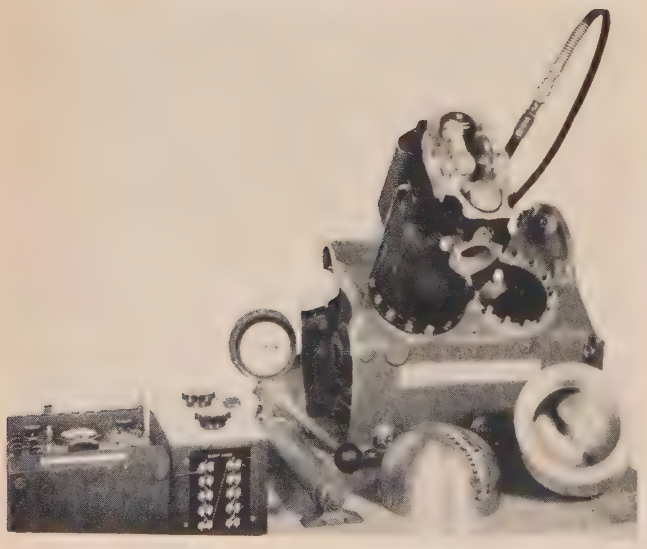

Fig. 25—Cylinder-head stress analysis

Evaluation

The complete powerplant performance is closely related to the installation and cooling condition in a specific vehicle. There is some difficulty in separating the bare engine specifications from the complete package, and the table of data must be viewed with this in mind. An attractive specific weight and bulk has been achieved with 5.0-lb per hp bare engine and 10.0 gross hp per cu ft of installed unit-cooled power package.

Furthermore, it should be mentioned that the 750-hp bare engine rating provides the same net installed power as the 825-hp unsupercharged gasoline engine by virtue of three items:

1. The diesel-engine heat rejection is less than the gasoline engine, and the cooling power is thus smaller.
2. The turbocharged engine requires no exhaust muffler and has no loss from this source.
3. The turbocharged-engine power is not primarily affected by induction air cleaner losses as the tur-

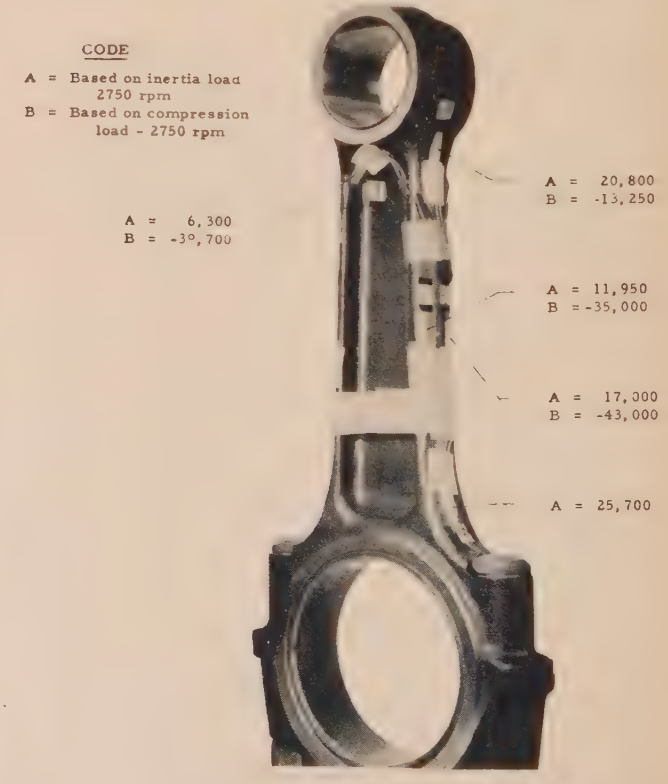

Fig. 26—Connecting-rod stresses

Table 2—Installed Engine Performance Analysis

	Diesel Bare Engine (Uncorrected)	Gasoline Bare Engine (Corrected)	Diesel Net Installed	Gasoline Net Installed
Gross Horsepower	750	825		
Power Losses				
Exhaust Muffler				25
Air Cleaner				30
Generator				10
Cooling Fan[a]			10	135
			110	
			120	200
Net Installed Horsepower			630	625

[a] Cooling fans are for unit-cooled installation, cooling engine cylinders, engine oil, and transmission oil on a 125 F ambient day, while installed in a vehicle hull.

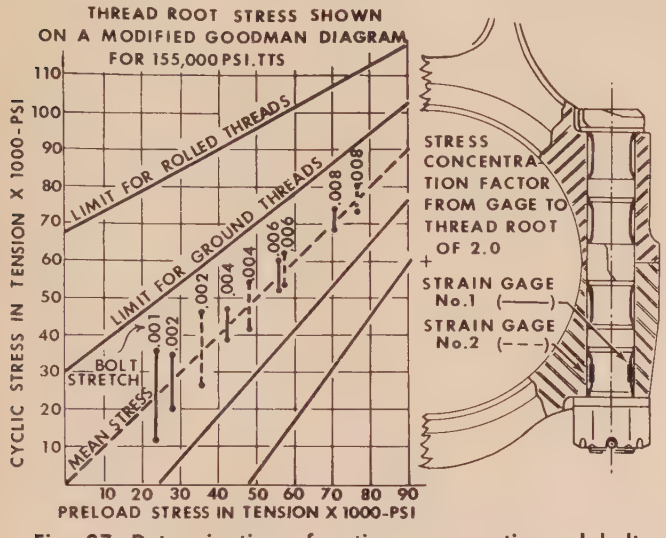

Fig. 27—Determination of optimum connecting-rod bolt stretch

bocharger merely operates at a slightly higher pressure ratio.

Table 2 gives the pertinent data of the AVDS-1790 aircooled diesel engine compared with the gasoline engine. Table 3 gives full data on the engine.

Acknowledgment

The interest and support of the Department of Ordnance, U.S. Army, and particularly of the Detroit Arsenal is gratefully acknowledged. The enthusiasm of the Continental Aviation and Engineering Corp. management as well as that of the fine development team of engineers was equally necessary to the success of this project.

DISCUSSION

Discusses Features Of Aircooled Diesel Engine

—P. H. Schweitzer

Pennsylvania State University

THIS paper is interesting as it is a report on an outstanding technical accomplishment. It is the first time that a diesel engine of this size has been aircooled, and its reported performance is way above the average. A specific output of 0.42 hp per

Table 3—Data on AVDS-1790 Aircooled Diesel Engine

Type	Four-stroke turbocharged 90 deg V 12-cyl aircooled diesel engine
Bore and stroke	5.75 in. x 5.75 in.
Displacement	1790 cu in.
Rated power gross	750 bhp at 2400 rpm
Peak torque gross	1740 lb-ft at 1600 rpm
Bmep gross	147 psi at 1600-rpm peak torque 139 psi at 2400-rpm rated power
Fuel consumption gross	0.38 lbs per bhp-hr at full power 0.35 lbs per bhp-hr minimum 7-lb per hr idling at 700 rpm
Engine weight	3800-lb bare engine 4200-lb fully equipped as unit-cooled package
Specific weight	5.0-lb per bhp gross (bare engine)
Engine size	66 in. long 41 in. high 55-in. basic width with turbochargers protruding locally
Induction airflow	2000 cfm
Cooling airflow	22,000 cfm for the unit-cooled package
Starter	18-hp 24-v electric starter or hydraulic starter
Generator	300 amp, 24 v

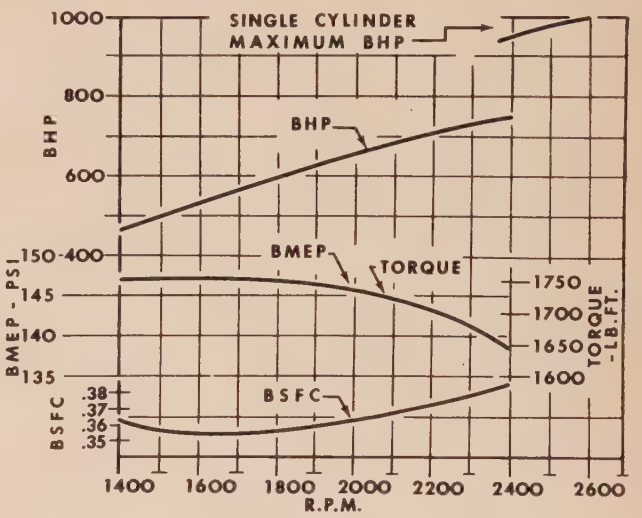

Fig. 28—Full-load performance

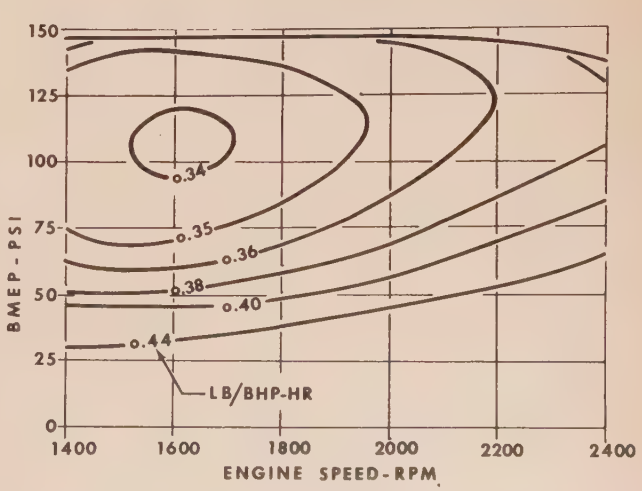

Fig. 29—Full consumption map

cu in. and 139-psi bmep would be creditable performance for a water-cooled turbocharged engine with intercooler. In fuel consumption, weight, and size, it excels any engine of its class which the writer knows about.

Noteworthy features are the structural stiffness of the crankcase, the overhead camshafts and absence of pushrods, the suction-type cooling fans and cowling assuring adequate aircooling of a 5⅜-in. bore engine, and the unique dry-sump lubrication system.

Additional information would be welcome on compression and firing pressures, rate of pressure rise, and other combustion characteristics of the "quiescent" combustion chamber which has seldom been used so far successfully in engines of such speed.

Questions Performance Features of Aircooled Engine

—A. F. Underwood
General Motors Corp.

THIS PAPER has many interesting points. It shows that a diesel engine need not be large and heavy. Even with aircooling and a moderate bmep, the engine is quite comparable to the gasoline engine which it can replace. Also if the piston shown in Fig. 23 is typical of the endurance results, one would expect that the endurance life is quite satisfactory for an engine which has been developed so recently.

The authors do not give the compression ratio, and it would be interesting to know what the mechanical compression ratio actually is. If it is in the usual range of values, it again illustrates the fact that the design of a modern diesel is not very different from the design of a modern high-compression-ratio gasoline engine. Both are approaching the same firing pressures and rate of pressure rise. It would be interesting to know what fuel is to be used in the engine; for instance, are JP-4, JP-5, and No. 2 fuel oil satisfactory. How much of the improvement shown in the upper part of Fig. 1 is due to the increased number of Btu's per gal in diesel fuel compared to gasoline. Is the fuel consumption shown in the lower part of Fig. 1 based on gross horsepower or net horsepower; that is to say, is it figured on the basis of including the fan horsepower?

While it has become modern practice to call such engines "aircooled," it would seem that half of the combustion chamber is now liquid cooled by oil. It would be interesting to know the relative proportion of the heat rejection through the cooling oil and directly from the cylinder heads to the air. We presume that it is difficult to determine how much of the heat from the cooling oil is dissipated through the engine case to the air and, therefore, does not have to be taken care of by the oil coolers.

It is noticed that the connecting rods are designed for a maximum load of 5400 psi. What type of bearing material is employed?

In Table 2 it is indicated that no air cleaner is used. How is the dirt removed from the combustion air?

Wants Reasons for Final Design Choice

—W. J. McCulla
Caterpillar Tractor Co.

DR HAAS AND MR. KLINGE are to be commended for a well-written paper describing the successful attainment of a difficult design objective. Their "divide and conquer" method of attack is certainly the most effective way to produce a new engine with minimum operating difficulties. Components separately tested and developed on bench setups yield valuable preliminary information. They still, of course, require final-proof testing in an engine. Parts developed in the single- or twin-cylinder engine must be further tested in the multicylinder engine.

The connecting-rod development outside the engine is a very logical approach. It is subject only to the danger of assuming that it yields all the answers. Although we have used the same technique, we have had to guard against a false sense of security. We know, for instance, that an otherwise sound rod can experience big end distortion under engine load, leading to fretting of joint surfaces and bearing backs. The roundness of the Continental connecting-rod big end was said to be proved over a wide range of load. Was this proved by a bench test with axial loading only or by an engine test? A 9000-lb horizontal component of load can be deduced from the bearing load diagram in Fig. 11.

A peak unit load of 5393 psi is shown also in Fig. 11. This load at 2500 rpm is a firing load greatly reduced by connecting-rod inertia. Can the authors tell us the peak load at 1600 rpm when the inertia force is less effective?

The authors state that the spider-type inlet manifolding insures an even distribution to each cylinder. Was a simple manifold tried and found to give uneven distribution?

It is common on our diesel engines to heat the inlet air by passing the exhaust of the gasoline-starting engine through a tube in the inlet manifold. The bellows used to compensate for expansion in an early design gave us considerable trouble. Have the bellows used in the exhaust manifolds of the Continental engine given good service?

The design of the piston-pin inserts shown in Fig. 13 is not obvious from the picture. Are these a close fit in the pin, and are they intended to add strength to the pin?

The cold-starting performance described is very good. What compression ratio was used, and what special design, if any, was involved in starting at −25 F without external aid?

The fuel injection lines appear in Fig. 15 to be rather long for a high-speed engine. Has their length or the location of the pumps and lines next to the exhaust manifold caused trouble?

We are quite envious of the low specific fuel consumption of this engine. What equipment was on the engine when the fuel consumption was determined?

No mention is made in the paper of the turbocharger performance other than 2000-cfm induction airflow, presumably at 2400-rpm rated load. Could you give us the pressure ratio and exhaust temperature at 2400- and 1600-rpm rated loads and the airflow at 1600-rpm rated load?

One last question—what grade and weight of lubricating oil is specified for this engine?

Discusses Features Of High-Output Diesel

—R. E. Taylor
Utica-Bend Corp.

THE Packard Motor Car Co. has for years been known for its work in the field of fine engines for aircraft and marine use. This work is now continuing with the same engineering personnel under the Utica-Bend Corp.

In 1950, we started on the design of a family of high-output diesel engines for the Navy minesweepers and have built many of these engines since. The engines are rated for continuous duty at 142 bmep with mean piston velocity of approximately 2000 fpm, or very close in these two respects to the new Continental engines. Our engines weigh between 5 and 6 lb per hp which is also close to the Continental.

At the present time, we have over 1,000,000 hr of service use on these engines and so have an appreciation of durability problems associated with high-output diesels. In the years to come, we expect to see many diesel engines with high specific outputs. The following comments are made with this in mind, since I know many of you have similar problems before you at the present time, or will have in the near future.

Dr. Haas and Mr. Klinge point out that because of the particular space requirements, with which they were faced, the aircooled engine appeared more feasible then the liquid cooled. In many instances, however, the engine size itself aside from the space occupied by radiators or coolers, is the critical factor in which case a liquid-cooled engine would be considerably smaller. The Continental engine has a 5¾-in. bore with a 9-in. cylinder-spacing in order to provide adequate space between cylinders for cooling fins. The liquid-cooled Packard engine has a 5⅜-in. bore with a 7-in. cylinder-spacing resulting in a considerable saving in overall engine length when comparing the liquid-cooled with the aircooled engine. It would seem to me almost impossible to use a precombustion chamber in an aircooled head and still get proper cooling. This fact, undoubtedly, entered into Continental's decision to use direct injection. The peak pressures associated

with direct injection of 1600 to 1800 psi, therefore, might be said to be a necessary part of an aircooled diesel. The Packard engines using a precombustion chamber obtain 142 bmep with peak pressures of 1000 psi and have operated at 200 bmep with peak pressures less than 1200 psi; in this sense then, the aircooling imposes additional loading on many parts of the engine.

The spherical or domed cylinder head used by Continental would also seem to be dictated by the aircooling. This presents a less favorable form of combustion space for compression ignition and also results in considerable increase of piston crown being exposed to the combustion gases.

While aircooling has displaced liquid cooling for most aircraft applications, it is of interest to note that this preference is associated almost entirely with the radial engine. For in-line engines, as associated with most recent airplanes employing piston engines, liquid cooling has been used resulting in minimum space and weight.

Crankcase

The aluminum crankcase design appears to be excellent. We have used the rabbeted, main bearing cap with lateral through bolts for many years with excellent results. We are also using the same aluminum alloy as Continental with a T-71 treatment and would be interested in learning what treatment is used on the Continental 355 aluminum crankcase. In order to obtain optimum roundness of bearing bores, our cases have been machined with the cap studs tightened to their final assembly stretch and the lateral bolts tightened to their final assembly angular position. Although these two methods of controlling bolt loads, that is, the measurement of bolt stretch and the measurement of angle through which the nut has been turned, are excellent, they are expensive and time-consuming. I think we would all be interested in learning whether Continental has found this necessary or whether they have been able to get by with the use of a torque wrench.

Crankshaft and Bearings

We feel that Continental has made an excellent choice in the use of nitrided crankshafts. It has been our experience that nitrided journals and pins show virtually no wear after many hours of service. The extremely high surface endurance limit of the nitrided shaft allows the use of stress risers in the form of oil holes and fillets which otherwise might cause trouble in these high-output diesel engines. We are very much interested in the heavy-wall main bearing used by Continental, since it is of the same thickness as our own. We have stayed with heavy-walled bearings more because of their successful history than because of any shortcomings with the thin wall strip type bearing. The strip bearings offer such a decided advantage from a cost standpoint that we are seriously considering their use in future engines and would be interested in finding out whether they have been tried in the Continental tank engine prior to the use of the heavy wall bearing.

Connecting Rod

Dr. Haas and Mr. Klinge point out that the rod weight has been kept to a minimum through the extensive use of stress-coat, strain gages, and fatigue tests. This is a method familiar to most of you and one that generally gives excellent results. In light of a recent experience of ours, I would like to point out that consideration should also be given to the torsional stresses in connecting rods. In one of our nonmagnetic engines using K-Monel for rod material, we encountered a rod failure in the "I" section of the rod which had all the appearances of a torsional fatigue. In searching for possible sources of torsional stress, it was found that the natural torsional frequency of the piston and rod system corresponded very closely to the natural torsional frequency of the crankshaft!

We have not proven experimentally as yet that the rod and piston are excited torsionally by the crankshaft but, it is a point worth keeping in mind when designing connecting rods.

Piston

The unusual configuration of the Continental piston raises a number of questions. The large surface area above the compression ring exposed to the combustion gases is probably responsible for the high operating temperatures indicated in Fig. 22. Brinell hardness surveys made on our pistons show temperatures 50–75 F less than those of the Continental piston after operation at 200 bmep. To those of you who use aluminum pistons and are acquainted with the extreme loss of physical properties in the neighborhood of 600 F this 50–75 F margin represents considerable additional strength in the piston. The rather thin edges resulting at the contouring of the Continental piston crown appear to be susceptible to thermal cracking.

Intake Valves

Packard has had a similar experience with intake valves in regard to seating on dry surfaces. The use of stellite on the valve seat has reduced the scalloping to a point where it is no longer objectionable. A number of engines have been operated with valve rotators which seem to cure this problem completely.

Lube Oil System

Dr. Haas and Mr. Klinge have pointed out in their discussion of the lubricating system that the Bosch APE-BB pumps are operated with lubricating oil from the engine pressure system. We have used this pump on all of our diesel engines and can appreciate the desirability of flowing lube oil through the pump in order to avoid eventual pump lubrication by fuel oil. This system means, however, that fuel leakage within the pump eventually winds up in the engine crankcase with resultant dilution of the engine oil. The Bosch people have offered a sealed-plunger design which greatly reduces the amount of internal leakage in the pump although it is not completely eliminated. I should like to ask Dr. Haas and Mr. Klinge if they are using the sealed plunger and if they expect difficulty from crankcase oil dilution with its attendant possibility of crankcase fires.

Several questions come up in connection with the oil-cooling jets. Using unfiltered oil to cool the pistons definitely reduces the filter requirements but would seem to introduce the danger of plugging the cooling jet. This has happened to us on a test engine, and we now filter our cooling oil.

From an engine performance standpoint, it would certainly seem advisable to cut off the piston cooling oil when operating at low loads and speeds. When this cut-off is controlled by lube oil temperature it would seem to expose the piston to dangerously low clearance conditions when fully loading a partially warmed up engine. Obviously, this is not a recommended procedure, but these things do happen.

Intake Manifold

The use of a spider-type intake manifold appears to be superfluous on an engine operating at 50 in. of Hg absolute intake pressure. We have obtained excellent exhaust temperature control with the use of a header-type manifold providing the injection pumps and nozzles are in proper calibration. The branched manifolds would offer some advantage in idle and low-speed operation or if the engine were ever used as a naturally aspirated engine rather than a supercharged engine.

No two engine builders, given the same engine requirements, will come up with the same design. We all have our favorite way of doing things which are based to a large extent upon our past experience. Through excellent papers such as Dr. Haas' and Mr. Klinge's these methods and experiences are presented to all of us for our broader education. I have tried to point out some of the differences between our thinking and that of Continental.

Authors' Closure To Discussion

THE authors appreciate the fine discussions of our paper presented by Messrs. Taylor, McCulla, and Underwood, and are happy to answer their questions as informatively as possible. We will answer Mr. Taylor first.

With regard to cylinder bore spacing, it is recognized that an aircooled engine must have a somewhat larger bore spacing than

Table A — Turbocharger Performance Numbers

	2400 rpm	1600 rpm
Airflow	8100 lb/hr	4300 lb/hr
P_2/P_1	1.90	1.42
Exhaust Temperature	1160 F	1190 F

Table B — Recommended Viscosities

Oil	Temperature Range, F
SAE 30	+20 to +120
SAE 10	−10 to +20
Arctic	−50 to 0

a comparable liquid-cooled engine, because of the cooling fins mounted on the cylinder head. This bore spacing, however, is also frequently influenced by the required bearing areas, space for crank checks and counterweights, space for satisfactory main bearing structure, and the bore-stroke ratio. The overall bulk of the aircooled engine, with all cooling included, is definitely less than the comparable liquid-cooled engine including the cooling system with radiator and fan. This reduced bulk of the aircooled engine is especially significant when cooling in severely restricted space at high ambient temperature is required. Under these conditions the effective differential temperature between the air and the coolant in a radiator is reduced by a much larger percentage than the differential temperature between the air and the air-cooled cylinder head. As a result, liquid cooling radiators, fans, and air ducting grow considerably in size and create the well-known difficulties of liquid-cooled engines installed in armored vehicles which operate in the desert.

We are happy to agree that the open combustion chamber chosen was important in achieving an excellently-cooled cylinder. That a precombustion chamber could not have been used in the aircooled cylinder is open to question in view of the engines which have been built this way. We refer in particular to the air-cooled Deutz engines, the Caterpillar 9-cylinder radial, 500-hp, aircooled diesel, and an Austrian Simmering-Graz-Pauker, 700-hp, aircooled diesel (another successful tank engine development during the war). It is a matter of fact that the open combustion chamber entails a somewhat higher peak combustion pressure than the precombustion chamber. The magnitude of the difference is not correctly stated, possibly because of no direct data on this matter in the paper. At comparable output, the open-chamber will have a 15 to 20% higher peak pressure than the precombustion chamber.

With respect to the domed cylinder head, we have had excellent results and consider it as a way to obtain large valves, high-speed breathing capacity, and a sound head structure.

The crankcase is 355 aluminum with a T-71 heat-treatment. We are not aware of a problem in this regard. The main bearing studs are currently stretched and we do not know at this time whether a less expensive procedure can be substituted.

The spark-ignition tank engine has used thin wall strip bearings for main bearings for many years with excellent results. An occasional difficulty has been encountered in "wrinkling" the shell under arctic conditions when the aluminum crankcase crushes in from differential thermal expansion. The diesel is using heavy wall shells partly to avoid this crushing trouble, partly to keep bearing clearances when very cold, and partly because of anticipated trouble in stabilizing thin wall shells with the diesel's very high loads.

We appreciate the warning from Mr. Taylor on connecting-rod torsional failures. Static twisting of the connecting rod in the laboratory is commonly tried, but we have had no way to evaluate this kind of test as yet.

No doubt the distribution of heat in the piston Mr. Taylor mentioned is quite different than in ours. The high temperature of the protruding parts of our piston dome did concern us, but the fact that these are unstressed areas has probably permitted running without any difficulty of any kind, even at 200 bmep. We would like to point out that our ring belt is quite reasonable in temperature, as is the under-dome temperature. The piston runs without lacquer formation except above the second ring, even when using MIL-O-2014B low detergent oil. No thermal cracks on the piston crown occurred.

We also use the sealed plunger design of American Bosch, and for us this design has stopped plunger leakage. Our experience is limited, however, and we appreciate the point taken regarding potential leakage into the crankcase.

With regard to intake manifolds on a supercharged engine, there can be many discussions. We have found the branched type induction manifold sufficiently better at lower and higher speeds than a straight runner manifold.

We would like to thank Mr. Taylor again for his stimulating and informative questions. Now to answer Mr. McCulla.

We appreciate the problem of a false sense of security from component testing. Extensive engine testing of the connecting rod has corroborated the bench results to a large degree.

Peak connecting rod bearing loads at 1600 rpm are about 10% higher than at top speed. The reduction in inertia load at lower speed is partially compensated by a concurrent reduction in peak gas loads.

The bellows in the exhaust manifold cover a slip joint which supports the pipe. Trouble-free operation has resulted.

The piston-pin plugs are a free fit in the pin and float at all times. No piston-pin support results. However, considerable attention has been given to a rigid piston and piston-pin structure.

The information on cold starting is somewhat incomplete in the paper. The engine will start readily at −25 F with starting fluid. The use of external aid in the form of heaters and special equipment is not necessary. Starting fluid is required below +15 F when kerosene is used. The compression ratio is 14.5/1.

The injection pumps and lines nestled about the exhaust manifold have caused no trouble at all due to the tremendous volume of cooling air passing over this part of the engine. Their length is 30 in.

The fuel consumptions of Figs. 28 and 29 are shown for an engine without cooling fans. Net installed figures are not shown for an engine without cooling fans. Net installed figures are not shown because they vary considerably with the various installations.

The turbocharger performance numbers requested by Mr. McCulla are about as shown in Table A.

Lubricating oil conforming to a low detergent level of MIL-O-2104B has been satisfactory throughout. Viscosities recommended are as shown in Table B.

We thank Mr. McCulla for his comments, and turn to Mr. Underwood's discussion.

Mr. Underwood states that he believes that the modern diesel-engine design is very similar to the modern gasoline engine. We would like to point out again that the subject diesel engine is not a conversion of its gasoline-engine counterpart but a complete new design. We do not believe that a spark-ignition engine can be converted to a high-performance diesel engine and still obtain the same power in the same space.

The present engine has shown the ability to use Nos. 1 or 2 diesel fuel, or JP-4. Extensive service experience may show some limitations, but we are not aware of them at the moment.

The upper curve of Fig. 1 showing fuel consumption gain of the diesel engine over the gasoline engine, is shown in miles per gallon, which includes about 15% difference in fuel heating value per gallon. The lower part of Fig. 1 shows the data in weight flow, so that one may use whatever ratio is best suited to the situation. The lower fuel consumption curves are based upon gross horsepower, without cooling. Similar curves based upon net installed power with all cooling included, show even larger gains for the diesel engine.

In regard to heat rejection, about 20% of the total measured heat rejection comes from the lubricating oil coolers. At the moment, we are not in possession of a more detailed breakdown of the total heat rejection.

The connecting-rod bearings are plated copper lead bearings, type F77, from Cleveland Graphite.

Mr. Underwood has mistakenly concluded that since no air cleaner power loss is shown, no cleaner is used. Conventional air cleaning practices must be followed, except that the flow loss through the air cleaner does not reflect proportionally into power loss. The turbosuperchargers can be readjusted to provide the required airflow at a somewhat increased pressure ratio, and the engine suffers only slightly in the process.

HIGH-SPEED TURBOPUMPS

J. E. Boretz, Fairchild Engine and Airplane Corp.

This paper was presented at the SAE Metropolitan Section Meeting, Hoboken, N. J., Oct. 18, 1956.

SELECTION and general design considerations of the components for high-speed, high-performance turbopumps are discussed here. A simple mathematical example of aerodynamic theory is presented for pump and turbine design.

The author discusses choice of design for the specific performance desired. Considerations of numerous factors influence selection.

Example of a typical high-speed turbopump is given based on the theory developed here.

THE best designs of a pump and turbine combination for aircraft or rocket-engine applications are those which are capable of producing the required discharge pressures, flows, and output power at low dry weight, low pump suppression heads, and high efficiences. These components should be integrated into a simple and compact turbopump unit which is reliable in operation and simple to install and maintain. As to how the turbine or pumps are to meet these requirements depends as much upon the specific application as on their design. In many instances it is not possible to achieve both minimum weight and optimum efficiencies for a given unit. The governing parameters of a specific application include flight duration, operating life, available net positive suction head, specific speed, turbine pressure ratio, and type of turbine drive utilized.

In addition, for fuel booster pumps or transfer pumps the ability to handle a boiling liquid becomes a requirement. Finally, for cryogenic pump applications, the handling of low-temperature liquid gases introduces gas-separating or recompression problems.

In the case of the use of a turbopump for short-duration (1-min) rocket engine operation, pump and turbine efficiencies are of secondary importance compared to minimum weight (Fig. 1) unless the unit is jettisonable. For in this application the expenditure of turbine-driving fluid occurs over a relatively small portion of the flight duration so that aircraft or missile performance is most adversely affected by turbopump dry weight. However, in those cases when an air-driven turbine is used with a limited supply of bleed air, higher combined turbopump efficiency may be required at the expense of dry weight.

For longer duration operation (approximately 5 min) combined turbopump efficiencies must be relatively high, and some sacrifice in dry weight is permissible. The saving in weight of the turbine-actuating fluid in this case will generally more than compensate for the increased dry weight required in order to obtain the high efficiencies. It is clear that for further increases in turbopump running time, the dry weight of the unit becomes a comparatively insignificant factor in the overall weight consideration, and com-

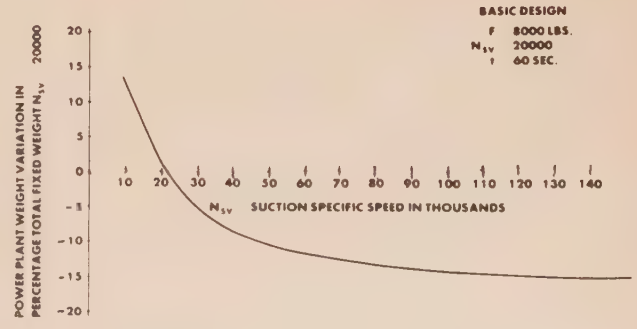

Fig. 1—Curve showing total powerplant weight variation as function of N_{sv}

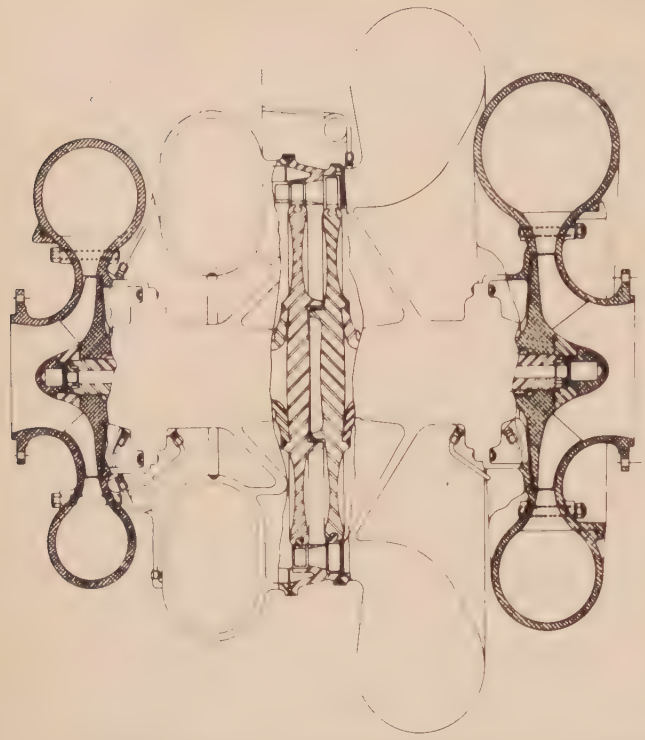

Fig. 2—Typical turbopump arrangement with two overhung pumps

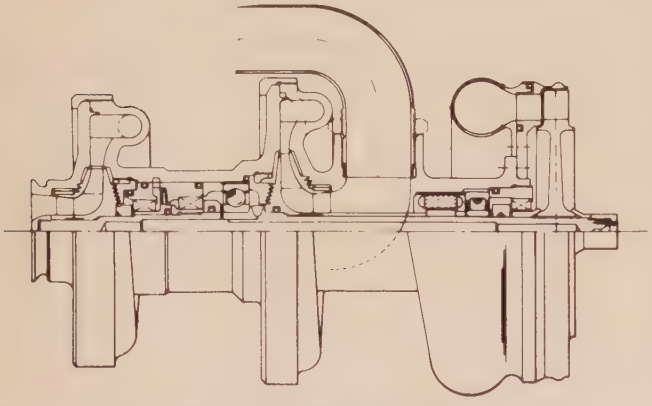

Fig. 3—Typical turbopump with overhung turbine exhausting to atmosphere

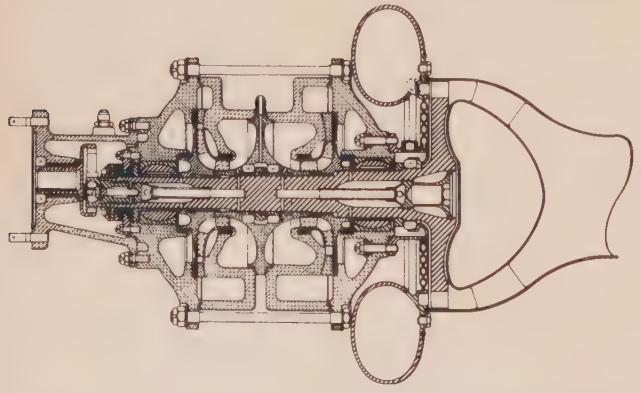

Fig. 4—Typical turbopump with overhung topping turbine

ponent efficiencies become of prime importance.

The selection and general design considerations of components for high-speed, high-performance turbopumps are discussed in this paper.

General Considerations

The first problem the engineer is confronted with in the design of a turbopump unit is the selection of the general arrangement of the components. When only one pump is involved, this problem usually resolves itself into whether a direct drive or a transmission-type drive is to be employed. Furthermore, for numerous applications, a high-speed electric motor or a hydraulic motor may be substituted for the turbine drive. These latter type drives are particularly suitable for fuel booster pumps which quite often operated submerged in a fuel tank. However, for larger horsepower requirements (approximately above 15 hp), the high-speed turbine usually results in a considerably lighter unit. Limiting the discussion to turbine drives, a direct-connected unit usually results in the lowest weight unit. However, suction specific speed requirements of the pump and turbine efficiency objectives will very often determine whether this is desirable or even feasible. When two pumps are involved, as is usually the case in rocket engine applications, additional considerations arise. For this case there are basically two general arrangements employed. Fig. 2 indicates a typical configuration employing two overhung pumps with the turbine mounted in the middle. This arrangement permits the use of two "open-eye" pumps which generally are more suitable for operation with small net positive suction heads. Furthermore, it effectively separates the fuel pump from the oxidizer pump thus minimizing the hazards of fire or explosion. However, it does have the disadvantages of requiring a fairly heavy turbine discharge gas volute and places both pumps in close proximity to the quite often very hot turbine

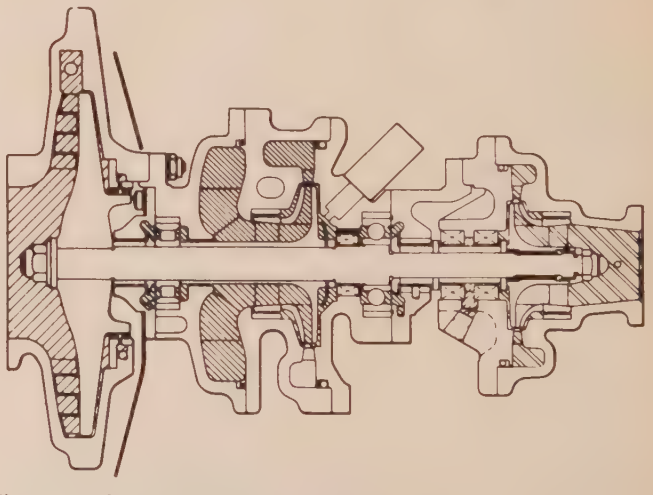

Fig. 5—Light-weight (22 lb) turbopump unit with LOX and kerosene pumps and 2-stage radial turbine

and turbine gases. This may be undesirable if liquid oxygen, liquid hydrogen, or any other low-temperature fluid is being pumped. With regard to these two latter problems, the arrangement shown in Fig. 3 is usually superior. This configuration "overhangs" the turbine on one end of the shaft and places the pumps in a "back-to-back" condition. This arrangement, however, introduces the mechanical complexities of a "closed-eye" pump. Furthermore, because of the close proximity of the two pumps to each other, additional sealing precautions are usually required. However, the hot turbine parts have been effectively separated from the low-temperature pump, and turbine exhaust gases are easily removed through a simple light-weight circular duct.

In addition, for high available pressure ratios, choking of the turbine exhaust to limit operating pressure ratios is more easily accomplished. For rocket engine applications, higher turbine efficiencies may sometimes be obtained by employing a topping-turbine arrangement as shown in Fig. 4. The basic difference between this unit and the previous one is essentially in the operation of the turbine. Low pressure ratio operation of the turbine, whose back pressure is now equal to rocket combustion-chamber pressure, results in high velocity ratios at comparatively low speeds (below 25,000 rpm). This results in a higher turbine efficiency (for single-stage, impulse-type units) and permits direct-drive operation of the pumps at speeds more suitable for high suction specific speed operation.

An example of a light-weight, high-performance unit is shown in Fig. 5. This unit consists of an overhung turbine with a LOX pump and a kerosene pump mounted back to back. The pumps are comparatively low specific speed units ($N_s = 334$ and 472, respectively), but are capable of achieving reasonably high efficiencies (65%) for this condition of operations. The performance attainable with these pumps is shown in Table 1. The high heads and low capacities create considerable design problems in maintaining high pump efficiencies. The pumps operate at a speed of 30,000 rpm. The overall performance for these pumps is shown on Figs. 6 and 7. Based on the positive suction head requirements, the LOX pump operates at a suction specific speed of 18,250. The kerosene pump is capable of operation at a suction specific speed of 20,000. Typical curves of total head versus net positive suction head are shown in Figs.

Table 1—Performance Attainable in Low Specific Speed Pumps

	Pumps	
	Kerosene	LOX
Density, lb per cu ft	50 at (−65 F)	72 at (−297 F)
Flow, gpm	39.8	11.75
Inlet Pressure, psia	11	23
Discharge Pressure, psia	1043	1040
Pump Developed Head, ft	2940	2060
Minimum NPSH, ft	20	10
Fluid Power, hp	24	7.05
Efficiency, %	65	65
Input Power, hp	36.9	10.85
Suction Specific Speed	20,000	18,250
Specific Speed	472	334
Speed, rpm	30,000	30,000

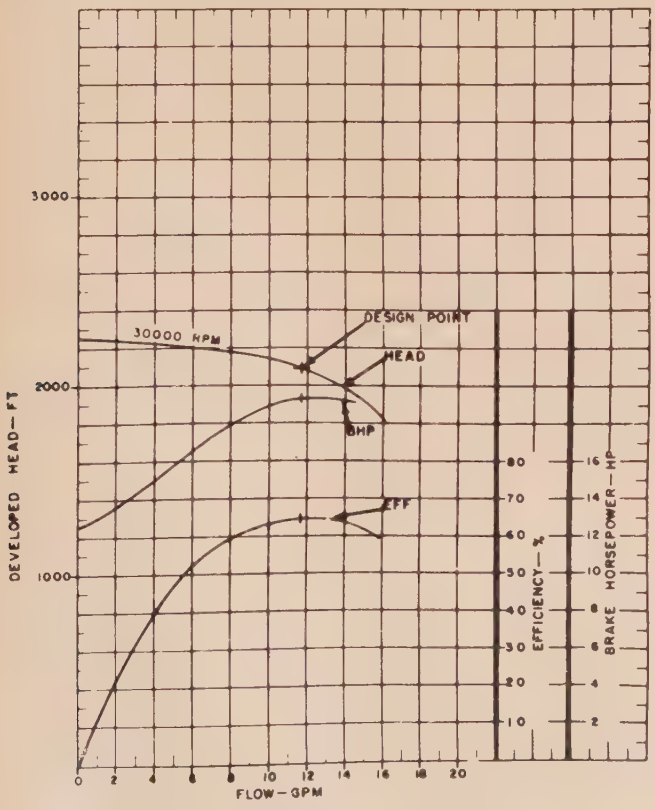

Fig. 6—LOX pump performance

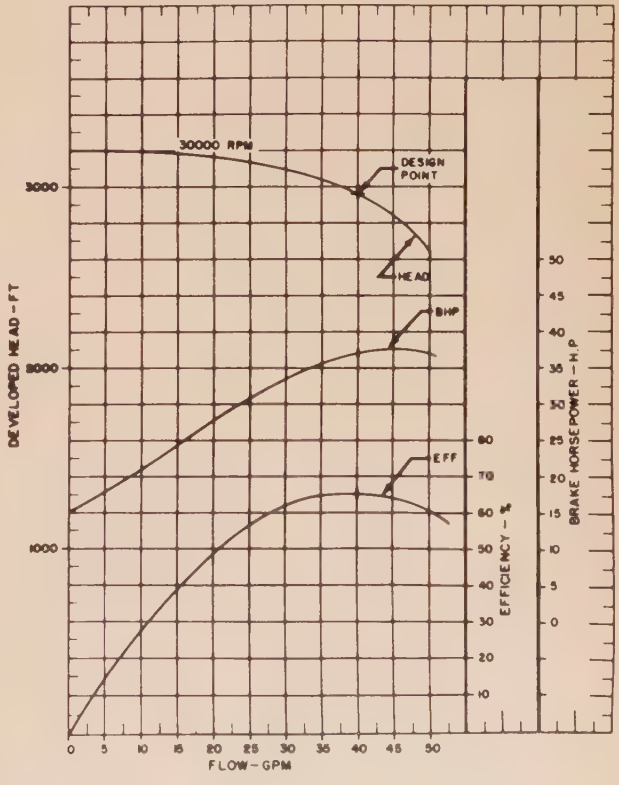

Fig. 7—Kerosene pump performance

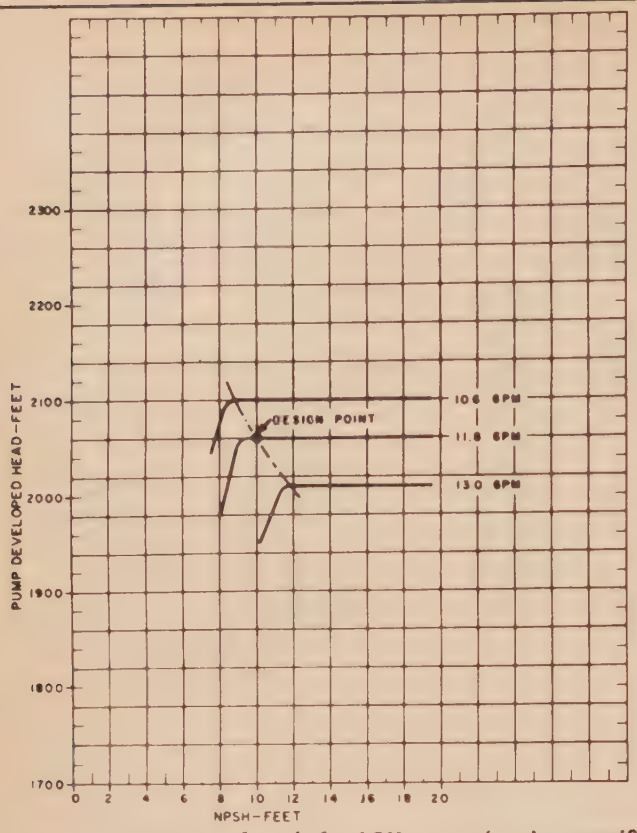

Fig. 8—Requirements of npsh for LOX pump (suction specific speed of 18,250)

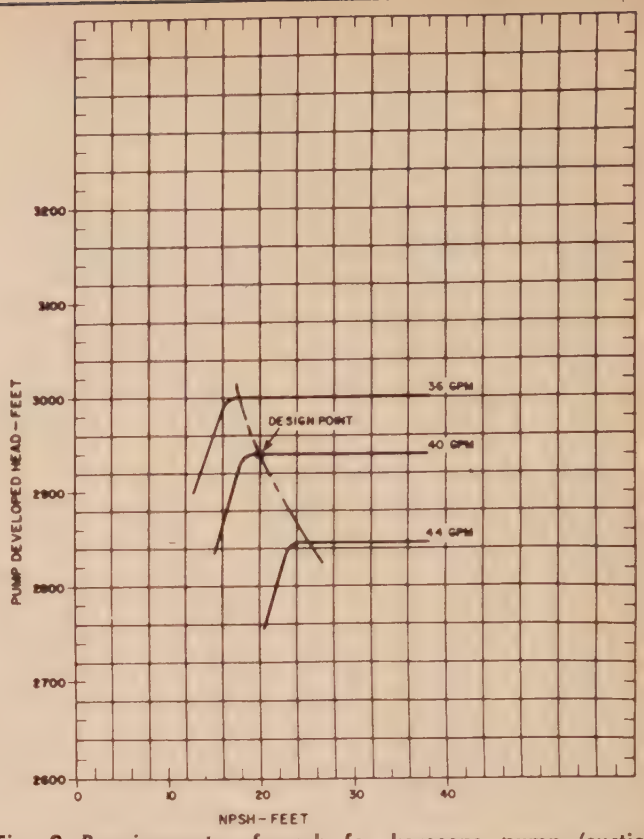

Fig. 9—Requirements of npsh for kerosene pump (suction specific speed of 20,000)

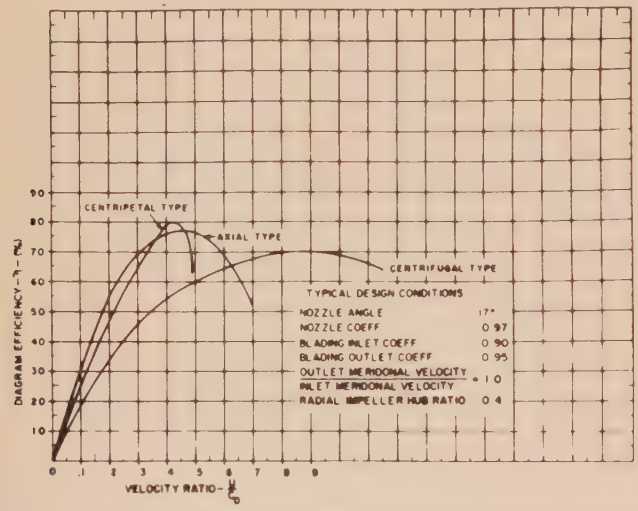

Fig. 10—Diagram efficiency versus velocity ratio of axial, centripetal, and centrifugal turbines for 0% reaction condition

8 and 9. This turbopump employs the use of a 2-stage, velocity-compounded, radial-inward-flow-type turbine. This configuration combines the features of high efficiency at low velocity ratios as well as inherent overspeed protection due to aerodynamic stalling of the blading under runaway conditions. A choked, circular exhaust duct provides flow-limiting features at high pressure ratios. The turbine was designed to operate with inlet gas temperatures as high as 1400 F and with operating pressure ratios up to 20 to 1. The overall weight of this unit was 22 lb. The unit employed a double, face-running, nose seal between the two pumps. Inert gas pressurization between the seals was also used to prevent leakage between the oxidizer and fuel pumps. Precooling of the LOX pump was obtained by jacketing, and means for separating entrained gas in the liquid was provided at the LOX discharge. Kerosene lubrication of the bearings was utilized, and an electric resistance element for cold starting provided. An electromechanical overspeed mechanism, employing a Bellville spring-type trip mechanism and a "microswitch" was utilized. This unit trips at speeds 15 to 30% below that of the turbine stalling speed. The weight, size, and performance characteristics of this unit are typical of the type of turbopump units being utilized in current aircraft and missile applications.

Turbine Selection and Performance

The choice of turbine configuration for turbopump applications varies widely. Such diverse considerations as pressure ratio, inlet gas temperature, velocity ratio, duration of operation, efficiency requirements, control components, matching of power output with pump characteristics, and desired overspeed characteristics are parameters having considerable influence upon this decision. Quite often additional considerations of weight and mechanical arrangement affect the engineer's selection.

Obtaining high efficiencies with high operating pressure ratios usually requires the use of pressure staging. This very often results in a mechanically complex and heavy unit. However, for long-duration operation, where turbine gas consumption becomes an important factor, high efficiency becomes the predominant design criterion. Thus, the complications of split nozzle blocks and sealing between stages must be considered. However, for numerous applications weight and mechanical simplicity become the dominant design criteria with the rate of turbine-driving fluid utilized taking a secondary role. This is most applicable when main engine compressor bleed air or ram air is available as the turbine power source. However, quite often this simpler turbine arrangement is employed when the turbine is driven by a "bootstrap" gas generator and start bottles. As a guide in the selection of turbine type, a good starting point is the diagram efficiency versus velocity ratio curve. These curves, based usually on theoretical velocity diagrams and correction coefficients, give a good indication of maximum achievable efficiencies for a given operating condition. Such a curve is shown in Fig. 10 for three different turbine configurations, namely, axial, centripetal, and centrifugal. The degree of reaction for these units has been set at zero to permit comparison of impulse-type units. As can be seen from this figure, centrifugal- and centripetal-type turbines permit operation at higher velocity ratios and, hence, higher rotative speeds. Also, higher peak efficiencies are obtainable with the radial-flow-type machines than with the axial. A further turbine design choice is the use of reaction. Fig. 11 has been prepared to show the difference in turbine performance when 50% reaction has been employed. As can be seen from these curves, somewhat higher efficiencies are obtainable. Also, since reduced nozzle-spouting velocities result because only 50% of the enthalpy drop occurs in the stationary nozzles, the obtaining of peak efficiencies can be achieved at lower rotative speeds. What is more important from the practical standpoint, where various loads must be carried at constant speeds, is that the efficiency is less sensitive to velocity ratio than in the zero reaction (impulse) type turbine. As a result, reaction-type units can operate off the design condition without too great a loss of efficiency and output.

However, blade clearance between the tips and the casing is more critical in a reaction- than in an impulse-type turbine, since there is a pressure difference across the blade, and available energy may be lost by gas flow through the clearance space. The maximum efficiency envelope for single-stage turbines with either 0 or 50% reaction is shown in Fig. 12. Hence, if peak efficiency at the design point is the major criterion, this curve may be used as a guide to the selection of turbine type.

In those cases where a direct drive is desired and pump performance characteristics limit the use of high turbine rotative speeds, a fair compromise can be achieved with the use of a velocity-compounded impulse-type unit. While overall peak efficiency is

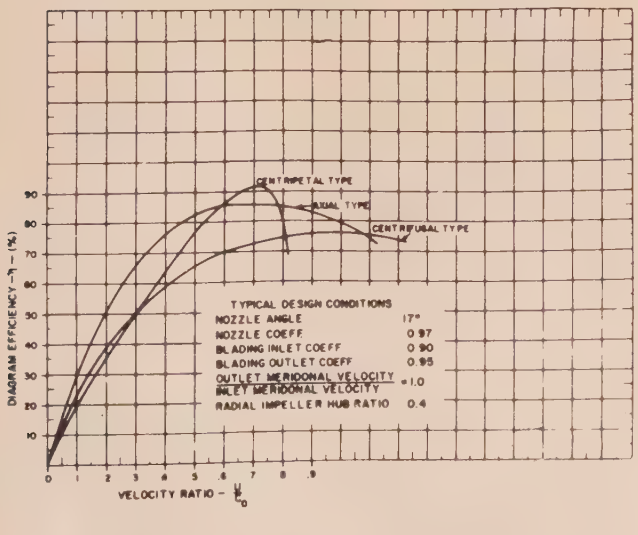

Fig. 11—Diagram efficiency versus velocity ratio of axial, centripetal, and centrifugal turbines for 50% reaction condition

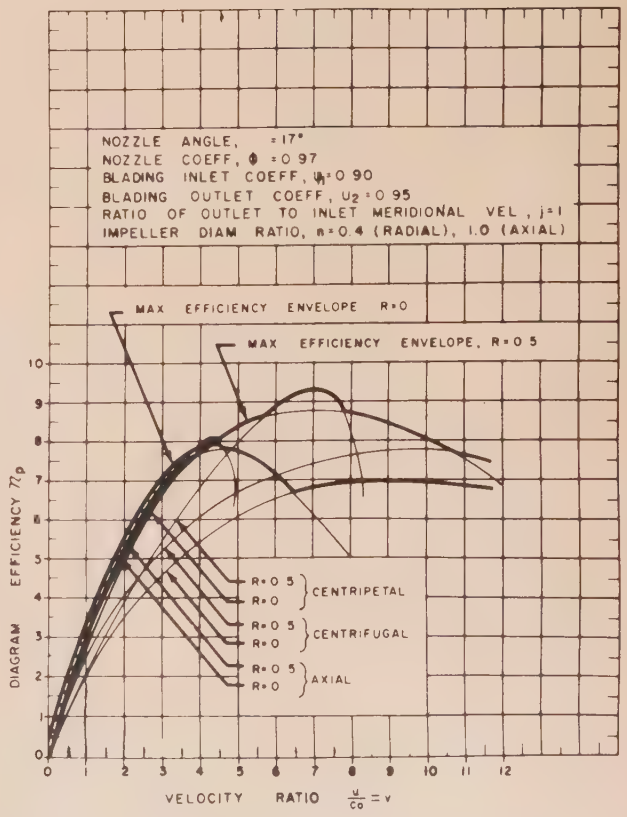

Fig. 12—Maximum efficiency envelope for turbines with 0 and 50% reaction

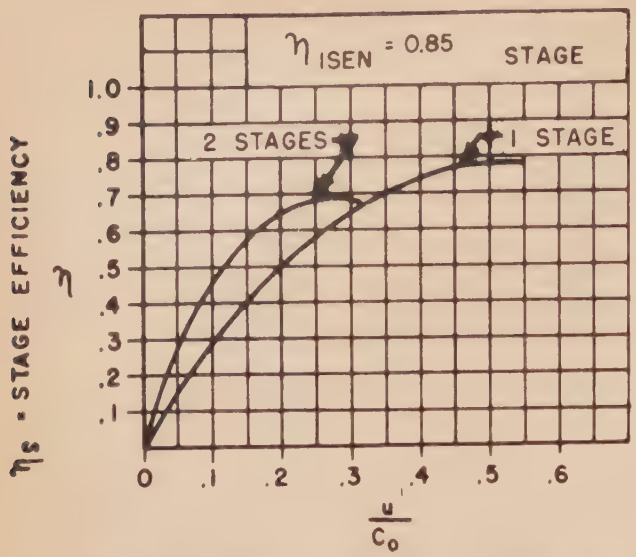

Fig. 13—Effect of staging on efficiency of typical impulse turbine

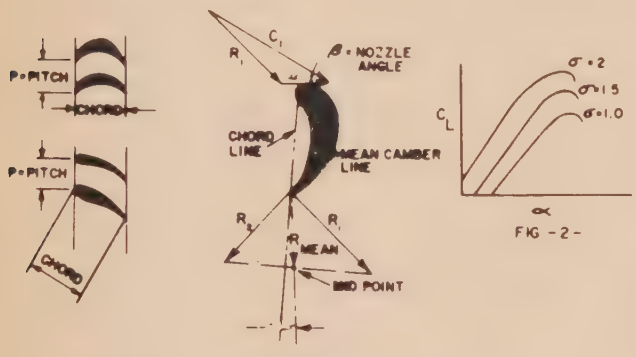

Fig. 14—Aerodynamic turbine blade design theory

somewhat reduced from that of the single-stage impulse unit (Fig. 13), operation at lower velocity ratios and, hence, lower speeds is permissible without too great a lowering of turbine efficiency. Another advantage to the 2-stage velocity-compounded turbine is its inherent overspeed protection if proper consideration is given to its blading design. With this type of feature, maximum runaway speeds can be limited to approximately 170% of design speed, thus providing increased reliability and eliminating the necessity for heavy containment features in the design.

The approach to the design of turbines varies widely. However, most methods usually resolve themselves into the use of momentum (channel theory) considerations or aerodynamic theory. It is the author's opinion that if sufficient experimental cascade data were available, the latter theory would prove more accurate in more nearly defining actual flow conditions. The experimentaly determined coefficients used with the momentum design approach, while producing equally accurate results, provide little in furthering the understanding of the actual pressure temperature and velocity distribution throughout the machine. With the use of the aerodynamic theory (Fig. 14), much currently available NACA data [1] and new mathematical approaches [2,3] may be utilized. In many cases a compromise which uses both theories is desirable and effective.

While the author recognizes that most turbine designers are familiar with both aforementioned theories, a typical simplified example of the aerodynamic design theory is presented for review.

For a single-stage impulse-type turbine operating under the following conditions where $P_1 = 450$ psia, $P_2 = 16.7$ psia, $T_1 = 1960$ R, $\gamma = 1.2$, $M = 29$, and blade velocity at pitch diameter = 800 fps, a typical solution is as follows

Nozzle exit velocity:

$$V_2 = C_v \sqrt{2g \frac{\gamma}{\gamma - 1} \times \frac{1545}{M} \times T_1 \left[1 - \left(\frac{P_2}{P_1}\right)^{\frac{\gamma-1}{\gamma}}\right]} \quad (1)$$

$$= 0.96 \sqrt{64.4 \times \frac{1.2}{0.2} \times \frac{1545}{29} \times 1960 \left[1 - \left(\frac{16.7}{450}\right)^{\frac{0.2}{1.2}}\right]}$$

$$= 3970 \text{ ft/per sec}$$

Velocity ratio:

$$\frac{V_b}{V_2} = \frac{800}{3970} = 0.201 \quad (2)$$

Adiabatic temperature drop:

$$T_2 = T_1 \left(\frac{P_2}{P_1}\right)^{\frac{\gamma-1}{\gamma}} = 1960 \times 0.576 = 1130 \text{ R} \quad (3)$$

Gas density in jet issuing from nozzle:

$$\rho_2 = \frac{144 P_2}{R T_2} = 0.0398 \text{ lbs/per cu ft} \quad (4)$$

Blade velocity triangle (at pitch diameter) is shown in Fig. 15.

For a blade configuration recommended by Weske and for an angle of attack of 6.5 deg, the data of "Weske-Turbine Grid Conference at Braunschweig, Germany in 1938" indicate a lift coefficient of 3.8.

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1 "Theory of Wing Sections," by I. H. Abbott and A. E. Von Doenhoff. Pub. by McGraw-Hill Book Co., New York, 1949.

2 "Theory and Design of Gas Turbines and Jet Engines," by E. T. Vincent. Pub. by McGraw-Hill Book Co., New York, 1950.

3 "On Calculation of Potential Flow Around Airfoil in Cascade," by N. Scholz. Pub. by Technical University, Braunschweig, Germany, 1950.

The solidity, that is, the chord divided by pitch, has correspondingly been chosen as 1.8, to be consistent with the experimental cascade data.

Dynamic pressure

$$q = \frac{\rho_2 W_m^2}{2g \times 144} = 14.6 \text{ psi} \quad (5)$$

where $W_m = 1840$ fps from Fig. 15.

Blade lift requirement:

$$L = C_L A_B q = 3.8 \times A_B \times 14.6 = 55.5 A_B \quad (6)$$

where A_B = area of blading acted upon by gas stream, sq in.

Internal turbine horsepower:

$$\text{ihp} = \frac{L \times V_b}{550} \quad (7)$$

Combining equations (6) and (7) and solving for A_B

$$A_B = \frac{550 \times \text{ihp}}{V_b \times C_L \times q} = \frac{550 \times \text{ihp}}{800 \times 3.8 \times 14.6} = 0.0124 \text{ ihp} \quad (8)$$

A blade chord (width) must now be assumed. For rocket engine applications, the following approximate criteria may be employed: for units from 5000- to 30,000-lb thrust use C = 0.625 in.; and for units from 100,000- to 300,000-lb thrust, use C = 0.750 in.

For a solidity of 1.8, the pitch, p for $c = 0.625$ in. becomes 0.348 in., and for $c = 0.750$ in., $p = 0.416$ in. These results are obtained from the following equation:

$$\sigma = \frac{c}{p} \quad (9)$$

where σ = solidity.

If the blade height is given by h, then the number of blades Z in the stream for the required area (assuming a blade activity coefficient of 0.80) is given by

$$Z = \frac{A_B}{0.80} \times \frac{1}{hc} = 1.25 \frac{A_B}{hc} \quad (10)$$

and the peripheral length of these blades is given by

$$l = Zp \quad (11)$$

where p = blade pitch.

Nozzle dimensions:

If n nozzles are assumed each handling $1/n$ of the total flow W, or w lb per sec, then the throat area per nozzle is given by

$$A_t = \frac{w}{P_1 \left(\frac{2}{\gamma+1}\right)^{\frac{\gamma+1}{2(\gamma-1)}} \sqrt{\frac{\gamma g M}{RT_1}}} = 0.194w \text{ sq in.} \quad (12)$$

The expansion ratio is

$$\frac{A_e}{A_t} = \left(\frac{2}{\gamma+1}\right)^{\frac{\gamma+1}{2(\gamma-1)}} \left(\frac{(\gamma+1)}{2(\gamma-1)}\right) \left(\frac{P_1}{P_2}\right)^{\frac{\gamma-1}{\gamma}}$$

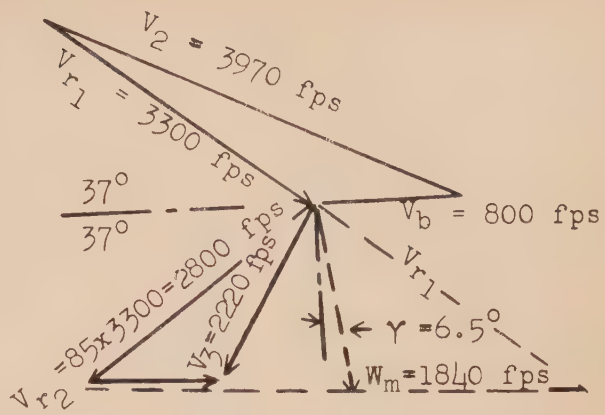


Fig. 15—Blade velocity vector diagram

$$\frac{1}{\sqrt{\left(\frac{2}{\gamma-1}\right)\left[1 - \left(\frac{P_2}{P_1}\right)^{\frac{\gamma-1}{\gamma}}\right]}} = 4.50 \quad (13)$$

Thus the nozzle exit area becomes

$$A_e = 4.50 A_t = 4.50 \times 0.194W = 0.874w \text{ sq in.} \quad (14)$$

If the height of the nozzle at its exit h_n is assumed to be equal to the blade height h and if furthermore the nozzle exit is taken to be square, the side of the square exit is given by:

$$h_n = \sqrt{0.874w} = 0.935\sqrt{w} \text{ in.} \quad (15)$$

Since the peripheral length of the n nozzles should equal the peripheral length of the ideal blades acted upon by the gas stream, then for n nozzles at an angle of 30 deg to the tangential speed of the blades, the nozzle peripheral length is:

$$l' = \frac{nh_n}{\sin 30} = 2nh_n \quad (16)$$

But $l' = l$ and hence,

$$2nh_n = Zp = 1.25 \frac{A_B}{hc} \times p = 125 \frac{A_B}{h\sigma} \quad (17)$$

Substituting for $h_n = h$ and solving for h,

$$h = \sqrt{\frac{1.25 A_B}{2n\sigma}} = 0.59 \sqrt{\frac{A_B}{n}} \quad (18)$$

where σ = solidity, $c/p = 1.8$ (assumed for this case).

Thus the flow per nozzle becomes

$$w = \frac{h^2}{0.874} \quad (19)$$

and the total flow is

$$W_{\text{tot}} = nw \text{ lb per sec} \quad (20)$$

Specific fuel consumption:

$$\text{sfc} = \frac{nw3600}{\text{ihp}} = \frac{nh^2}{0.874} \times \frac{3600}{\frac{A_B}{0.0124}} \quad (21)$$

Equation continued on following page

$$= \frac{n \times 0.59^2 \times \frac{A_B}{n} \times 3600 \times 0.0124}{0.874 \times A_B}$$

$$= 17.75 \text{ lb per ihp-hr}$$

Internal efficiency of turbine:

$$\eta_T = \frac{\frac{3600nw}{17.75}}{(T_2 - T_1)\, c_p \frac{nw}{550}} = 0.42 \qquad (22)$$

Overall turbine efficiency:

$$\eta_e = \eta_m \times \eta_t = 0.42\eta_m \qquad (23)$$

where η_m = mechanical efficiency

Pitch diameter of turbine:

$$\text{pd} = \frac{720V_b}{\pi N} = \frac{720 \times 800}{\pi N} = \frac{183{,}500}{N} \text{ in.} \qquad (24)$$

The above calculations represent a general solution to a typical turbine using lift coefficient, solidity, and angle of attack considerations. The selection of free-vortex versus solid-body rotation blading, the use of shrouding, and the effects of blade cooling (if utilized) are additional factors which must be taken into account in a final design. An additional concept which may lead to improved turbine performance is the use of boundary-layer-control techniques such as slots and flaps. Use of suction slots energized by an independent pressure source or slotted blades energizing the boundary-layer flow from the lower to the upper blade surface should permit operation with larger angles of attack and hence produce larger lift coefficients. Considerable testing along these lines is suggested in order that the required design criteria may be established.

Pump Selection and Performance

The design of a centrifugal-type pump is usually governed by a series of criteria which attempt to establish compatibility between arrangement, performance, and net positive suction head requirements. Perhaps the most generally used criterion for determining pump configuration is that of specific speed. The definition and derivation of this concept is covered quite well [4] so that only the practical considerations of application of this relationship will be discussed in this paper. The specific speed relation is given by the equation

$$N_s = \frac{N\sqrt{Q}}{(H)^{3/4}}$$

where N represents the speed in rpm, H the pump developed head in ft, and Q the flow in either gpm or cfs. Centrifugal pumps suitable for most aircraft and rocket-engine applications would usually operate in the range of specific speeds from 350 to 3500 (on a gpm basis). Higher values lead to mixed flow or axial flow pump configurations. This is best seen by referring to Fig. 16 where pump type (limited to impeller shape), specific speed, and efficiency have been related in a graphical sense. This curve can be used in a general way for establishing the type of pump to design for a particular application, but for performance estimates the efficiencies indicated in the lower specific speed (500 to 1500), lower flow (10 to 500 gpm) ranges is considerably below that attainable with current aircraft quality high-speed pumps. Another factor which directly affects pump configuration and often requires a compromise on attainment of maximum efficiency is that of suction specific speed. In fact, those applications where net positive suction head requirements are extremely limited or nonexistent (as in the case of fuel booster pumps handling boiling liquids), considerable deviation from conventional pump design concepts may be required. The phenomenon of cavitation has been exhaustively described in the literature,[5-9] and while over the years considerable disagreement has remained regarding the fluid mechanics of the inception and resulting damage of cavitation some generally accepted concepts have evolved. In their simplest terms the inception criteria concern the effects of local static pressures, velocities, additional thermodynamic properties of the fluid, heat-transfer properties, compressibility effects, and finally time. Cavitation damage (while essentially in the effect rather the cause category) is basically a materials problem, it generally being ac-

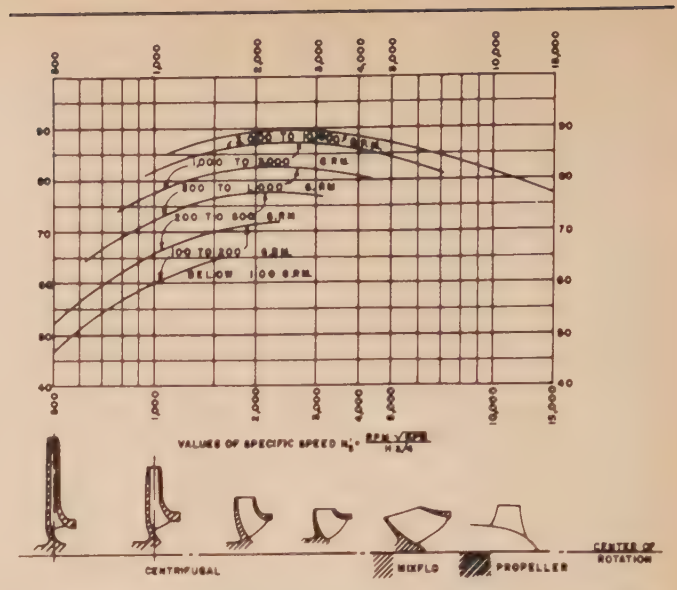


Fig. 16—Approximate relative impeller shapes and efficiencies for variation with specific speeds

[4] "Fluid Mechanics of Turbomachinery," by G. F. Wislicenus. Pub. by McGraw-Hill Book Co., New York, 1947.

[5] *Technology Report*, Tokyo Imperial Institute, Vol. 8, 1929: "Aerofoil Theory of Propeller Turbines and Propeller Pumps with Special Reference to Effects of Blade Interference Upon Life and Cavitation," by F. Numachi.

[6] *VDI Zeitschrift*, Vol. 92, August, 1950, pp. 629–635: "Cavitation Limit in Centrifugal Pumps and Turbines," by C. Pfleiderer.

[7] ASME Paper 48-A-107, 1948 "Dynamics of Cavitation Bubbles," by M. S. Plesset.

[8] ASME Paper 55-A-136, 1955, "Thermodynamic Aspects of Cavitation in Centrifugal Pumps," by H. A. Stahl and A. J. Stepanoff.

[9] ASME Paper 55-A-142, 1955, "Turbulence and Boundary-Layer Effects on Cavitation Inception from Gas Nuclei," by J. W. Daily and V. E. Johnson, Jr.

cepted that fatiguing or rupture of the metal occurs due to the pressure waves produced from the resulting implosions of the gas bubbles formed. The most fundamental cavitation relationship for pumps is that known as the Thoma parameter, which was introduced in this country over 27 years ago by Professor C. F. Moody and is expressed as follows:

$$\sigma = \frac{H_{sv}}{H}$$

where:

H_{sv} = Net positive suction head
H = Pump developed head

This relationship is applicable for pumps of constant specific speed. The other expression for relating the cavitation parameter to pump similarity relations is the previously mentioned suction specific speed. This relationship is:

$$S = \frac{N\sqrt{Q}}{(H_{sv})^{3/4}}$$

While many pump authorities consider this relation valid in maintaining similarity with respect to inlet flow and cavitation conditions, to the extent of utilizing this relationship independent of specific speed, it is doubtful whether such an oversimplification is justified except in the purest theoretical sense. Such an unsophisticated approach can quite often result in misleading if not outright incorrect conclusions regarding cavitation performance in pumps. For example, a simple increase in pump head by increasing impeller outside diameter and leaving the inlet flow conditions and geometry unchanged would lead to the conclusion that cavitation conditions would remain unchanged. In theory this is true. In practice, the subtlety lies in the assumption of constant inlet flow conditions. It is apparent that increased recirculation flow due to the increased head will have considerable effect on the cavitation performance of the pump. Another factor omitted in the suction specific speed relation is the various properties of the fluid being pumped. The effects of viscosity, surface tension, latent heat of vaporization, vapor/liquid volume ratio, thermal conductivity, convective heat-transfer rate, and compressibility are only recently being given consideration in the design approach to cavitation. Finally, the all-important condition of time is critical in evaluation of cavitation performance. With regard to the latter criterion, it is obvious that sufficient time must be available to the fluid particle at the pump inlet for absorption of heat from the liquid, to provide for vaporization of the particle, and to permit vapor bubble growth and subsequent implosion in critical regions of the pump.

Within the considerable limitations described above, it is possible to establish a pump design using primarily specific speed and suction specific speed relations. If a value of S is assumed, based on previous experience, and with pump flow and npsh requirements known, the operating speed of the pump can be established. With the pump speed determined and with the additional information concerning required pump head, the specific speed, and hence the pump impeller configuration can be determined. A typical example of an "open-eye" pump in the 1200 to 1500 specific speed range is shown in Fig. 17. The essential elements of this pump include a centrifugal impeller, vaned radial diffuser, helical scroll, inlet axial labyrinth, rear face-type seal, and axial inducer. An example of a "closed-eye" type pump in the 700 to 800 specific speed range is shown in Fig. 18. As can be seen this unit is very similar in appearance to the higher specific speed unit. The major difference is in the mechanical requirements of an inlet volute and inlet guide vanes necessitated by the closed-eye arrangement. The two types of pumps are generally used in the "overhung turbine" arrangement described previously. An additional consideration in pump selection and design is axial thrust. One method of minimizing this problem is through the use of axial labyrinths. Thrust balancing is accomplished by equalizing pressure forces on both sides of the pump

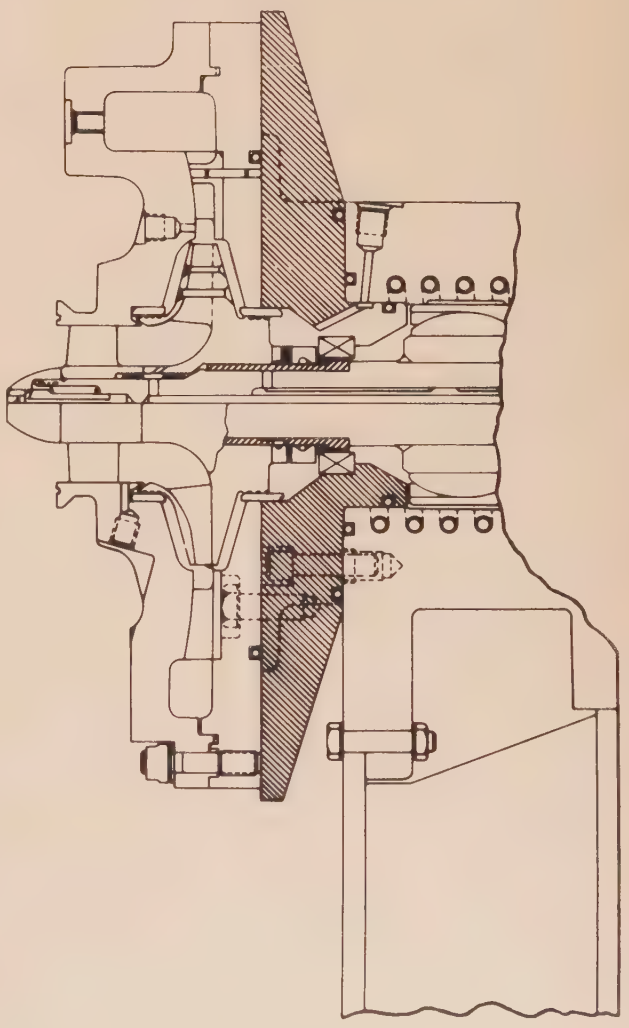


Fig. 17—Typical arrangement of open-eye pump

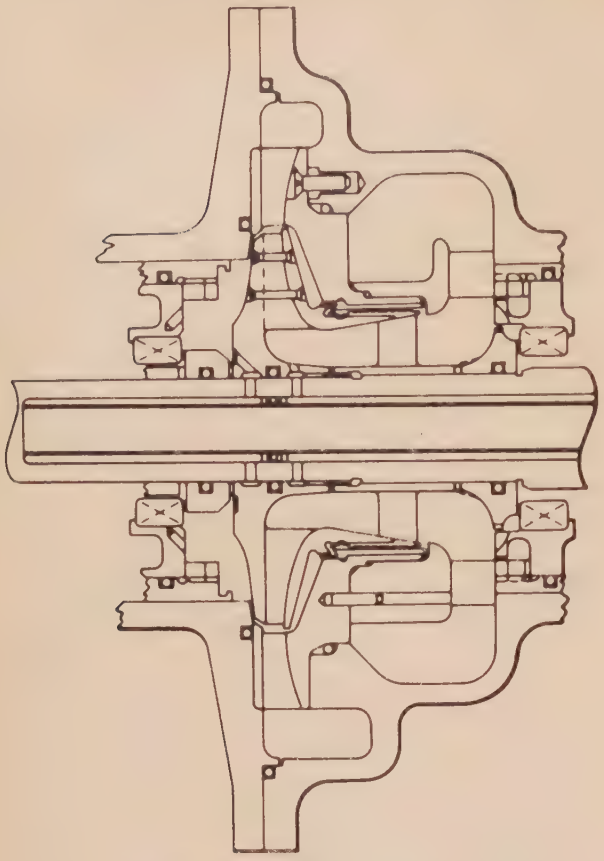


Fig. 18—Typical arrangement of closed-eye pump

impeller. A typical arrangement of such a configuration is shown in Fig. 19. The advantages of such an arrangement are that each pump is individually balanced, and loss of suction of one pump will not result in an unbalanced condition for the turbopump unit.

An important requirement, though not necessarily of prime consideration, is that of pump efficiency. Since the volumetric efficiency or so-called "leakage losses" contribute so greatly to the overall efficiency of a centrifugal pump, the control of the leakage past the front and rear sealing surfaces becomes of major importance. By utilizing such mechanical devices as axial and/or radial labyrinths, face-type seals, and centrifugal slingers an effective sealing arrangement can usually be achieved. An additional requirement in keeping pump efficiencies high is to minimize recirculation flow losses in the impeller vane passages and to prevent so-called "shock losses" at the radial impeller inlet. One method of accomplishing this objective is by the use of airfoil-type vanes and shrouding. The pressure distribution over this type of vane is more uniform, thus minimizing the intensity of internal vortex formation and resulting in increased efficiency. In addition, the rounded leading edge, characteristic of airfoil vanes, is much less sensitive to variations in the relative inlet flow angle of the pumped fluid. A typical example where this precaution was not taken with resulting cavitation damage is shown in Fig. 20. Finally, shrouding helps to reduce the recirculation flow losses by effectively separating the backflow leakage from the main fluid stream. A net performance improvement usually results in spite of the additional friction and boundary-layer growth produced by the shroud. Further steps that can be taken, particularly for high-head-producing single-stage machines is the use of vaned or vaneless diffusers and helical-type discharge scrolls. These latter components can be designed to minimize turbulence and vortex formation in the diffusion passages of the pump. An example of what these design considerations can achieve is best seen in the curves shown in Fig. 21. The lower curve represents typical efficiencies currently obtained in most high-speed pumps for aircraft and rocket engine applications. The upper curve represents the magnitude of improvement attainable with pumps employing the aforementioned design principles. In those cases where overall turbopump efficiency is of major concern, such an improvement can represent a considerable increase in range or reduction in missile take-off weight.

An additional, and perhaps the most challenging requirement involving the design of high-speed centrifugal pumps, is the achievement of high suction specific speeds. The cavitation limits for operation of industrial-type, low-speed centrifugal pumps may vary (according to the Hydraulic Institute) from 5500 to 10,000 depending upon the pump configuration. Such values are usually associated with cavitation-free pump operation. Recent tests on special, highly efficient pumps have extended this range to approximately 12,000 to 14,000. Beyond this point, based upon current technology in the pump field, it is doubtful that truly cavitation-free operation can be obtained. Unfortunately, this state of development is wholly inadequate for aircraft applications where minimum weight requirements impose the necessity for operation at high speeds and minimum net positive suction heads. Most applications for aircraft fuel pumps require operation of pumps at speeds from 15,000 to 40,000 rpm. This is particularly true when a direct turbine drive is involved in order to maintain reasonable turbine efficiencies. This, coupled with

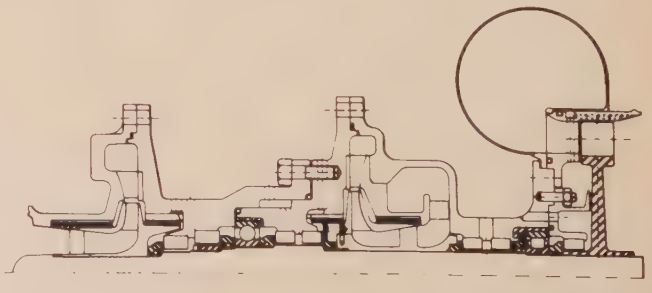


Fig. 19—Thrust balancing through axial labyrinth seals

extremely low suppression pressures, usually results in suction specific speed requirements of from 15,000 to as high as 70,000. In fact for some pump applications, namely fuel booster pumps and liquid gas pumps, where the pumping of a boiling fluid may be required, the usual concept of suction specific speed becomes meaningless since a 0 npsh would result in the requirement of an infinite suction specific speed. Since aircraft and missile type pumps are currently operating in the aforementioned ranges, it is apparent that some new concept of pump operation is required for this field.

At this point, perhaps it is best to review the special conditions under which high-speed aircraft turbopumps operate. The first and perhaps most important item is that of operating life. Rarely do unit life requirements exceed 2000 hr of operation, and in many missile applications this requirement may be reduced to as little as a few minutes. Under such conditions cavitation damage per sec becomes of less interest than the rate at which it occurs, and effects of cavitation on performance become of greater concern. Secondly, hydrocarbon fuels, liquid gases, and certain other chemical fuels and oxidizers have demonstrated a much smaller tendency to evoke cavitation damage than that experienced with such conventional fluids as water.

The reasons for this latter condition are not fully understood, but various authorities have proposed theories [7,8,10] which tend to shed some light on this subject. In general, these reasons resolve themselves down to the variations in fluid properties as discussed previously in this paper and the effects of these in resisting cavitation inception and reducing cavitation damage. Such being the case, it becomes evident that the direction in which the aircraft pump designer would go would involve operation in the partial or fully cavitated region. At this point the only question remaining in this regard is that of the nature of the pump configuration to be employed under these conditions. As is usually the case in most complex technical problems, while there is probably one optimum solution for this requirement, it is not readily evident, and, hence, many compromise solutions are usually suggested. One such solution involves the supercharging of the pump inlet from an external pressure source or from the pump discharge by utilizing an injector principle at the pump inlet. Such an approach, while effective in some cases, usually results in a considerable lowering of pump or overall system efficiency.

A second approach and one commonly used in applications where pumping of a boiling liquid is involved is to provide extremely large pump inlet volumes and to provide for removal of the entrained vapors by judicious venting of the pump configuration. In some cases recompression of the gases formed can be accomplished by utilizing vaneless diffuser inlets and locating the impeller vane inlet edges further out on the periphery of the centrifugal impeller. Prerotation imparted to the fluid in this type of inlet considerably reduces the comparatively high V_a/U_e ratio which would normally result. Another method, more often used with vane or regenerative-turbine-type pumps, than with radial and axial flow pumps is to provide for elimination of the gases by employing the use of centrifugal separator techniques in conjunction with proper venting. Still another principle is the use of an axial flow impeller followed by a vaneless diffuser and possibly a vaned diffuser and discharge volute. Separation of gas and vapor bubbles is again accomplished by proper venting at various locations.

One method which has been employed rather successfully by the author and other pump designers is

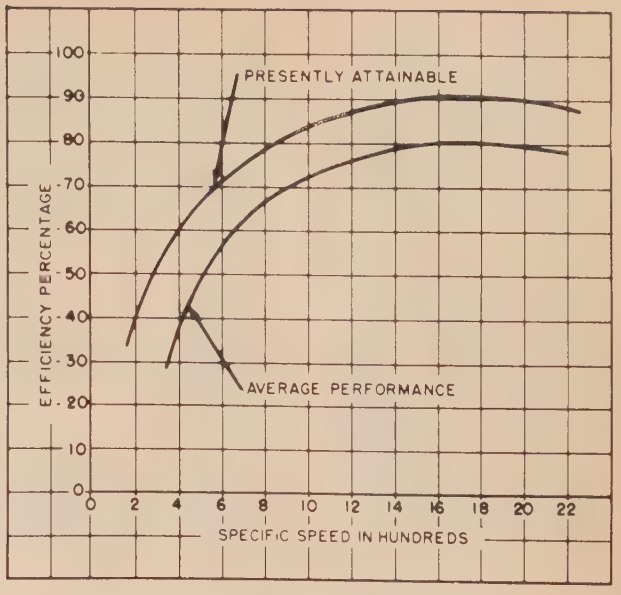


Fig. 21—Efficiency versus specific speed for present pumps and pumps based on aerodynamic design principles

[10] ASME Paper 55-A-144, 1955, "Critical Considerations on Cavitation Limits of Centrifugal and Axial Flow Pumps," by G. F. Wislicenus.

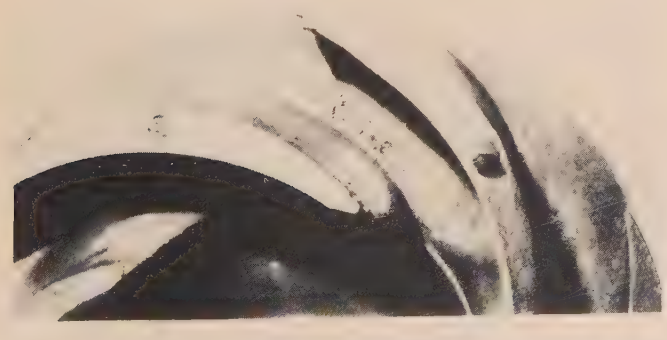


Fig. 20—Effects of cavitation on typical centrifugal impeller

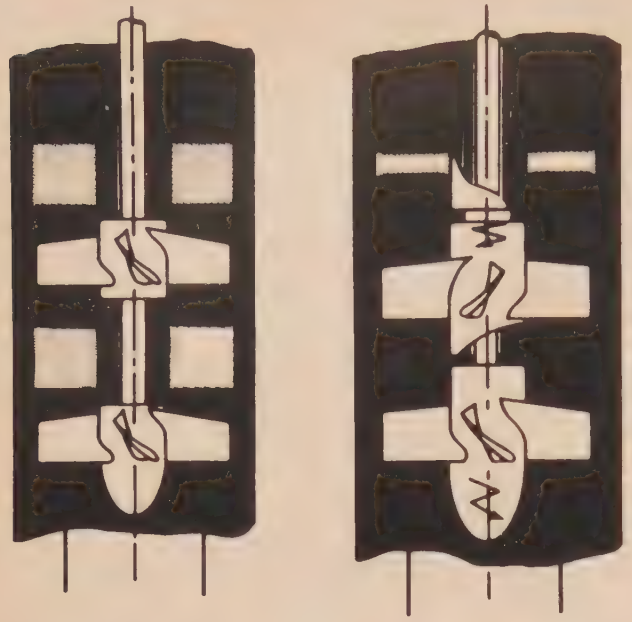


Fig. 22—Various axial impeller configurations

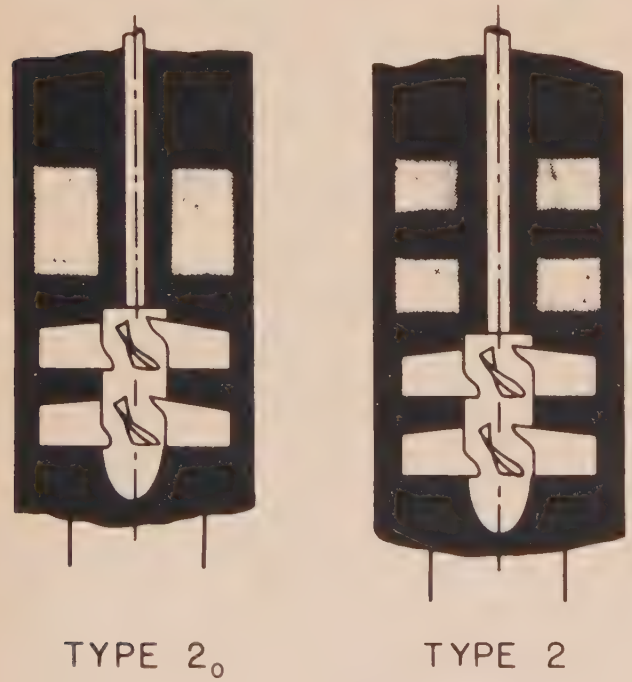


Fig. 23—Various axial impeller configurations

the use of a cavitating inducer at the pump inlet. Before going into detail about this design perhaps it would be well to review various types of inducer configurations that might be utilized. Professor Numachi [11] proposed various arrangements of axial impellers with and without guide vanes. These are

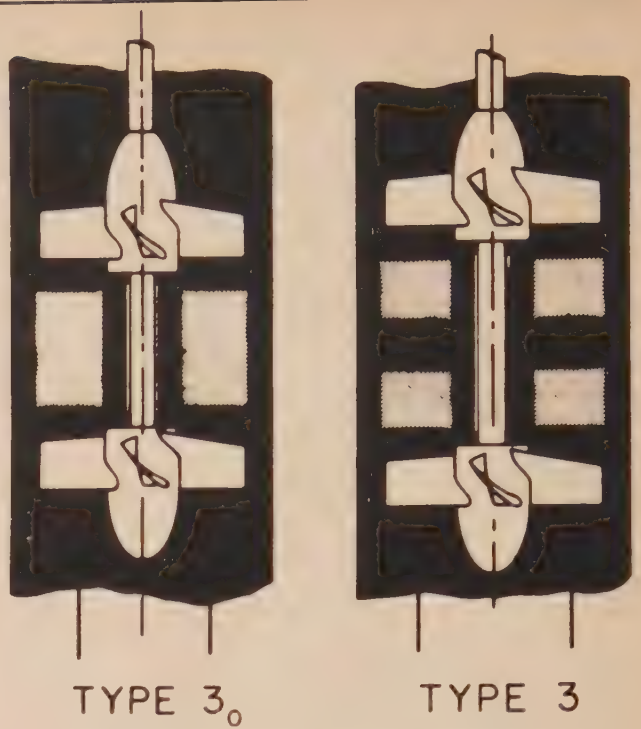


Fig. 24—Various axial impeller configurations

shown in Figs. 22, 23, and 24. While not necessarily intended for use as cavitating inducers, these configurations demonstrated the wide range of possible arrangements. Still another arrangement is shown in Fig. 25. This consists of a shrouded axial inducer with discharge guide vanes at the inlet to a centrifugal impeller. Other promising possibilities for high cavitation performance with axial inducers include the use of staging and boundary-layer control techniques through the use of slots and flaps. The latter type of configurations has already been applied to both pump and axial-flow blower [12] configurations with, in some instances, rather startling results.

At this point one may well ask, "Why the cavitating inducer in place of the more conventional Francis-vane-type centrifugal impeller?" Aside from the complexities of design and manufacture perhaps this question can best be answered by referring to Fig. 26. As can be seen from these curves, a centrifugal impeller when entering the cavitation region experiences a rather sharp drop in developed head and in the fully cavitated region completely loses its head-producing capacity and is quite unstable in operation. On the other hand, the axial impeller produces a rather stable characteristic throughout the entire cavitation regime. The reasons for this phenomenon, though not fully understood are usually attributed to lower blade loading, larger through-flow areas, and reduced fluid containment time. As to which of the aforementioned and axial inducer ar-

[11] *Technology Report*, Tokyo Imperial Institute, Vol. 9, 1930: "On Two-Stage Propeller Pumps," by F. Numachi.
[12] ASME Paper 55-A-156, 1955, "Slotted-Blade Axial Flow Blower," by H. E. Sheets.

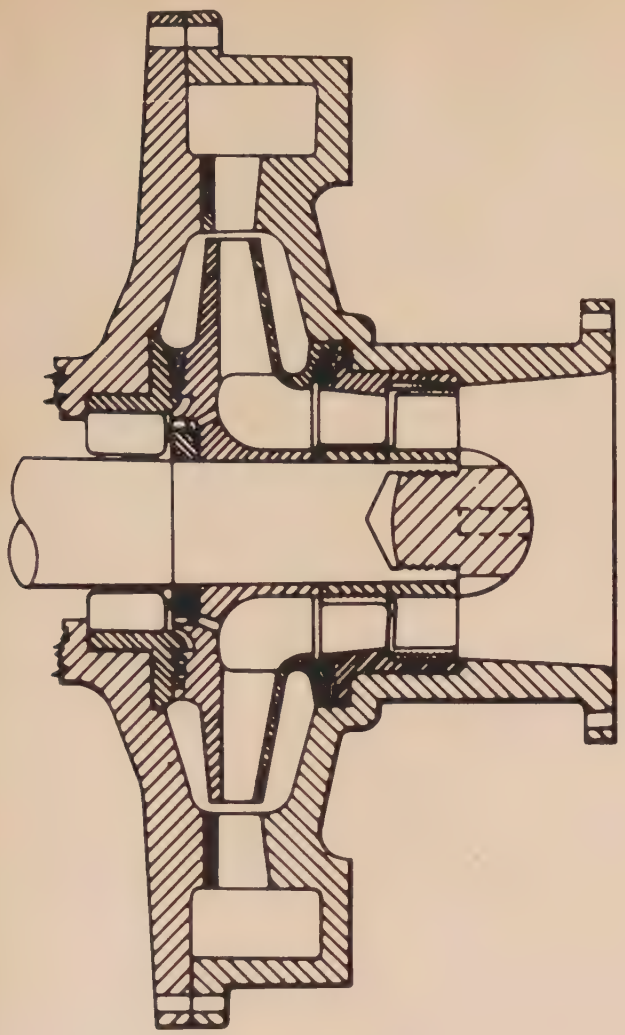
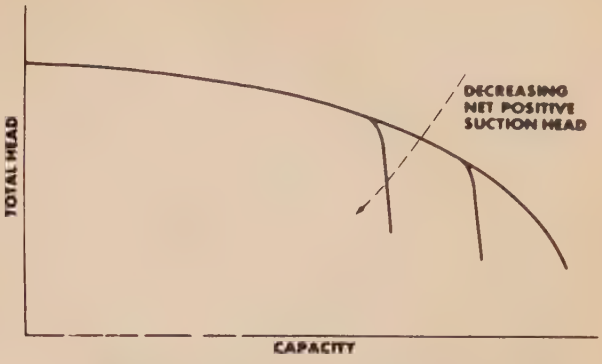
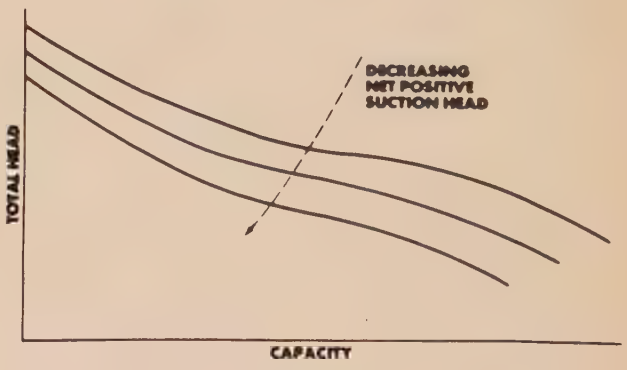


Fig. 25—Pump configuration with shrouded axial inducer and discharge guide vane



PERFORMANCE—TYPICAL CENTRIFUGAL PUMP



PERFORMANCE—TYPICAL AXIAL FLOW PUMP

Fig. 26—Performance characteristics of different type pumps

rangements might produce the best results, it is doubtful if any current theoretician would be bold enough to venture a guess with any hope of providing analytical justification for his choice. For, if to the rather complex 3-dimensional considerations involved in turbomachinery design [13] is added the further complication of operation in the incipient or fully developed cavitation region, little hope or optimism can be held for an early solution. With the aforementioned in mind, the best solution initially appeared to be the selection of the simplest configuration, namely, the use of an axial inducer alone. However, even with the acceptance of this simpler concept many possibilities existed. One type of inducer currently in use is the so-called "screw" type axial impeller as shown in Fig. 27. A design of this type used in conjunction with a radial impeller has been successful in achieving quite high suction specific speeds (20,000–40,000)

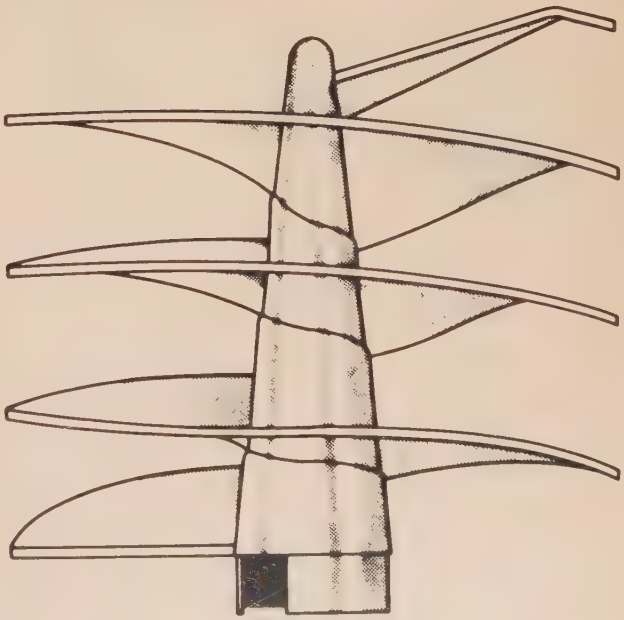


Fig. 27—Typical screw-type axial impeller

13 ASME Paper 50-A-79, 1950, "General Theory of Three-Dimensional Flow with Subsonic and Supersonic Velocity in Turbomachines Having Arbitrary Hub and Casing Shapes—Parts I and II," by C. H. Wu.

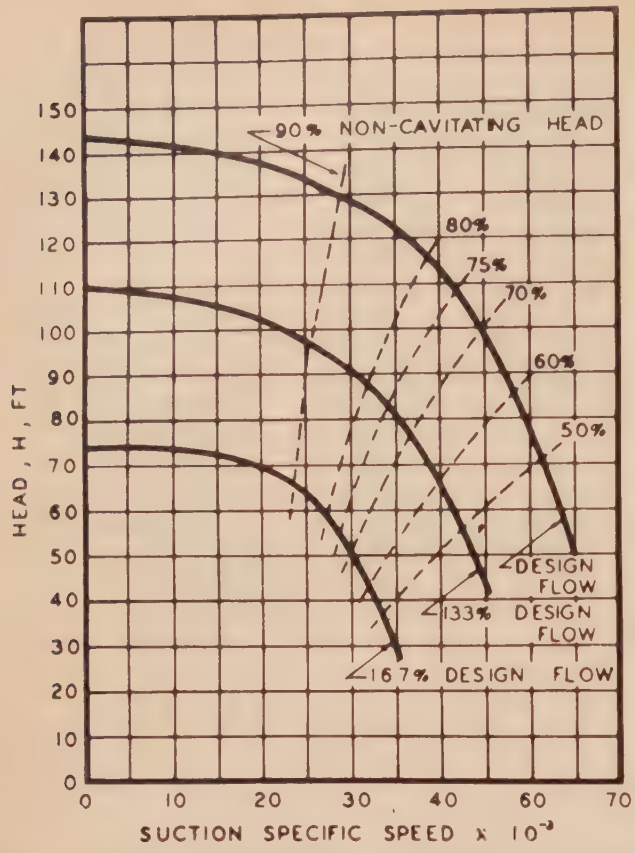


Fig. 28—Performance characteristics of cavitating screw-type pump

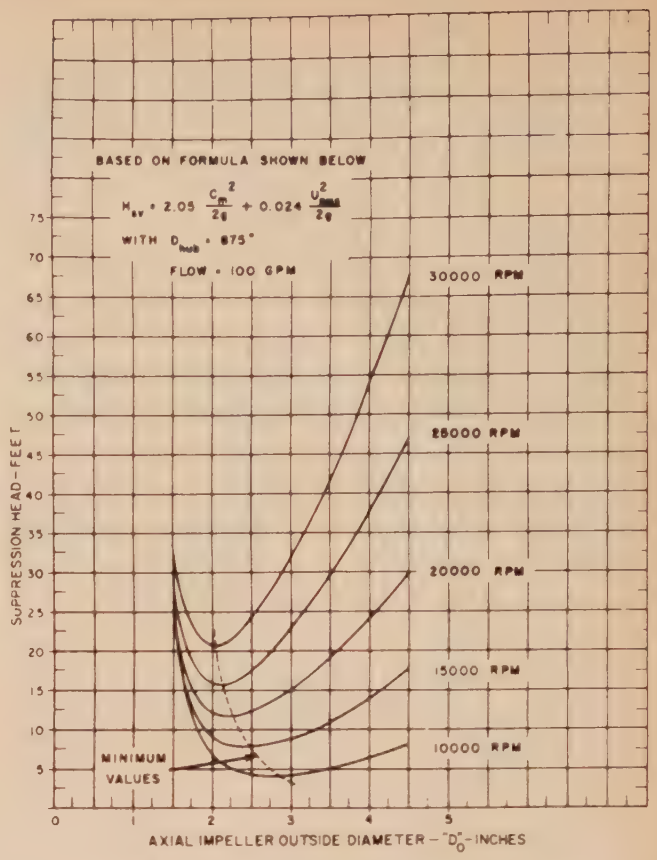


Fig. 30—Suppression head versus axial impeller outside diameter (attainable with high-performance pump)

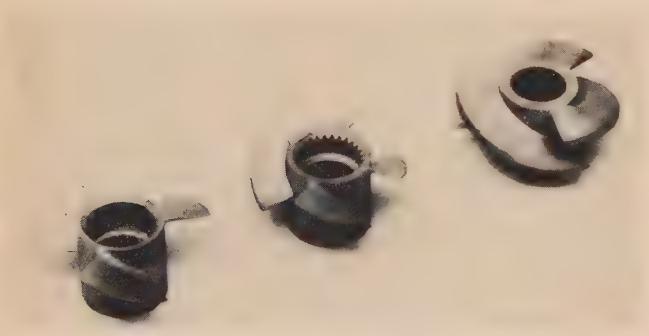


Fig. 29—Typical axial impellers

with pumps operating at comparatively high rotative speeds (15,000 to 30,000 rpm). The performance characteristics of this type of inducer are shown on Fig. 28. As can be seen from this curve, the concept of "per cent of noncavitating head" is introduced. This is an attempt to determine the degree of cavitation occurring in the axial inducer. This type of inducer demonstrates a fairly stable characteristic in the cavitation region. However, this type of unit has a rather steep characteristic of the head-capacity curve in the highly cavitating regions of operation which is quite undesirable, since small increases in flow result in a large drop in inducer-developed head. This may produce a condition where the head developed in the axial stage is insufficient to prevent cavitation from occurring in the main radial impeller. An additional shortcoming of this type of inducer is the extremely long blade length. This long surface produces increased friction and boundary-layer growth thus reducing the inducer and, hence, the overall pump efficiency. Furthermore, the increased time it takes for the fluid to traverse the screw impeller region provides an additional period for bubble growth and convective heat transfer. These latter conditions undoubtedly contribute to the reduced stability characteristics in the cavitation region. Finally, the use of such an extremely high solidity inducer produces a considerable reduction of the through-flow area at the pump inlet. This would lead to increased axial velocities in the unit and produce a further lowering of the available net positive suction head. Another form of axial inducer, and one which the author believes will produce similar (if not greater) high suction specific speeds but with less penalty to pump overall efficiency is the lower solidity, airfoil-vane-type impeller shown in Fig. 29. These axial impellers employing a modified NACA 2409 vane represent designs in the solidity range from 0.5 to 1.5. A pump utilizing an axial impeller of this type used in con-

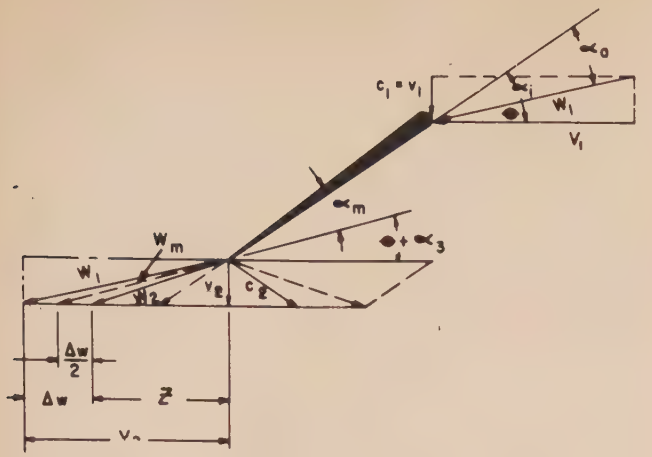


Fig. 31—Effect of axial velocity remaining constant on calculation of axial impeller performance

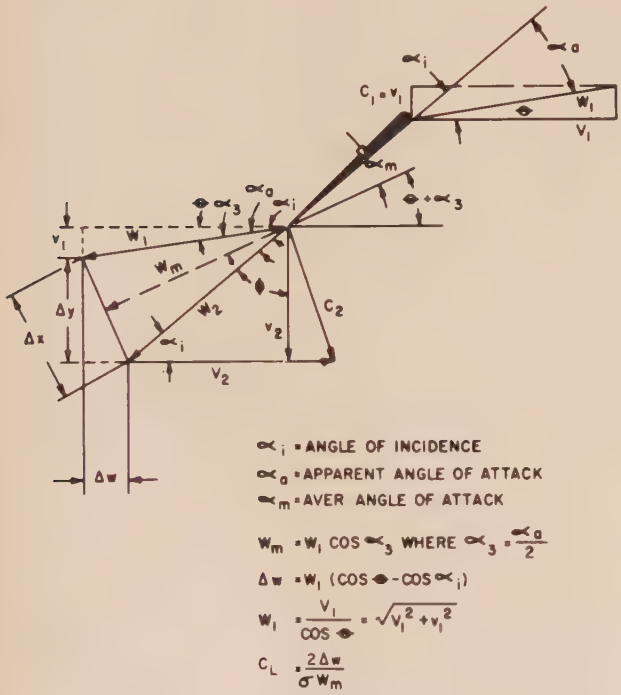


Fig. 32—Effect of assuming relative velocity remains constant ($W_1 = W_2$) on calculation of axial impeller performance

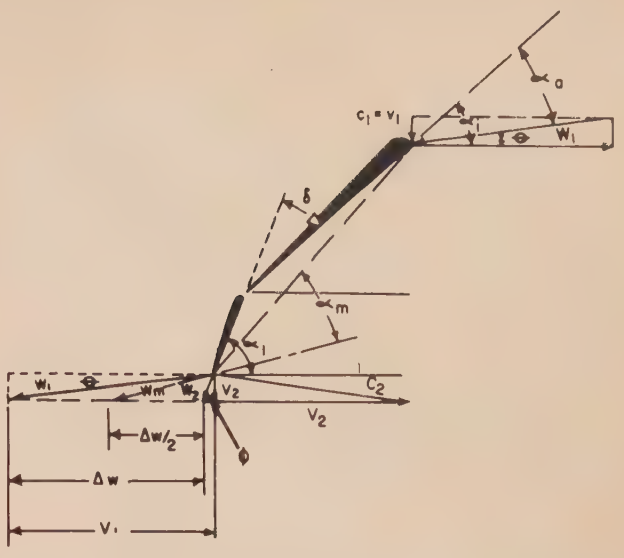


Fig. 33—Improved performance attainable by addition of flap to axial impeller assuming axial velocity remains constant

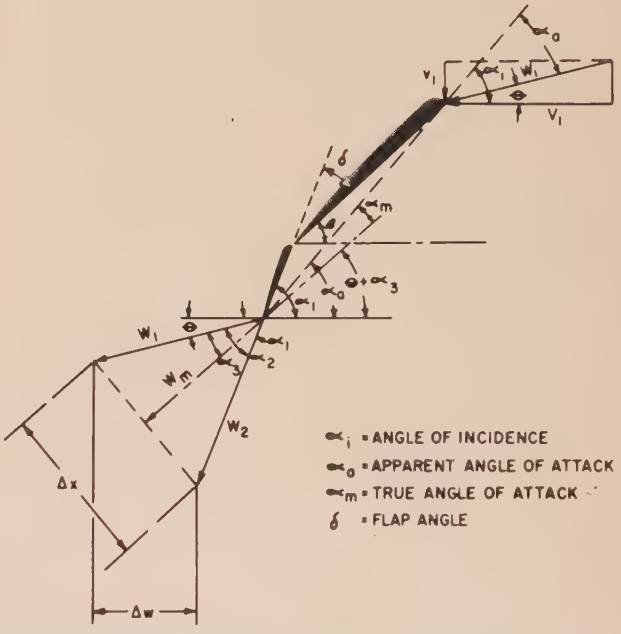


Fig. 34—Improved performance attainable by addition of flap to axial impeller assuming relative velocity remains constant

junction with a high-performance radial impeller was tested, and the coefficient in the simple cavitation equation given below was determined.

$$H_{sv} = \tau_1 \frac{V_a^2}{2g} + \tau_2 \frac{U_e^2}{2g}$$

where:

$\tau_1 = 2.05$

$\tau_2 = 0.383$

The results of this evaluation are plotted in Fig. 30. Suction specific speeds at the minimum points on this curve are in the 24,000 to 32,000 range. An additional modification to the use of axial impellers for pump inlet supercharging is the introduction of boundary-layer-control techniques. This is usually accomplished by employing a high cambered airfoil with a slot or using a lower cambered airfoil with a trailing edge flap. These configurations are generally capable of producing larger pressure rises in a single stage at equivalent or lower net positive suction heads than the low cambered units. However, a much narrower operating range usually results, and extreme care must be taken in the configuration selection and design. For comparison purpose only, vector diagrams for both these configurations are shown in Fig. 31, 32, 33, and 34, respectively. In addition,

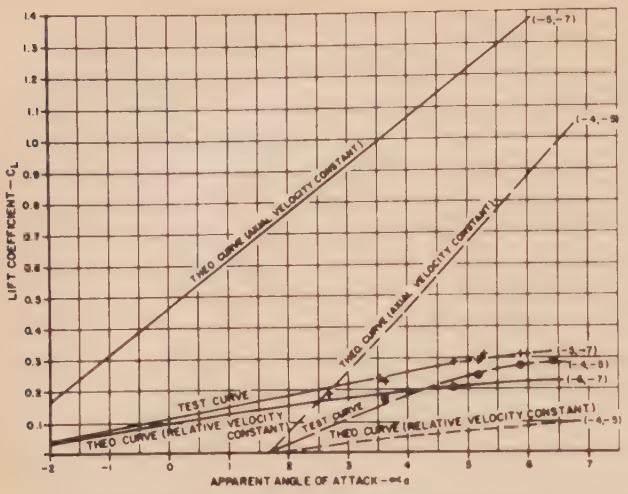


Fig. 35—Lift coefficient versus angle of attack

two varying assumptions have been made: the maintenance of constant axial velocity and the retention of constant relative velocity through the vanes. Of necessity due to the extreme complexity of their nature, the effects of incipient cavitation and boundary-layer control have been omitted. In general, from the vector diagram it can be seen, barring separation, that the flapped design should be capable of producing higher heads. In this vector diagram a high cambered airfoil producing the same degree of turning should be able to produce an equivalent head to the flapped design. However, in this latter case, due to the resulting much higher angles of attack, separation would most likely occur. A further interesting facet that may be noted is that, theoretically, much higher heads can be produced with constant axial flow than can be achieved by producing constant relative flow velocities. Results of tests (Fig. 35) using flapped inducers and flooded suctions indicated that actual heads produced by this type of configuration would lie somewhere in between the upper and lower values of theoretical heads attainable.

An additional consideration in axial impeller selection is that of the so-called dynamic depression. This perhaps can best be understood by referring to Fig. 36. As can be seen, a considerable pressure drop occurs in the suction side of the impeller vane. If this drop is of sufficient magnitude, cavitation may occur. Fortunately, there is a wide selection of airfoil shapes available for axial impeller design, and by proper and judicious choice the pump designer can avoid the pitfalls of excessive dynamic depression. The mutual interference that occurs between vanes is another phenomenon which, if properly applied, can be instrumental in reducing or eliminating the occurrence of incipient cavitation. This condition has the effect of decreasing the suction-side dynamic depression due to the influence of the pressure side effect on the adjacent vane. Thus, the choice of axial inducer vane shape, solidity, angle of attack, and stagger angle become important parameters in the determination of this component's ultimate performance.

Some additional factors affecting pump performance and cavitation characteristics are those of fluid properties and flow conditions immediately upstream of the unit. With regard to the former, when testing with typical jet-engine fuels a rather broad band of possible vapor pressures exists for a given fluid which will still fall within the limits of government specifications. (See Fig. 37.) In testing pumps with fluids of this type, it is important to determine the exact vapor pressure of the fluid being pumped. Read bomb tests are one way of determining this properly. Furthermore, in those cases where pump performance is dependent upon the use of cavitating inducers, boil-off or separation of the more volatile fractions of the hydrocarbon fuel may occur, thus considerably reducing the initial vapor pressure of the fluid. Hence, sufficient sampling and checking must be undertaken during cavitation testing to insure accurate evaluation. Incidentally, this boiling-off of lighter fractions and consequent lowering of the fuel vapor pressure and the lowering of temperature due to evaporative cooling effects is of considerable importance in the successful operation of tank-amount fuel booster pumps. Another interesting phenomenon occurs in the pumping of boiling liquids at or near their critical pressure. Pumps used in the cryogenic field for handling liquid gases such as oxygen, hydrogen, nitrogen, and helium often indicate the ability to operate at considerably higher apparent suction specific speeds (50,000 to 70,000) than would be possible with these same pumps using water. This is attributable to

$$\Delta h_v = \frac{C_h H}{j} \qquad j = a/b$$

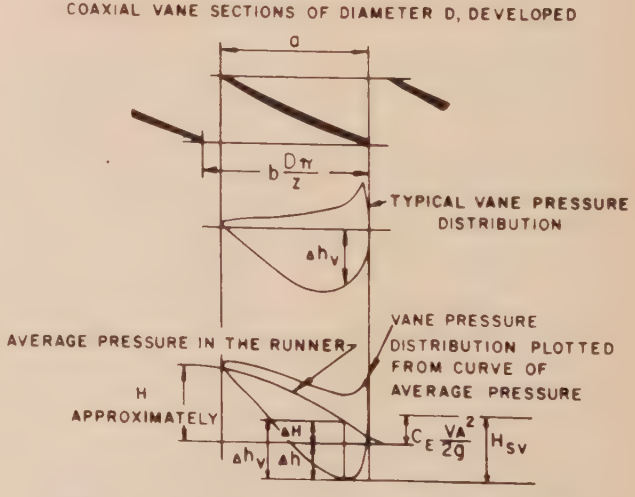


Fig. 36—Dynamic pressure versus chord length

the low specific volume of vapor/liquid ratio at this condition as well as the extremely high ratio of latent heat of vaporization to a available Btu per pound for a given pressure drop in the fluid. An example of this is shown in Fig. 38 where a pump handling water at temperatures from 70 to 300 F and operating with a constant npsh is able to extend its capacity considerably and run at a greatly increased suction specific speed. Also liquid gases, particularly liquid hydrogen, by comparison with most fluids have a fairly high degree of compressibility. Thus, the pressure waves produced by the implosions of cavitation-induced vapor bubbles in a unit pumping this type of fluid are more readily cushioned by the fluid thus reducing their intensity. A marked reduction in cavitation damage usually results, and advance effects on performance are considerably minimized.

An additional fluid property which can have a marked effect on pump performance is viscosity. For moderate values of increased viscosity a small lowering of pump efficiency (Fig. 39) will occur. However, for extremely viscous liquids a considerable reduction in both pump efficiency and capacity will be experienced. Fluids less viscous than water usually result in small efficiency increase, but the results are not as marked in this direction. The small change of friction factor with Reynolds Number in the turbulent region undoubtedly accounts in part for the reduced effects in the latter case. Other items which can have considerable effects on pump performance are entrained vapors (usually resulting from use of pump inlet pressurizing medium during testing) and foreign particles. The effect of entrained vapors results again in reduced performance and a lowering of the suction specific speed and contributes to the production of increased cavitation damage. Foreign particles

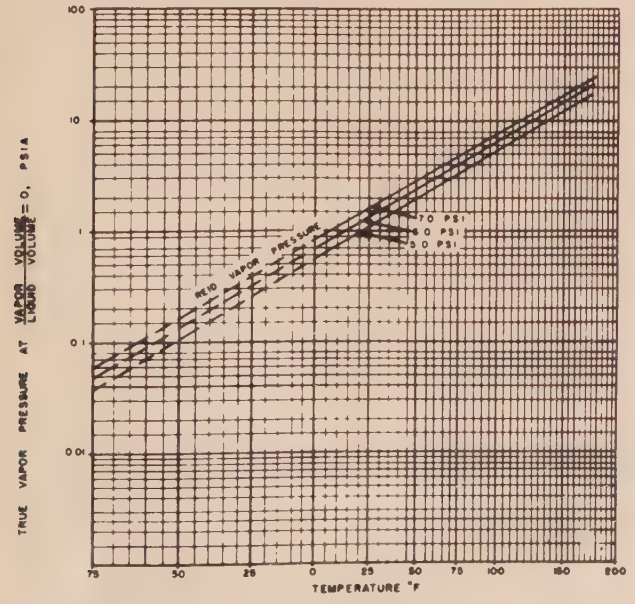


Fig. 37—Vapor pressure of typical jet-engine fuel

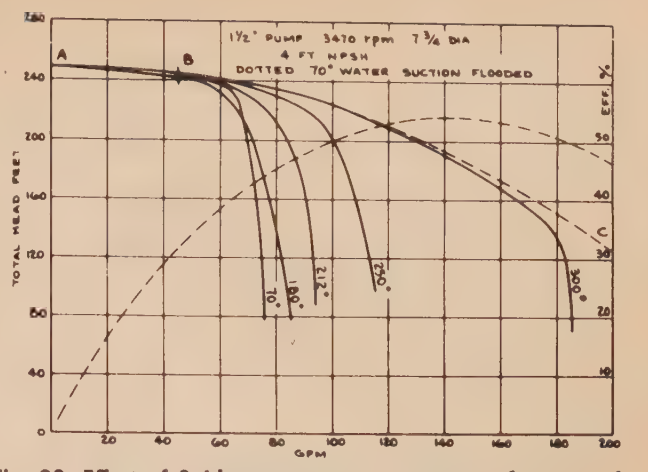


Fig. 38—Effect of fluid temperature on pump performance for constant npsh condition

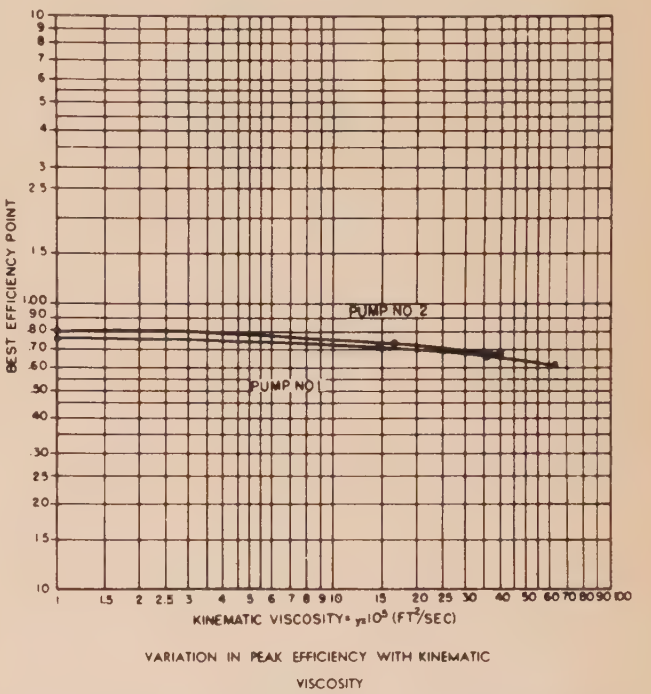


Fig. 39—Variation in peak efficiency with kinematic viscosity

themselves may contain undissolved gas volumes and produce similar effects. However, mechanical damage to seals and labyrinths can be a much more detrimental effect of this fluid contaminant. Since most fuels contain varying amounts of foreign material, good pump selection necessitates acceptance of this condition by proper design of components rather than dependence on filters which may become clogged or produce adverse high pressure drops at the pump inlet.

As mentioned previously, pump inlet flow conditions must also be taken into consideration. Prerotation, turbulence, adverse velocity gradients, and flow-restricting elements all take their toll in affecting pump performance and cavitation characteristics. A

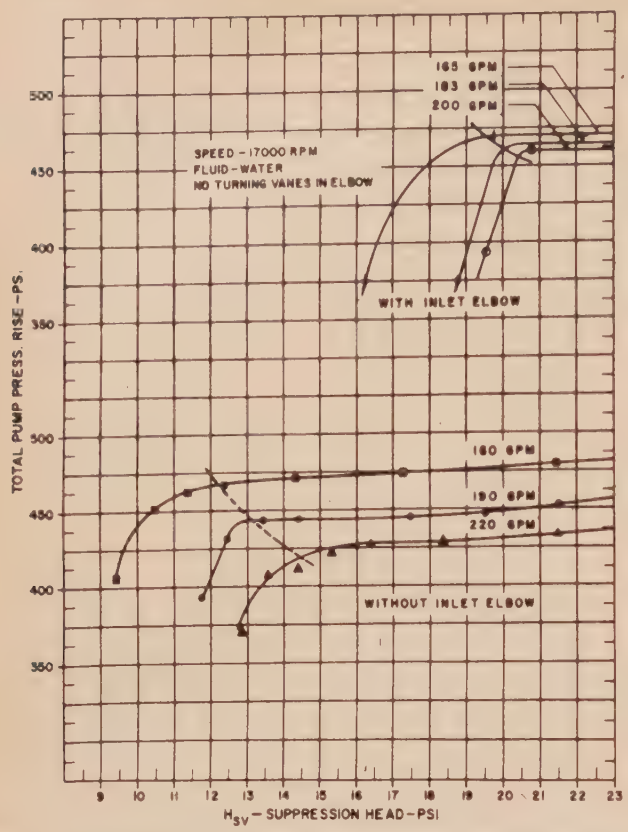


Fig. 40—Comparison of pump suppression heads with and without elbow

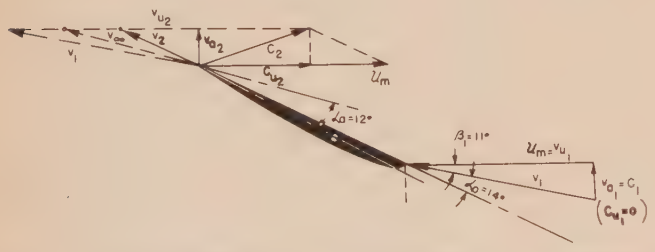


Fig. 41—Vector diagram for oxidizer pump axial impeller

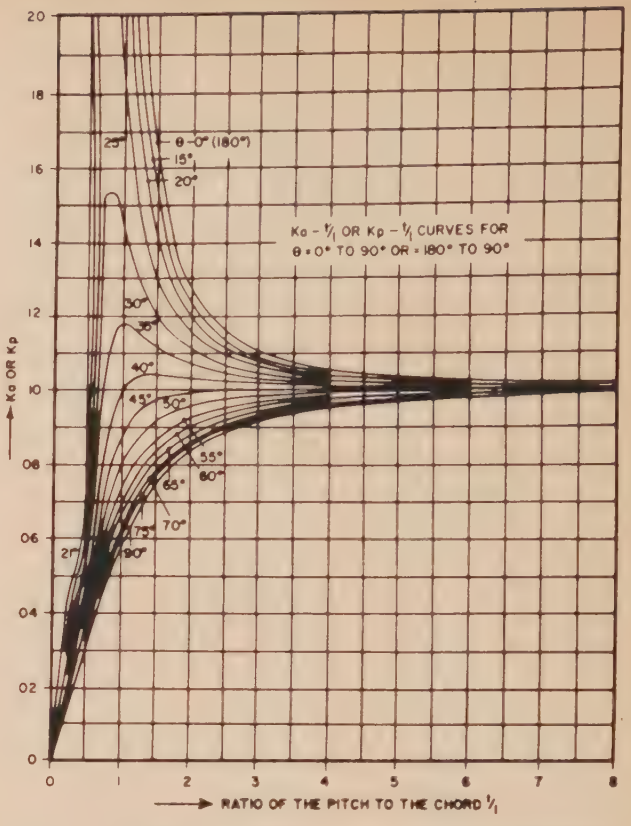


Fig. 42—Lattice coefficient versus inverse of solidity

typical example of this is shown in Fig. 40 where the introduction of a 90-deg elbow at the pump inlet (required for turbopump installation purposes) adversely affected the cavitation performance of a particular pump. For the latter condition, the use of turning vanes in the elbow, of proper shape and solidity, is usually effective in reducing this difficulty.

From the foregoing, it is obvious that many considerations enter into the design of a high performance pump. Only by carefully evaluating all factors can satisfactory results be achieved.

Typical High-Speed Turbopump Unit

Design: By proper utilization of the principles previously outlined, the design of a high-performance, light-weight, turbopump unit becomes possible. A typical example of such a unit utilized as the main pumping plant for a 10,000-lb thrust ATO rocket engine is herein described to indicate the conversion of theory to practice. In general, 1-dimensional flow concepts were used. However, when possible, the application of 2-dimensional aerodynamic principles have been employed in order to predict more accurately the flow performance of the turbomachinery under consideration. By the proper application of previously established design and performance coefficients and the utilization of current aerodynamic concepts, coupled with the classic analysis of fluid mechanics and thermodynamics, the design of a turbopump unit to meet specific operating requirements has been accomplished.

In general, the basic requirements were for a low dry-weight unit with high overall turbopump performance. It was necessary to develop two low-capacity, high-head pumps with efficiencies of approximately 70% and suction specific speeds in the 18,000 to 20,000 range.

The turbine was required to develop approximately 155 hp. The fluids to be pumped were JP3 fuel or aviation gasoline (AN-F-58) and white fuming nitric acid. Available power for the turbine was provided by bleed air from the main jet engines. The turbopump unit was arranged with the turbine overhung, thus necessitating the design of a closed-eye pump as well as an open-eye unit.

The pumps were designed utilizing axial inducers at the pump inlet. The vector diagram for the oxidizer pump axial impeller is shown in Fig. 41. A four-bladed axial impeller with a solidity of 0.75 was

selected on the basis of previous tests. This impeller employed the use of a twisted vane which very nearly approached the "free-vortex" type. A NACA 2409 airfoil shape was selected because of its lift-drag characteristics and its pressure distribution curve. The design approach was to calculate the ideal head produced (assuming no slip and zero prerotation) with the following equation [14]

$$\Delta P = \rho\,(V_{u_1} - V_{u_2})\left(U + \frac{C_{u_1}}{2}\right)$$

This ideal head must be corrected for operation in the incipient cavitation region by a coefficient K_{cav}. This value had to be obtained from previous tests. Because these data were available for an axial impeller configuration that was different for the selected design, a correction to this cavitation coefficient was required. This was accomplished with the use of Figs. 42 and 43. The grid coefficient K_a for each design was determined based upon the variation of solidity and stagger angle and the following correction was made

$$K_{cav_2} = K_{cav_1} \times \frac{K_{a_2}}{K_{a_1}}$$

With the actual pressure rise obtained, the remainder of the axial impeller design was accomplished in the conventional manner. The design of the fuel pump axial impeller was accomplished in a similar fashion.

The radial impellers were designed utilizing test data available on Figs. 44 and 45 and empirically determined design ratios. The radial impeller outside diameters were established by use of the conventional formula [15]

$$D_o = \frac{1840 \times \phi \times \sqrt{H}}{N}$$

14 "Theory and Performance of Axial Flow Fans," by C. Keller and L. S. Marks. Pub. by McGraw-Hill Book Co., New York, 1937.

15 "Centrifugal Pumps and Blowers," by A. Church. Pub. by John Wiley and Sons, New York, 1944.

16 "Die Kreiselpumpen," by C. Pfleiderer. Pub. by Julius Springer, Berlin, 1928.

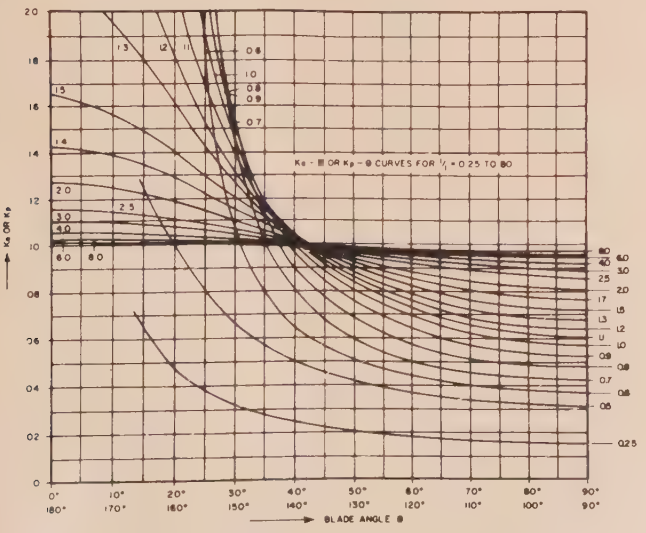


Fig. 43—Lattice coefficient versus reciprocal of solidity

The required suppression pressure was obtained from the equation [16]

$$h_{sv} = \frac{\phi_c \times 4\,\pi\,V_{r1} \times H_{vir}}{U_1 \times Z\left[\left(\frac{r_2}{r_1}\right)^2 - 1\right]}$$

The generalized vector diagram for the oxidizer

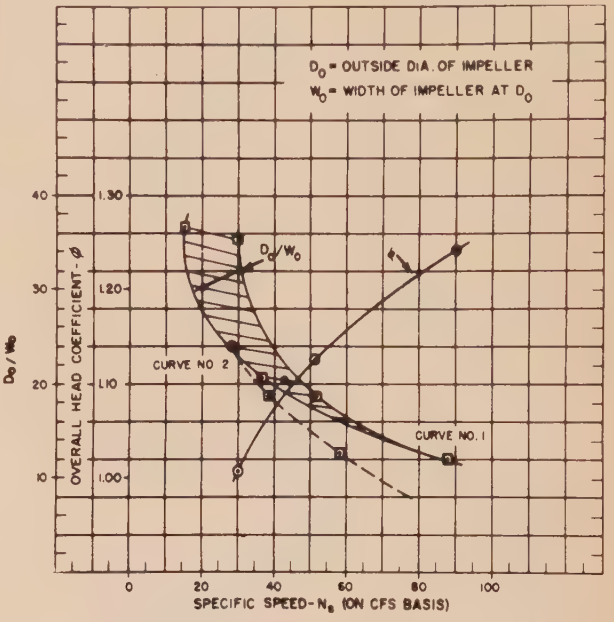


Fig. 44—Specific speed relations for centrifugal pumps A, B, and C

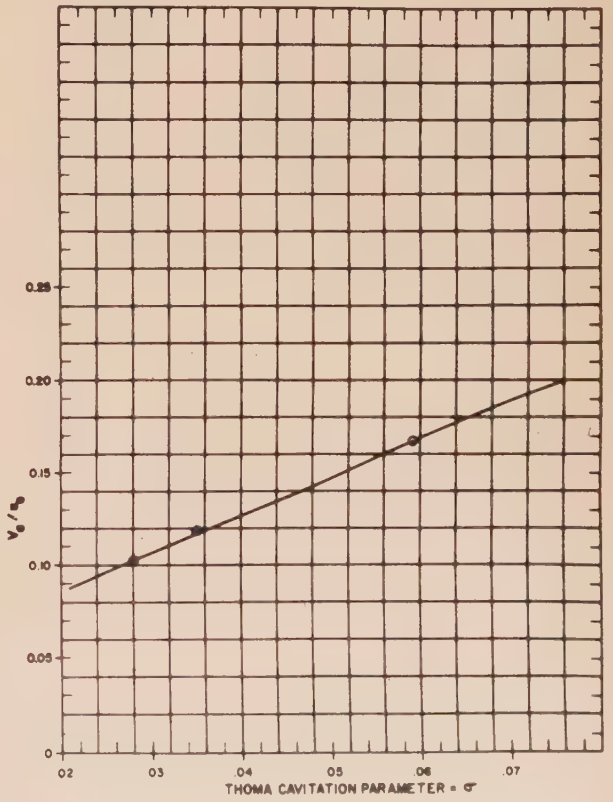


Fig. 45—Suction side characteristics

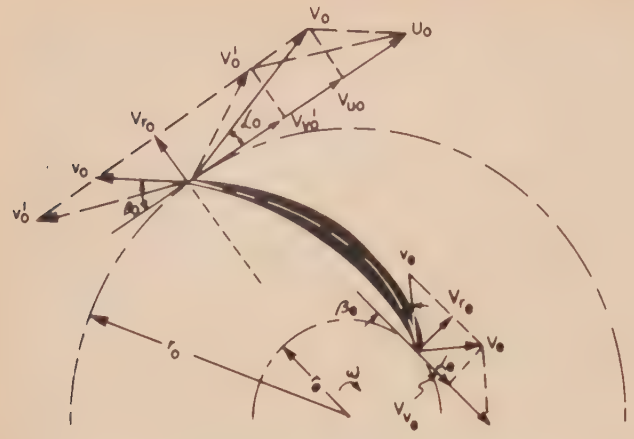


Fig. 46—Generalized vector diagram for pump radial impeller

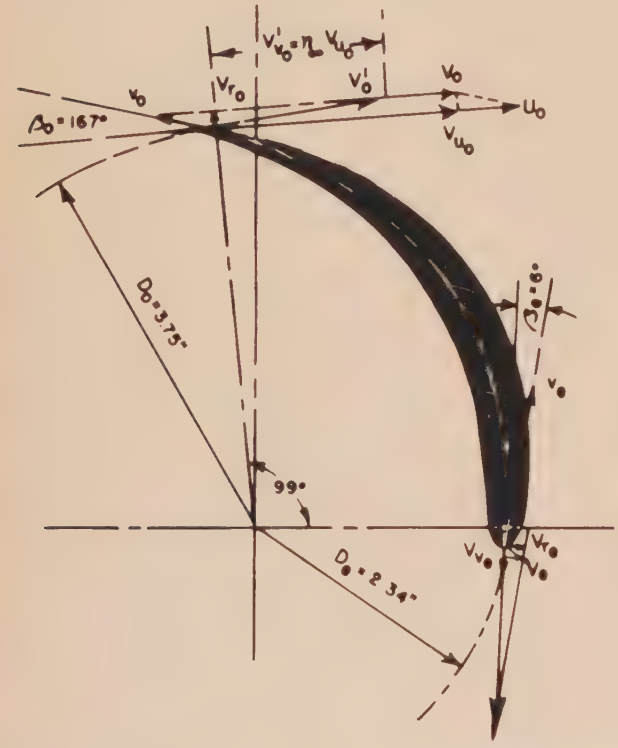


Fig. 47—Vector diagram for oxidizer pump radial impeller

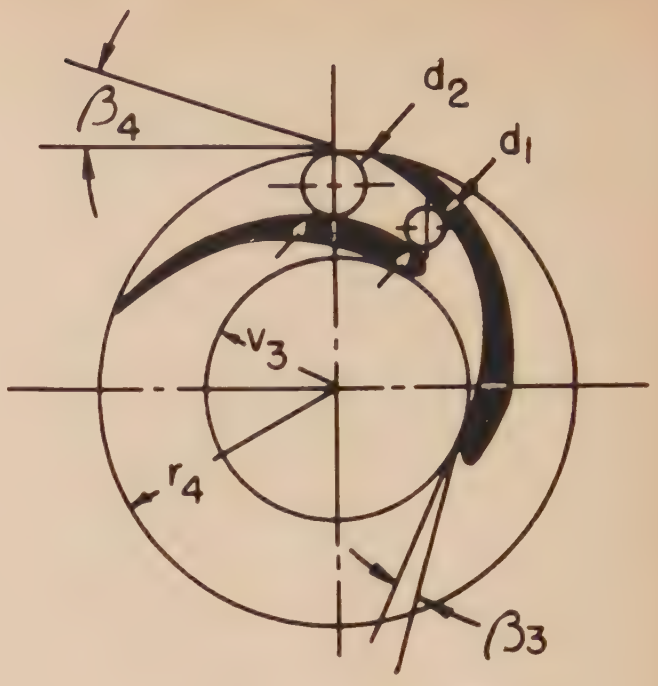


Fig. 48—Typical airfoil vane diffuser

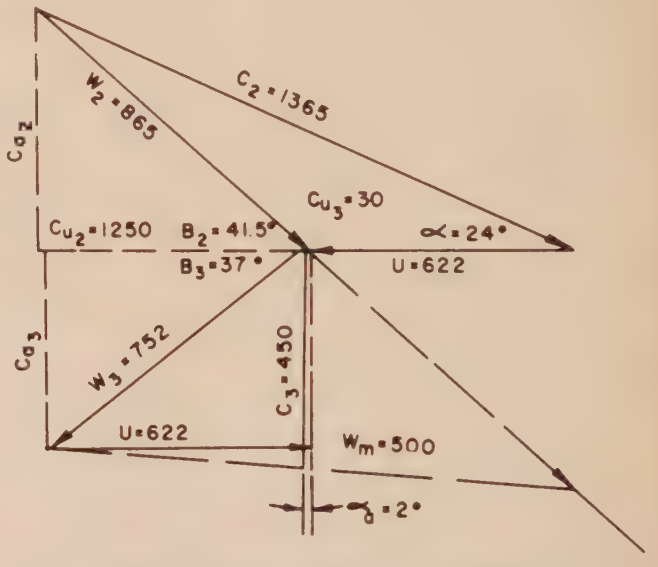


Fig. 49—Air-driven turbine blading vector diagram at pitch diameter of rotor

pump is shown in Fig. 46, and the actual design vector diagram for the oxidizer pump is shown in Fig. 47. Both pumps utilized a vaned diffuser of the type shown in Fig. 48. Both pumps also utilized helical-type scrolls.

Finally, because the axial impellers were designed on the basis of no prerotation at the pump inlet, considerable care had to be taken to maintain this condition. The "open-eye" oxidizer pump was preceded by a 90-deg inlet elbow which was necessary to keep the engine within the required envelope dimensions. A set of airfoil-type guide vanes were required in the bend of the elbow to reduce the pressure drop and to minimize the formation of vortices which would cause adverse disturbance at the axial impeller vane leading edge. The "closed-eye" fuel pump required an inlet volute which normally would give rise to a high degree of prerotation. However, here again a set of inlet guide vanes were provided to achieve as nearly prerotation-free flow as possible. The performance requirements for these pumps are as shown in Table 2.

The turbine selected was a single-stage impulse type operating at a design velocity ratio of 0.456.

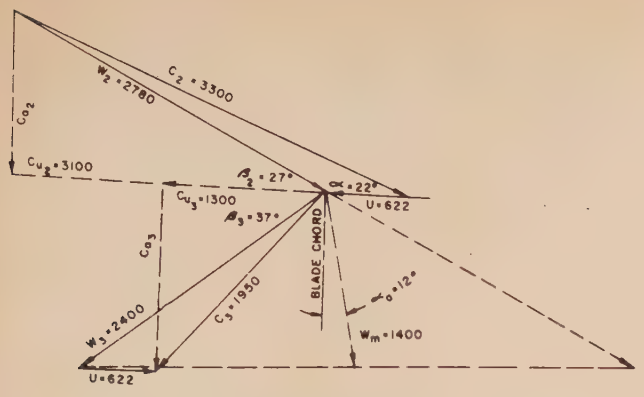


Fig. 50—Gas-driven turbine blading vector diagram at pitch diameter of rotor

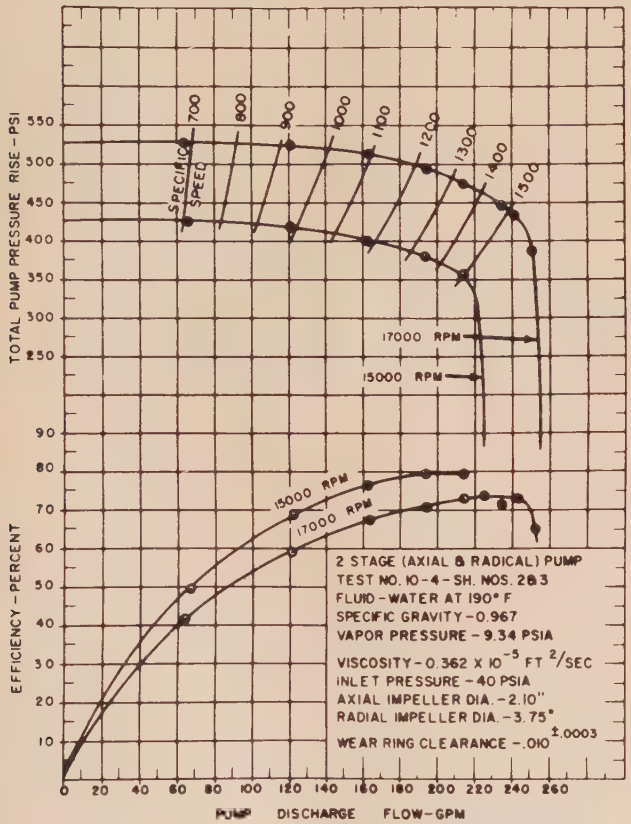


Fig. 51—Performance of oxidizer pump showing head, capacity, and efficiency curves

The additional conditions of Table 3 were established. The blading vector diagram at the pitch diameter of the rotor is shown on Figs. 49 and 50.

Test Results: The performance test results of the "open-eye" oxidizer pump are shown in Figs. 51 and 52. An efficiency of 72% was obtained at the design flow of 190 gpm and finally selected design speed of 17,000 rpm. A suction specific speed of 24,000 was obtained which was quite a safe margin over the 17,500 design requirement. The cavitation results are shown on Fig. 53. The performance test results for

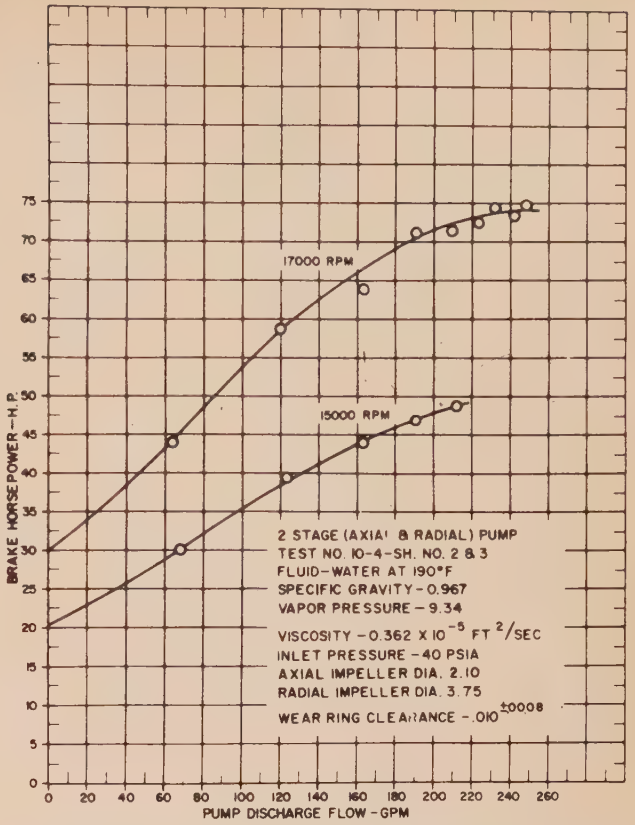


Fig. 52—Performance of oxidizer pump showing brake horsepower and capacity curves for various inlet velocities

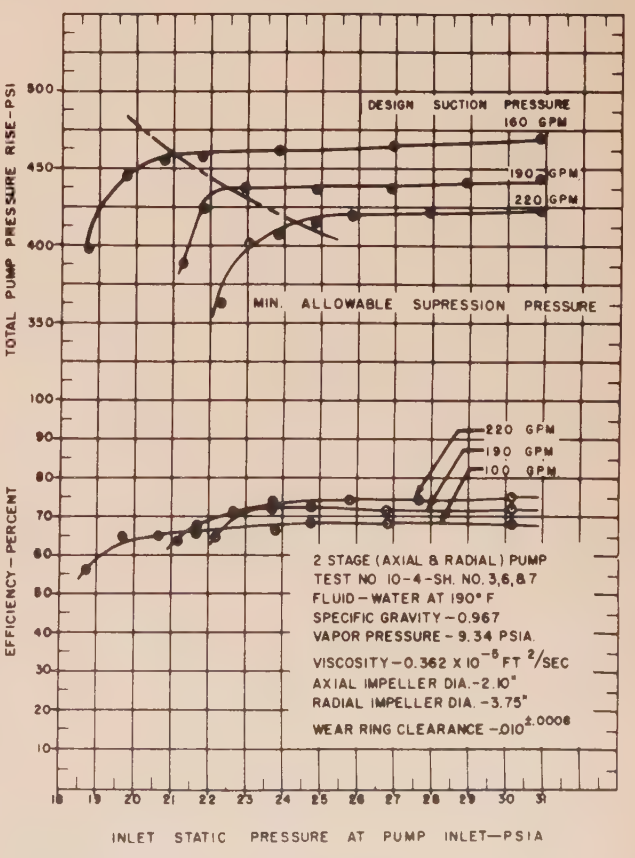
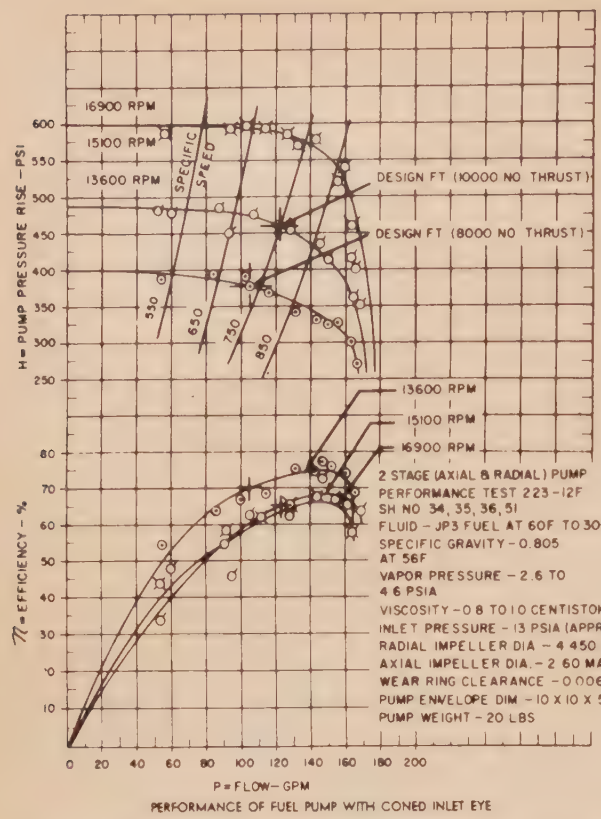


Fig. 53—Cavitation characteristic of oxidizer pump



PERFORMANCE OF FUEL PUMP WITH CONED INLET EYE

Fig. 54—Performance of fuel pump with coned inlet eye

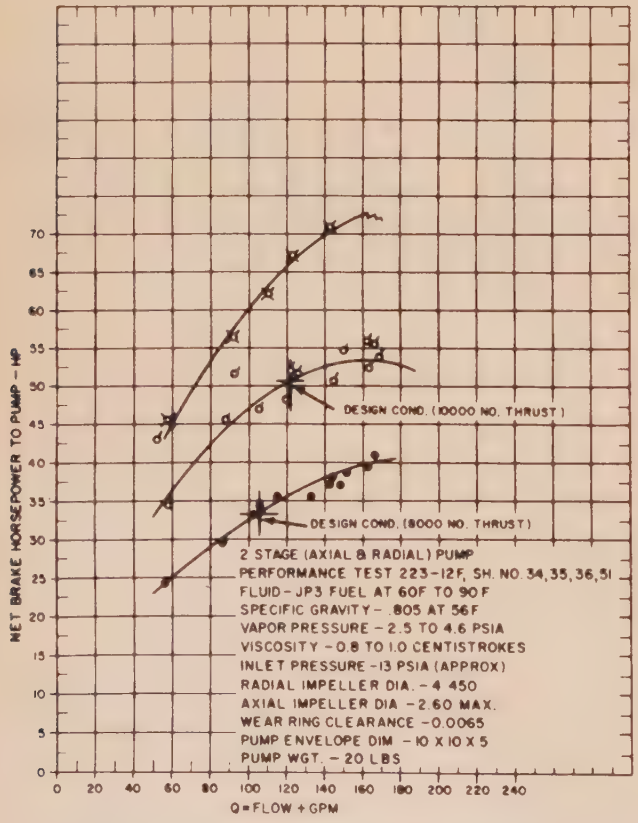


Fig. 55—Performance of fuel pump with coned inlet eye for brake horsepower and capacity curves for given inlet velocities

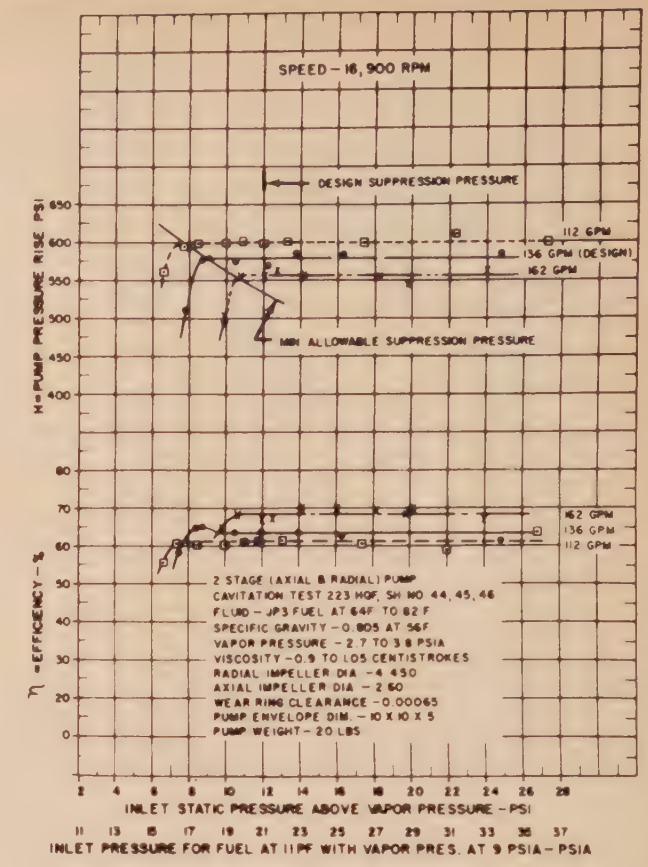


Fig. 56—Cavitation characteristics of fuel pump with coned inlet eye

Table 2—Pump Performance Requirements

	Oxidizer Pump	Full Pump
Flow, gpm	190	122
Pressure rise, psi	650	500
Ns gpm basis	1,420	800
S	17,500	13,000
N, rpm	19,000	19,000

Table 3—Additional Conditions

Nozzle Angle, deg	21
Blade Entrance Angle, deg	37
Blade Exit Angle, deg	37
Nozzle Jet Deflection, deg	3
Velocity Coefficient	0.94
Nozzle Discharge Coefficient	0.99
Solidity	2
Number of Blades	51
Nozzle Inlet Pressure, psia	33.6
Turbine Back Pressure, psia	16.7
Nozzle Inlet Temperature, F	510
Rotor Speed (Design), rpm	19,000
Pitch Diameter, in.	7.5
Nozzle Type	Convergent Parallel
Angle of Attack (Design), deg	12

the "closed-eye" fuel pump are shown on Figs. 54 and 55. At the revised design flow of 136 gpm an efficiency of 67% was obtained. The cavitation tests results for the pump are shown on Fig. 56. The suction specific speed for this pump was 18,000 as against a design requirement of 13,000. The lower value of suction specific speed of the fuel pump compared to the oxidizer pump is attributable to the higher pressure drop and turbulent flow conditions due to the closed-eye configuration. A similar fuel pump designed as an open-eye unit achieved a suc-

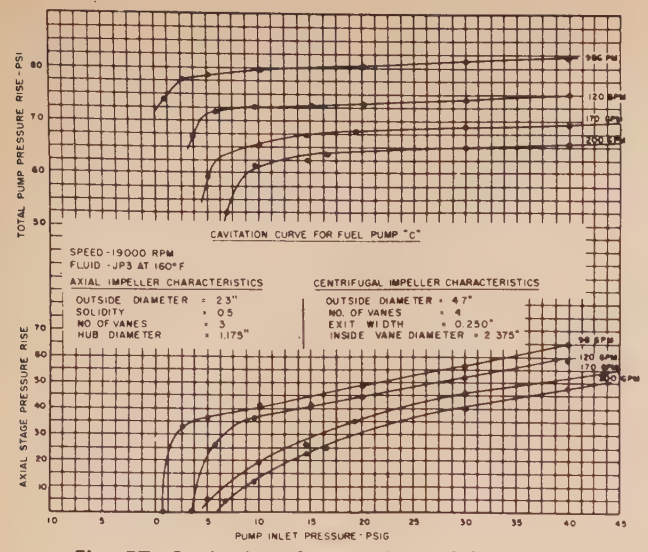
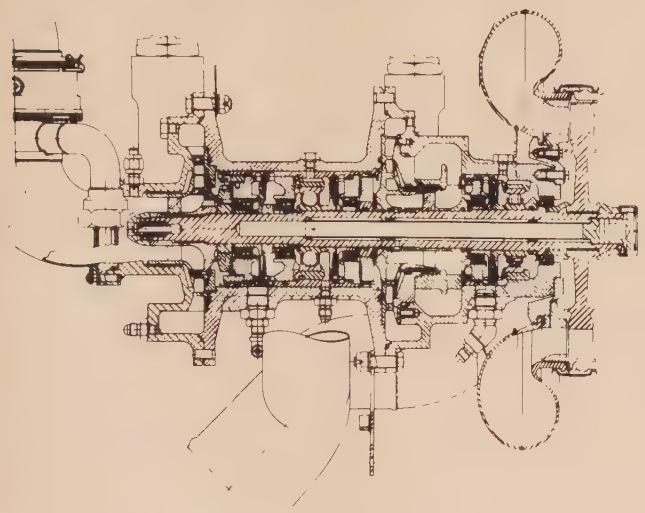


Fig. 57—Cavitation characteristic of fuel pump



TURBO-PUMP UNIT

Fig. 59—Turbopump unit

tion specific speed in the 27,000 to 37,000 range. Results of this test are shown on Fig. 57.

The turbine achieved an internal isentropic efficiency of 61% which was adequate to meet the design requirements. Turbine test results are shown on Fig. 58. The general arrangement of the turbopump unit outlined above is shown in Fig. 59.

Conclusions

The results of the foregoing may be summarized as follows:

1. The take-off weight of rockets and missiles is directly affected by turbopump overall efficiency and required suction specific speed.

2. The arrangement of components of a turbopump unit depends to a great extent upon the application, the fluids being pumped, and the nature of the turbine-driving fluid.

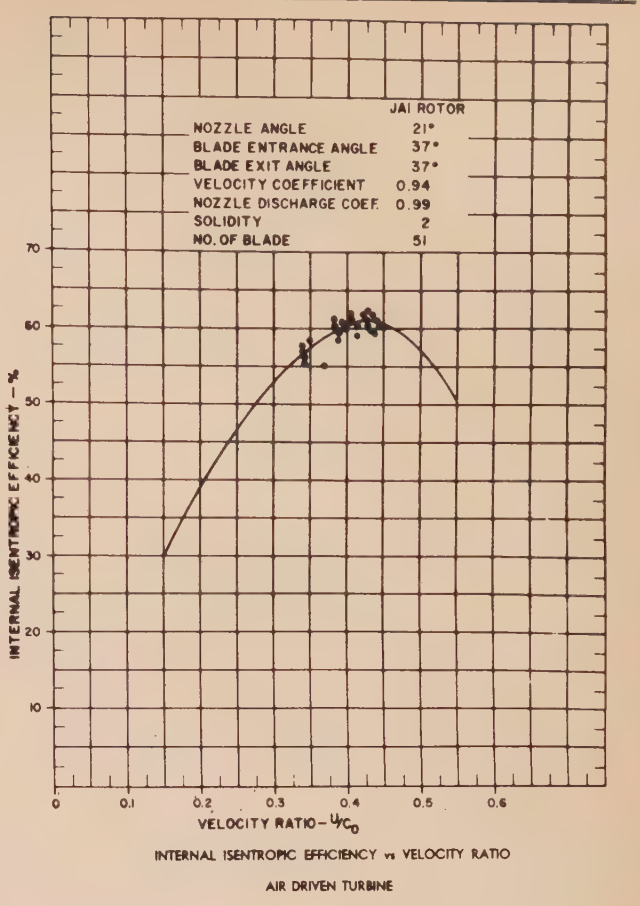


Fig. 58—Internal isentropic efficiency versus velocity ratio for air-driven turbine

3. Turbine selection is governed by weight and efficiency considerations. Velocity ratio, overspeed, and control requirements influence the detailed design of a particular turbine type.

4. The use of the aerodynamic theory and cascade data in turbine design provides a more rational understanding of fluid flow and may permit the extension of turbine operation to include boundary-layer control.

5. Aircraft turbopumps are capable of operation at much higher suction specific speeds due to their shorter operating life and in many cases the nature of the fluid pumped.

6. The use of cavitating axial inducers to provide sufficient pressure rises at the pump inlet to permit overall pump operation at minimum npsh is a currently applied technique in aircraft turbopump design. An extension of this technique to include boundary-layer control through the use of slots and flaps may produce the next step forward in the attainment of even higher suction specific speeds.

Acknowledgment

The author would like to acknowledge the considered opinions and constructive criticism offered by D. O. Moeller, Chief Engineer, and E. Henkel, Public

Relations Manager of the Stratos Division, in the preparation of this paper.

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APPENDIX

A_B = Blading Area, sq in.
A_e = Nozzle Exit Area, sq in.
A_t = Nozzle Throat Area, sq in.
c = Blade Chord, in.
C_L = Life Coefficient
C_v = Nozzle Velocity Coefficient
C_{u_1}, C_{u_2} = Tangential Components of Absolute Velocity, fps
D_o = Outside Diameter, in.
g = Gravity Acceleration Constant, 32.2 fps ^{2}
H = Pump Developed Head, ft
H_{sv} = Suppression Head, ft
H_{vtr} = Pump Virtual Head, ft
h = Blade Height, in.
h_n = Nozzle Height, in.
ihp = Internal Turbine Horsepower, hp
K_a = Lattice Coefficient
K_{cav} = Incipient Cavitation Coefficient
L = Lift, lb
l, l' = Peripheral Length, in.
M = Molecular Weight
N = Speed, rpm
N_s = Specific Speed (gpm or cfs basis)
npsh = Net Positive Suction Head, ft
N_{sv} = Suction Specific Speed (gpm or cfs basis)
n = Number of Nozzles
p = Blade Pitch, in.
pd = Pitch Diameter, in.
P_1 = Nozzle Inlet Pressure, psia
P_2 = Nozzle Exit Pressure, psia
Q = Flow, gpm
q = Dynamic Pressure, psi
R = Gas Constant, ft-lb per lb-R
r_1 = Impeller Inner Radius, in.
r_2 = Impeller Outer Radius, in.
S = Suction Specific Speed (gpm or cfs basis)
sfc = Specific Fuel Consumption, lb per ihp-hr
T_1 = Nozzle Inlet Temperature, R
T_2 = Nozzle Exit Temperature, R
U_m = Tangenital Velocity at Mean Diameter, fps
U_e,u_1 = Peripheral Velocity at Radial Impeller Inner Diameter, fps
V_{a_1}, V_{r_1} = Axial Inlet Velocity, fps
V_b = Turbine Peripheral Velocity at Pitch Diameter, fps
V_2 = Nozzle Exit Velocity, fps
VR = Velocity Ratio
V_{u1}, V_{u2} = Tangential Component of Relative Inlet and Exit Velocity, Respectively, fps
w = Nozzle Flow Rate per Nozzle, lb per sec
W = Nozzle Flow Rate, total lb per sec
W_m = Mean Relative Velocity, fps
Z = Number of Blades
γ = Ratio of Specific Heats (c_p/c_v)
Δ_P = Pressure Rise, psi
η_e = Overall Turbine Efficiency
η_m = Turbine Mechanical Efficiency
η_t = Turbine Internal Isentropic Efficiency
ρ = Fluid Density, lb per cu ft
θ = Stagger Angle, deg
λ_1, λ_2 = Suppression Pressure Constants
σ = Thoma Parameter, H_{s_v}/H
σ = Solidity, c/p
ϕ = Overall Head Coefficient
ϕ_c = Radial Impeller Cavitation Coefficient

Temperature-Strength-Time Relationships in Mufflers and for Truck Muffler Materials

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This paper was presented at the SAE National West Coast Meeting, San Francisco, Aug. 6, 1956.

FAILURE of materials in the high-temperature portion of the engine's breathing system of the exhaust manifold, muffler, and tail pipe has always been a problem. In some parts of the engine such as the exhaust manifold the problem of material failure has been solved by increasing the thickness of cross-section of the material. In other cases, scarce, high-temperature materials or more plentiful but less temperature-resistant materials with ceramic coatings [1] have been used. An increased material cross-section in the muffler does not seem feasible either from a weight or cost standpoint. Hence some other means of providing long life must be found.

The use of higher horsepower engines in over-the-road trucks has resulted in larger mufflers. This increase in muffler size, together with the high brake mean effective pressure at which today's engines operate, has placed even more severe demands temperature-wise on the muffler. It seems imperative, therefore, that either better materials or better design be used to maintain or increase muffler life under today's operating conditions. This paper presents the results of studies conducted by the University of Wisconsin under the sponsorship of the Nelson Muffler Co. in an attempt to determine the temperature and gas-composition conditions under which mufflers operate and the materials which would show promise of long life under these conditions.

In a preliminary investigation samples from a number of failed mufflers and from different locations in the same muffler were examined metallographically in an attempt to determine the cause of failure. These metallographic tests indicated a very definite need for information as to the temperatures encountered in mufflers under actual service conditions. Accordingly two standard mufflers were specially instrumented and, after testing, installed in trucks chosen

* Deceased May 23, 1956.

1 *Journal of Research Bureau of Standards*, Vol. 38, March, 1947, pp. 293–307: "Ceramic Coatings for High-Temperature Protection of Steel," by W. N. Harrison, D. G. Moore, and J. C. Richmond.

DATA presented in this paper show temperature-time diagrams obtained from mufflers mounted on trucks which were traveling over their regular routes.

Using these temperature data, specimens made of possible muffler materials were subjected to laboratory tests. A wide range of possible muffler materials and gas composition were covered in these tests.

Results of the tests indicate that under long-run heavy-duty truck service, muffler failure occurs primarily because of high metal temperatures and that coated mild steel showed the most promise of longer muffler life.

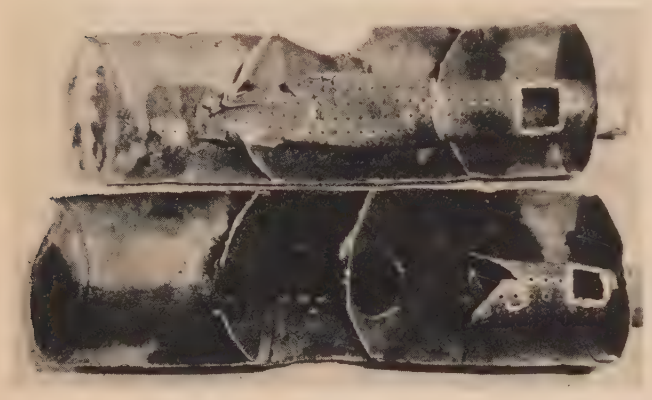

Fig. 1—Typical failed muffler, showing locations from which samples were taken

at random from over-the-road fleets. The temperatures obtained in the mufflers using these trucks were recorded under the actual service conditions encountered by the trucks. After obtaining this information a laboratory test rig was constructed, and specimens were subjected to conditions similar to those encountered in actual engines. After exposure to these high-temperature conditions these specimens were tested for changes in tensile strength, and a portion of them was examined metallographically.

Inasmuch as the tests fall both chronologically and logically into the sequence mentioned above, the paper is divided into three sections: metallographic examination of failed mufflers, temperature data obtained under service conditions, and laboratory tests.

Metallographic Examination of Failed Mufflers

In all, specimens from the most seriously damaged areas in four failed mufflers were examined. The four mufflers were described as follows:

Muffler A used in diesel service, heavily coated with soot inside and badly scaled.

Muffler B used with gasoline, showing what was judged to be "heavy lead deposit" on all inside surfaces.

Muffler C used with gasoline, in which the inside surfaces of the tubes had just begun to blister and were jet black in appearance as if covered with soot.

Muffler D used with gasoline and having a heavy, moderately hard, and quite adherent red coating.

Specimens were cut from five different locations in muffler A, as shown in Fig. 1 and from the most badly damaged areas in the other mufflers. The specimens were cut by hand, using a jeweler's saw to minimize cracking and flaking of the scale. The specimens were mounted in bakelite, polished, and studied metallographically. After metallographic examinations sulfur prints were made of all specimens.

Although for brevity the micrographs are not included here, pronounced changes in the microstructure of the steel were found in the specimens cut from the inlet and outlet regions of muffler A. No completely satisfactory explanation of these changes can be offered although such changes might occur if extensive combustion occurred in the muffler. Steel from the failed ends of the inner tubes showed only thinning due to oxidation and some ferrite grain growth. The sulfur prints showed that the scale contained more sulfur than was originally present in the steel, and a very pronounced congregation of sulfur was noted at the scale-metal interface.

The specimens taken from muffler B indicated quite a different method of attack leading to failure. The surface in muffler B seemed to have been attacked exclusively by intergranular penetration. The sulfur prints did not show much sulfur in the so-called "lead deposit," but again the sulfur seemed to congregate unevenly at the scale-metal interface.

As would have been expected from the appearance of the muffler, the specimen cut from muffler C showed less indication of attack than any of the others. Although some surface roughening had occurred no evidence of intergranular attack was found. The specimens from both mufflers B and C showed a much larger number of inclusions than would be expected of a low-carbon steel. The origin of these inclusions is not known. Again the sulfur was found to be congregated unevenly at the surface of the metal.

The red layer previously noted on muffler D was found to exist not only at the outside surface of the scale but in a layer between the steel and the scale and also in some of the fissures and cracks in the scale. Tests showed the muffler material to be some grade of stainless iron. The method of attack that caused failure was not clear from examination of the micrographs, but it was clear that the failure was quite rapid. Again sulfur was found to be congregated at the surface of the material.

As a result of these studies it was concluded that failure occurred when the wall thickness of the metal

was reduced until it could no longer carry the imposed load and that both temperature and gas composition influenced the extent of the corrosion. It was also decided that information as to metal temperatures actually attained in mufflers was essential to any studies of materials. Since a reasonably thorough search of the literature did not reveal any such data over an adequate range of conditions it was decided to obtain such data on trucks operated over their normal runs.

Muffler Temperatures under Service Conditions

In order to determine the temperatures encountered in mufflers under actual service conditions two commercial mufflers were instrumented with thermocouples at the locations shown in Figs. 2 and 3. Fig. 2 shows a return-flow-type muffler while Fig. 3 shows a straight-through-type muffler.

In order to record automatically the voltage output of the thermocouples a 16-point recording potentiometer was used. This recording potentiometer printed once every 2 sec and thus recorded the output of any one thermocouple once every 32 sec. A small, aircooled, 110 v, a-c motor generator set provided the neecssary electrical power for the instrument.

Since gas composition was considered to be of interest an attempt was made to obtain exhaust-gas analyses. Two techniques were tried: one, the equivalent of a hand Orsat and the other, a commercial exhaust gas analyzer. Although the commercial unit performed quite well during trial runs the severe jolting experienced during actual service conditions prevented its proper operation. Thus, although some data were obtained from the hand Orsat apparatus, the data obtained were not extensive enough to enable any definite conclusions to be drawn regarding the variation in exhaust gas composition under actual service conditions.

After preliminary road testing the specially instrumented return-flow mufflers and test equipment were taken to Portland, Oregon. There, they and the test equipment were installed in a diesel truck whose regular muffler had been removed, and a large number of temperature recordings were made while the truck was covering its regular route. A typical temepraturetime record is shown in Fig. 4 while a temperature-time record obtained on a chassis dynamometer is shown in Fig. 5. It is of considerable interest to note

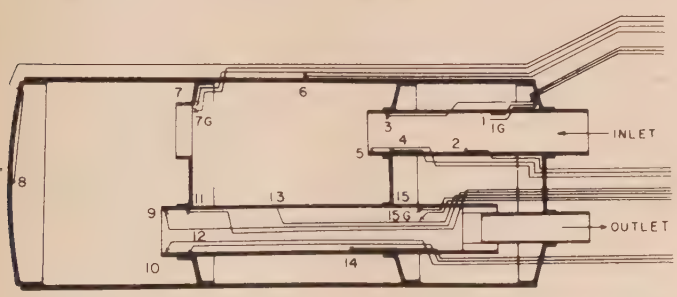

Fig. 2—Schematic diagram of return-flow muffler, showing location of thermocouples

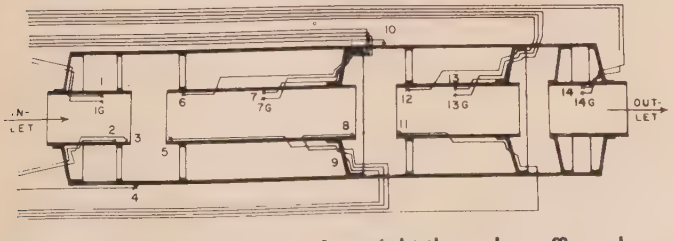

Fig. 3—Schematic diagram of straight-through muffler, showing location of thermocouples

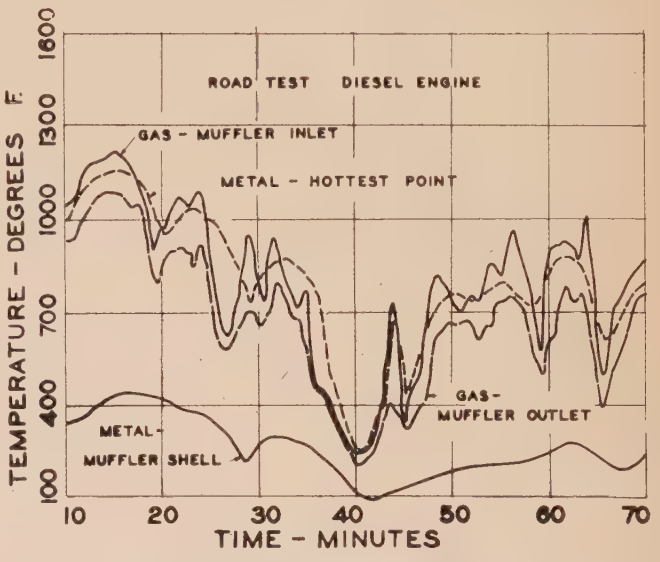

Fig. 4—Temperature-time record for muffler on truck diesel engine in over-the-road service

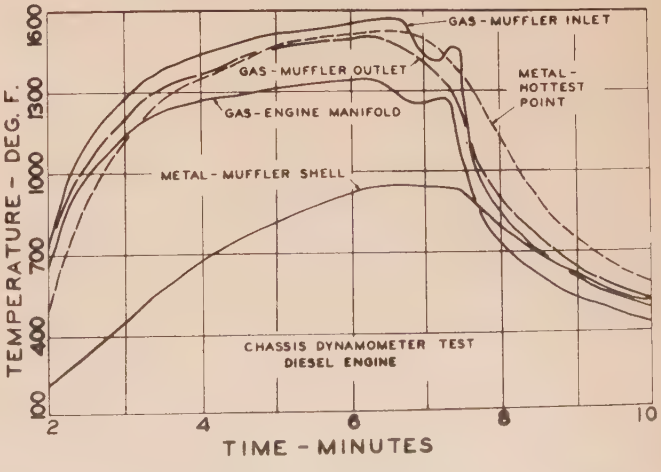

Fig. 5—Temperature-time record of muffler on truck diesel engine on chassis dynamometer

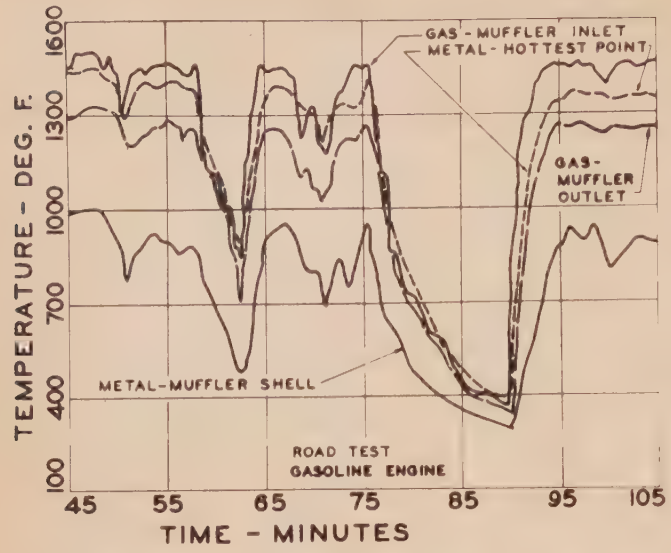

Fig. 6—Temperature-time record for muffler on truck spark-ignition engine in over-the-road service

Table 1—Test Data

	Materials Tested	Temperature Range	Variable
Series I	a, b, c, d, e, f, g, h, i	Constant except for inevitable temperature drop along pipe. Pipe insulated. Range approximately 1500 1600 F	Temperature essentially constant. Effect of location in pipe studied
Series II	a, b, c, d, e, f, g, h, i	1000–1650 F	Effect of temperature and time studied
Series III	a, b, d, f, g, h, i, j, k, l, m	1000–1650 F	Effect of temperature-time and of lead and sulfur in gases studied. Conditions similar to Series II except for atmosphere and change in combustion chamber
Series IV	a, b, d, f, g, h, i, k, l, m, n	1000–1650 F	Effect of temperature-time, richer mixtures, and sulfur in a richer mixture studied. Conditions similar to Series III
Series V	a, b, d, f, g, h, i, k, l, m, n	1000 1650 F	No cycling: temperatures maintained at constant value. To check effect of cycling

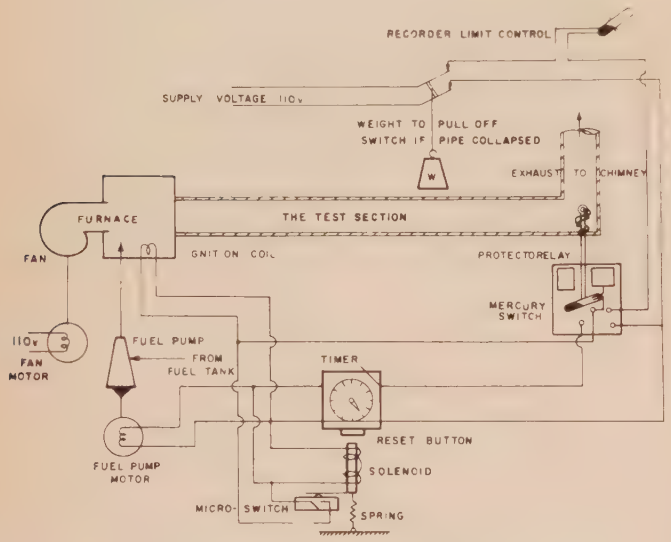

Fig. 7—Arrangement of apparatus for laboratory tests of specimens

the quite wide and fairly rapid swings in temperature as the load on the engine varied.

Following these tests both the straight-through and the return-flow type muffler were taken to Denver. There they were installed on both gasoline and diesel trucks, and again a large number of temperature readings were taken while the trucks were covering their regular routes. The temperature-time data obtained with the diesel trucks were similar to those shown in Fig. 4 while Fig. 6 shows typical data obtained on a gasoline truck.

It is impossible to present all of the data taken during these runs but the following generalizations made after inspection of the data may be of interest. These generalizations are of necessity limited to the conditions under which data were obtained.

1. As would be expected the muffler was continuously heating and cooling.

2. The maximum temperature achieved in the muffler seemed to depend only on throttle opening and to be comparatively independent of vehicle speed and altitude.

3. The gas temperature in the muffler was usually nearly as high as the gas temperature in the intake manifold and sometimes higher. It should be noted that unshielded thermocouples were used for all gas-temperature measurements.

4. From the information presented in 2 and 3 it was deduced that appreciable combustion often took place in the muffler, especially under high-engine-load conditions. The difference in measured gas temperatures could be caused by the use of unshielded thermocouples, but the lack of relationship between vehicle speed (and thus external cooling) and muffler temperature indicates that some heat release is taking place in the muffler.

5. The highest gas temperature noted in service was 1220 F for the diesel engine and 1430 F for the gasoline engine. The highest metal temperature achieved inside the muffler was 1180 F when the muffler was used on the diesel engine and 1360 F when the muffler was used on a gasoline engine.

6. The highest metal temperature found in the return-flow type muffler was near the end of the perforated inlet tube while the highest metal temperature found in the straight-flow type muffler was at the front end of the perforated inner tube.

Although, as was previously mentioned, only a small amount of data concerning exhaust gas composition was obtained it was the general opinion after

the above tests that for the conditions under which data were taken, high metal temperatures were by far the most important factor in muffler corrosion.

Laboratory Tests of Muffler Materials

After having obtained information as to the temperature range encountered in the mufflers a laboratory test rig was set up for test purposes. Samples of possible muffler materials were placed in this test rig and exposed to high-temperature gases for varying periods of time. In this manner information was obtained on the expected service life of possible muffler materials.

Although there were some small variations in the test equipment between different runs the schematic diagram shown in Fig. 7 illustrates the general arrangement used. As is shown in Fig. 7 the oil pump from a home oil burner was driven by a separate motor thus permitting the blower to run continuously. The hot combustion gases or the comparatively cold room air (depending upon whether or not the oil pump was operated) passed through a combustion chamber and into an 8-in. diameter pipe about 6 ft long. After passing over the specimens located in this pipe and a stack thermostat the gases were exhausted to the atmosphere. The previously mentioned variations in test equipment were primarily changes in the combustion chamber used.

As is also illustrated in Fig. 7 the operation of the oil pump motor was controlled by a timer whose operation was in turn controlled by the stack thermostat. The sequence of operations was as follows: When the stack thermostat had cooled to about 400 F it actuated the timer which in turn actuated the oil pump motor. Using the normal safety controls the burner operated for a length of time determined by the timer setting. When the timer turned off the oil pump motor, air from the continually operated blower cooled the test rig. When the stack thermostat had again cooled to about 400 F the cycles was repeated.

Considerable thought was given to the heating and cooling rates to be used. Because of the high external cooling and comparatively low thermal inertia of the engine and exhaust system it seemed impossible to achieve cooling rates obtained in actual mufflers. Accordingly, an arbitrary heating time of 40 min was chosen. The cooling time varied with furnace design and operating conditions but was in the neighborhood of 20 min. The heating and cooling curves for a particular furnace arrangement are shown in Fig. 8. For comparison purposes a heating-cooling curve from the muffler road tests is also shown in Fig. 8.

A drawing of specimens used is shown in Fig. 9. In order to keep a continual check on the temperature of the specimens thermocouples were welded to dummy specimens. The output of these thermocouples was recorded on a multipoint self-balancing potentiometer. In this connection it should be noted that temperature gradients were found across the pipe. In order to minimize these gradients turbulent flow was caused by locating fire bricks in the pipe, and dummy specimens were placed on the outside of the test specimens for radiation shielding. However, even after taking these precautions temperature variations of 30–50 F were found.

In all, test data were obtained for five different conditions. These five conditions, the temperature range covered, and the materials tested are shown in Table 1. The abbreviations and the symbols used for the materials are shown in Table 2.

The ultimate strength data obtained in the Series I test at a temperature of 1600 F are presented in Fig.

Table 2—Nomenclature

Symbol	Description	Abbreviation
a	Mild steel	Mild
b	Mild steel coated with base ceramic material	Mild base
c	Mild steel coated with A19 ceramic material	Mild A19
d	Mild steel with rolled aluminum coating	Aluminum
e	Mild steel painted with aluminum paint	Al pt
f	Stainless steel type 302	302
g	Stainless steel type 405	405
h	Stainless steel type 405 coated with A19 ceramic material	405—A19
i	Corten	
j	Two pieces of mild steel with rolled aluminum coating butt welded together	Al weld
k	Mild steel painted with silicone-aluminum paint	Sil pt
l	Enamel iron coated with base ceramic material	Enamel base
m	Enamel iron coated with A19 ceramic material	Enamel A19
n	Mild steel with chromium plating	Cr

After being subjected to the indicated temperature and atmosphere the specimens were photographed and tested for their ultimate strength. Elongation data were taken for most of the specimens, and a few of them were studied metallographically.

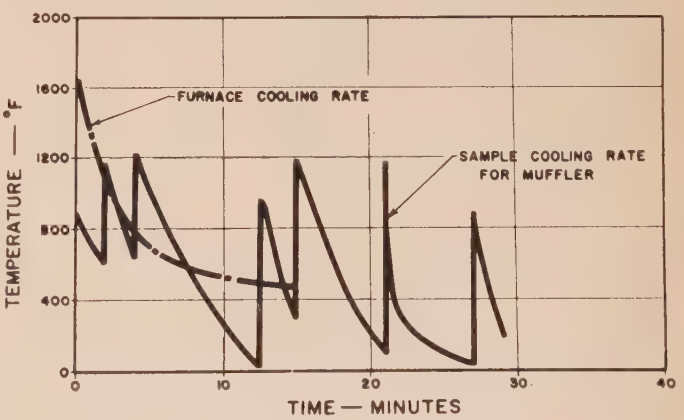

Fig. 8—Comparison of heating and cooling curves for muffler and for laboratory specimens

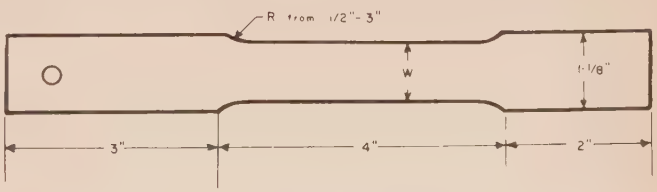

Fig. 9—Example of specimen used

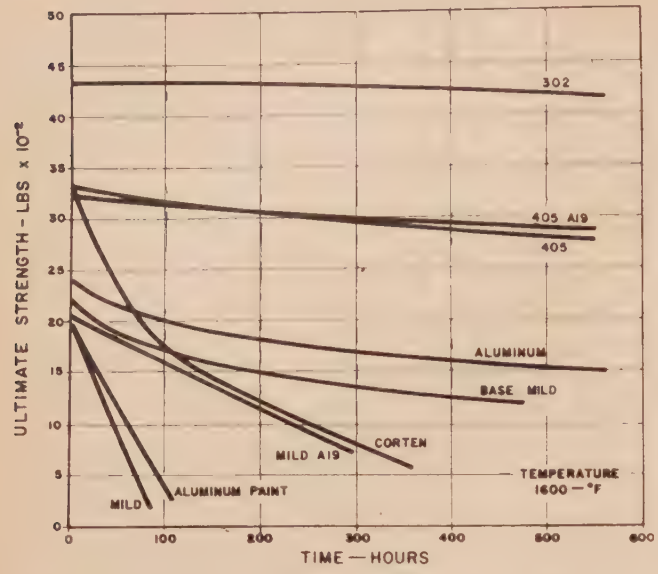

Fig. 10—Variation in ultimate strength under Series I test conditions

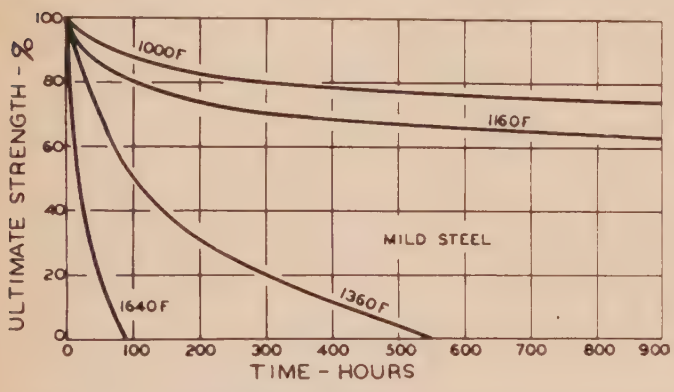

Fig. 11—Per cent ultimate strength versus time for mild steel at constant temperatures for Series II tests

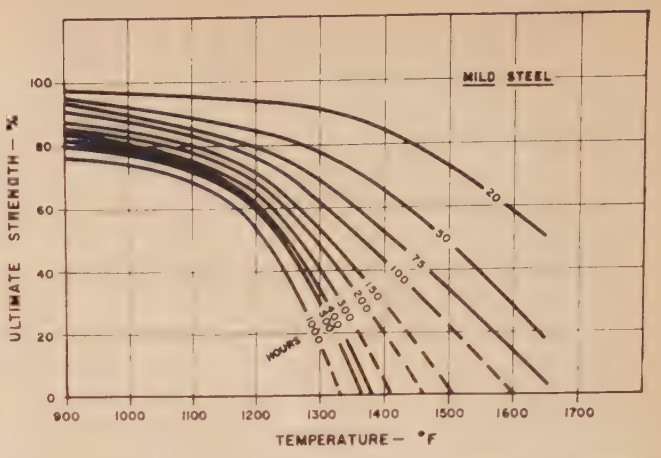

Fig. 12—Per cent ultimate strength as a function of temperature for constant times

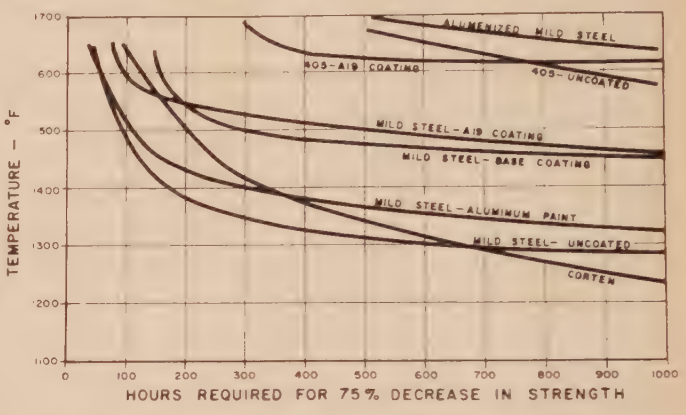

Fig. 13—Temperature versus time for 75% decrease in strength of different materials

10 where ultimate strength is plotted against time. These curves show both the variation in the initial ultimate strength and the ultimate strength after being subjected for a time to the test conditions.

The data obtained in the Series II tests can be presented in several different ways. For example, Fig. 11 presents the data obtained for mild steel as a function of per cent of ultimate strength and time with lines of constant temperature. Fig. 12 presents the same data but cross-plotted to give per cent ultimate strength as a function of temperature with lines of constant time.

When the variable of the material tested is included, the number of variables is thus four, and it is difficult to show all the relationships on a single graph. In order to decrease the number of variables the following reasoning was used. The point of interest is: how long will a particular material last at different temperatures? Let us arbitrarily say that failure occurs when the strength of the material has decreased a certain percentage of the strength of mild steel, for example 25 or 50%. For a particular material and using data similar to that presented in Figs. 11 and 12 we can then plot a curve of time required at a particular temperature to cause a strength reduction to 25%, for example, of mild steel against temperature. Such curves for a variety of different materials are shown in Figs. 13 and 14 and present a summary of the results obtained in the Series II tests. The use of these curves can be illustrated by reference to Fig. 13 as follows: suppose that the maximum temperature encountered is 1400 F. Following along a 1400 F line, failure for mild steel would be expected to occur at about 170 hr, for corten at about 330 hr, and for mild steel painted with aluminum paint at about 300 hr. The graph would also indicate that the coated and stainless steels would have an expected life of more than 1000 hr, the maximum time for which the tests were conducted.

The Series III tests were concerned with the effects of lead and sulfur additions to the fuel. For the lead tests 3 cc per gal of tel were added to the fuel oil while for the sulfur tests 2.8% by weight of sulfur

was added by adding carbon disulfide to the fuel. The results are shown in Figs. 15 and 16. Fig. 15 shows the results obtained for mild steel for the maximum time of these tests of 500 hr. The differences shown in Fig. 15 are probably not greater than the experimental error. Similar results were found for mild steel with silicone paint and for corten. The results for enameling steel coated with A19 are shown in Fig. 16. Similar results were obtained for enameling steel coated with a base material. The results for the aluminized steel also showed similar results except that the lead and sulfur curves were interchanged. The stainless materials also showed effects similar to Fig. 16 but with lead having somewhat more of an

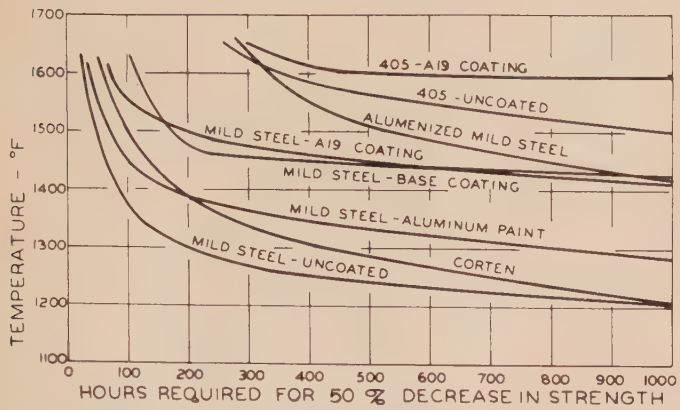

Fig. 14—Temperature versus time for 50% decrease in strength of different materials

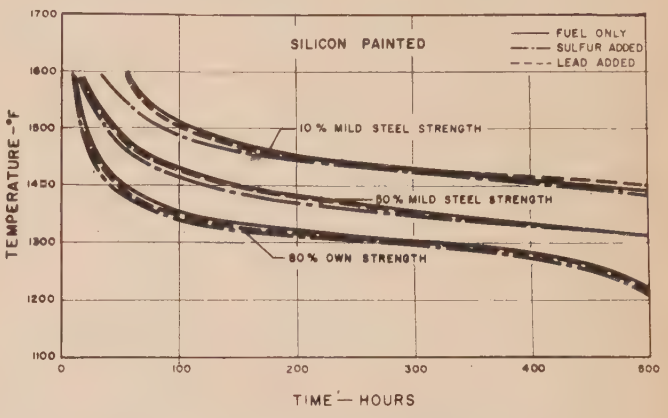

Fig. 17—Effect of sulfur and lead additions to fuel on mild steel specimens painted with silicone paint

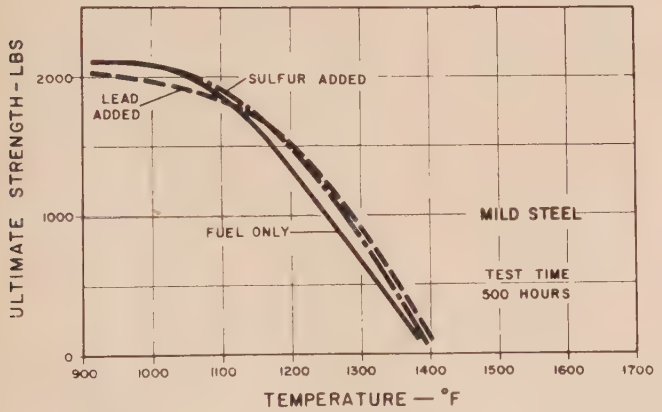

Fig. 15—Effect of sulfur and lead additions to fuel on mild steel specimens

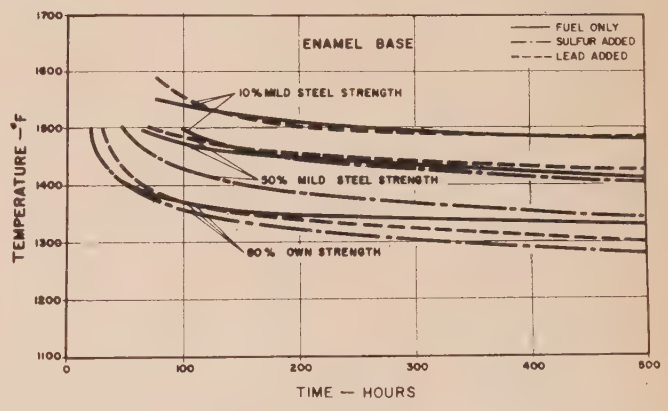

Fig. 18—Effect of sulfur and lead additions to fuel on enameling steel specimens coated with base enamel

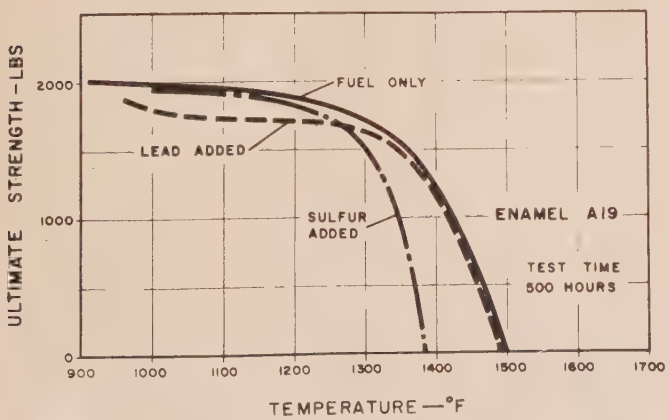

Fig. 16—Effect of sulfur and lead additions to fuel on mild steel specimens coated with A19

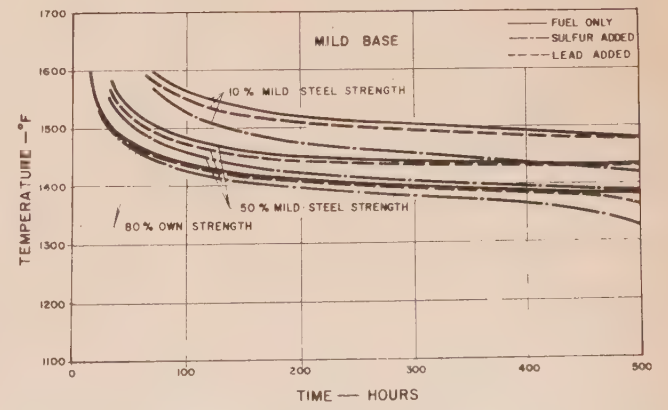

Fig. 19—Effect of sulfur and lead additions to fuel on mild steel specimens coated with base enamel

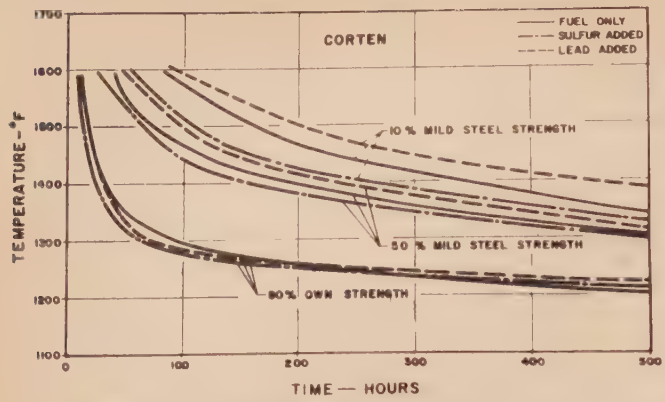

Fig. 20—Effect of sulfur and lead additions to fuel on corten specimens

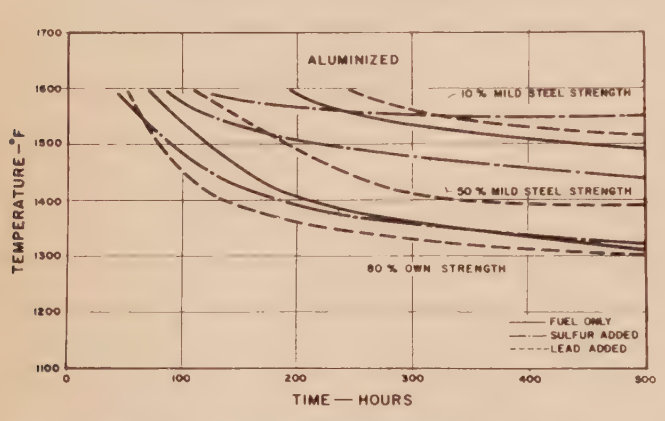

Fig. 21—Effect of sulfur and lead additions to fuel on aluminized specimens

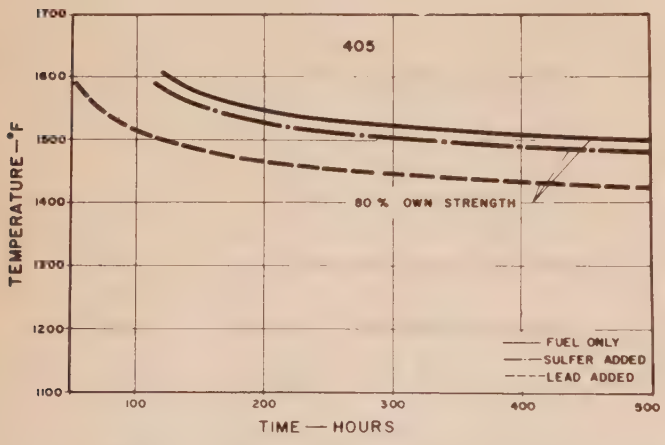

Fig. 22—Effect of sulfur and lead additions to fuel on 405 specimens

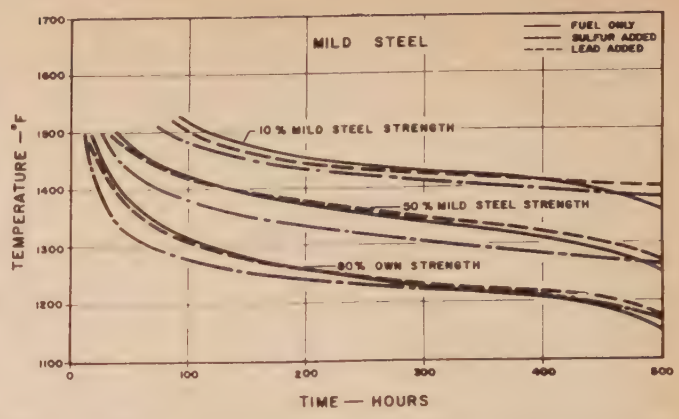

Fig. 23—Effect of sulfur and lead additions to fuel on mild steel specimens

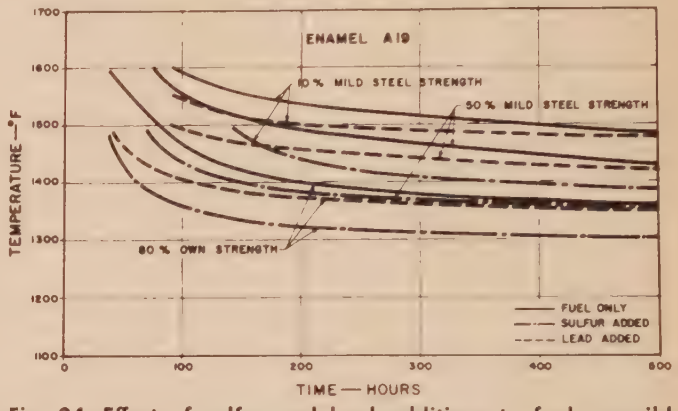

Fig. 24—Effect of sulfur and lead additions to fuel on mild steel specimens coated with A19

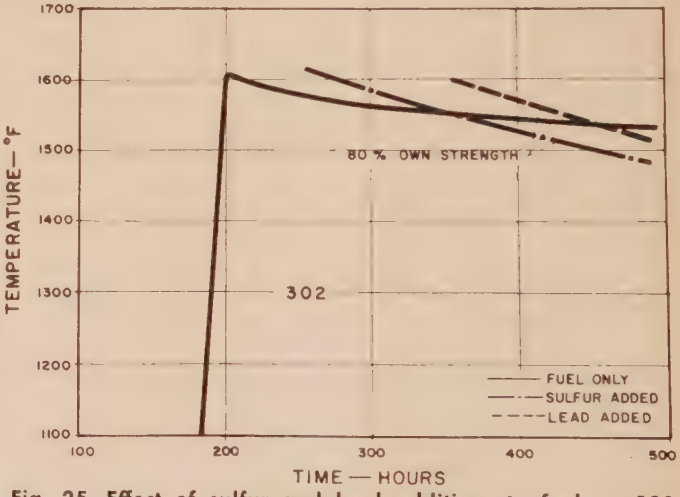

Fig. 25—Effect of sulfur and lead additions to fuel on 302 specimens

effect than sulfur. The results of the sulfur and lead tests are summarized in Figs. 17 through 26.

The previous tests were run with considerable excess air. It was decided, therefore, to repeat the tests with a richer mixture to check the effect of fuel/-air ratio both with and without sulfur added. In general the effect of sulfur was about the same at the richer mixtures as at the leaner mixtures. Fig. 27 presents comparison data showing the effect of different fuel/air ratios without sulfur or lead present.

In addition to the data taken using a cycling procedure, comparison data were taken with the temperature maintained at a constant value except for an occasional shutdown due to equipment failure. Fig.

28 summarizes the results of these tests and compares them with previous data obtained during the cycling tests.

Discussion of Results

The first conclusion reached from the field tests is that the primary factor in muffler failure (under the test conditions) is high temperature. It is true that the metallographic examination of the failed mufflers suggested other contributing factors. However comparison of the experimentally obtained temperature-time curves for mufflers in service with the laboratory obtained strength-temperature-time data for mild steel (Fig. 12) indicate that muffler operation is at the edge of the temperature range where any small increase in temperature will produce a very marked decrease in muffler life. This conclusion is further borne out by the oft-repeated statement in the field that, "the driver can burn out the muffler any time he wishes," presumably by simply overloading the engine and thus increasing the operating temperature of the engine and exhaust system.

Speaking of the experimentally obtained temperature-time histories there is contradictory evidence as to the importance of aircooling of the muffler. For example, examination of Figs. 4, 5, and 6 will show that the highest muffler temperatures were reached when using the chassis dynamometer. However, other temperature-time muffler data under service conditions show that the highest temperatures were reached not under high-load low-vehicle-speed conditions but under high-speed level-road runs indicating either that aircooling was unimportant or that the heat load on the muffler was higher. In this connection there is some evidence that combustion took place in the muffler. For example the data in Figs. 4, 5, and 6 show that under certain conditions the outlet gas temperature was higher than the inlet gas temperature and that the inlet gas temperature was higher than the temperature of the exhaust gas in the manifold.

It is true that unshielded thermocouples were used and that acoustic and velocity effects might affect the temperatures at a particular location. Nevertheless the general impression was obtained that under high and overload conditions some combustion occurred in the muffler. This condition might be responsible for the metallographic findings on the failed mufflers.

If the temperatures are too high for mild steel the obvious two remedies in so far as the muffler is concerned are either to reduce the operating temperature by better engine or muffler design or to use better material. Since it is difficult, if not imposible, to effect an appreciable temperature reduction by design, other materials were investigated as previously indicated. These materials were primarily either coated mild steel or stainless steels. The test results indicate that if the coating is diffused or diffuses into the metal it increases life at a particular temperature. For example metallographic examinations indicated that both the aluminum coating and chromium plating diffused into the mild steel but that the silicone and aluminum paints did not. Correspondingly the test results indicate that the aluminum and chrome coatings increase the life of the material.

Surprisingly enough, the base ceramic coating on the mild steel seemed to be as effective as the A19

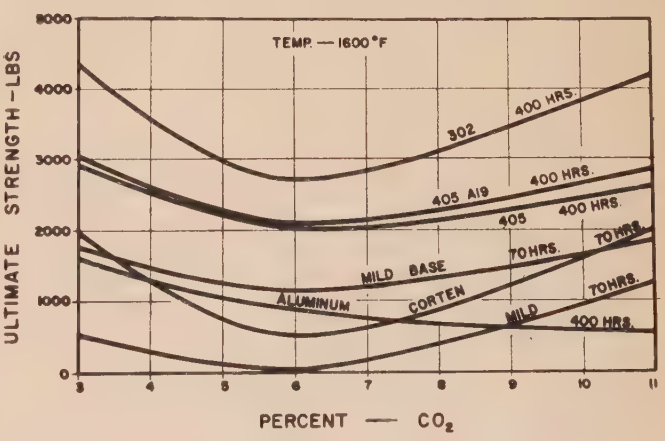

Fig. 27—Effect of fuel/air ratio on life of specimens

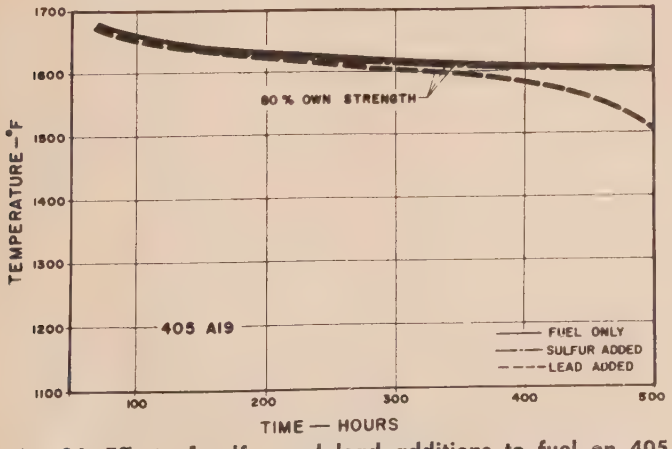

Fig. 26—Effect of sulfur and lead additions to fuel on 405 specimens coated with A19 enamel

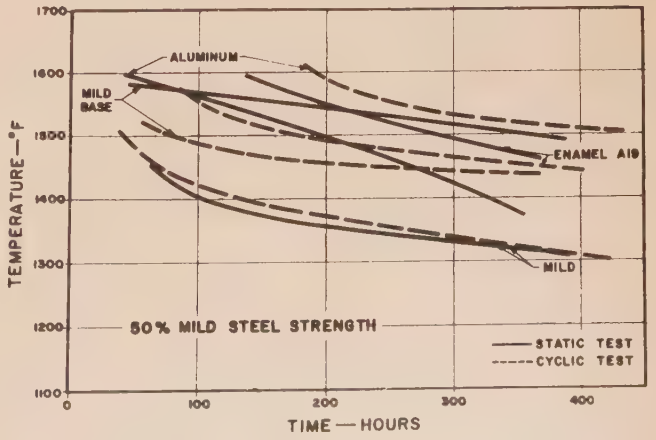

Fig. 28—Comparison of results from static and cycling tests

Fig. 29—Results of tests for ultimate strength on 302 specimens

ceramic coating. This result is somewhat different from the results of steady-state tests conducted on these materials at the Bureau of Standards [1] and from tests conducted in England.[2] However, both the test conditions and duration of tests were markedly different between these two programs and the program reported herein. It is also of interest to note in Fig. 28 that in short high-temperature steady-state tests the A19 coating appears to be best while under longer time lower temperature conditions the base coating appears to be best. It is also of interest to mention that some thermal chipping of the A19 coating was encountered although none was found with the base coating. Referring again to Fig. 28 it will be noted that all of the materials except the aluminized steel showed the steady-state test to be less severe. While it is understandable that the cycling test is more severe it is not clear why this trend should be reversed for the aluminized steel.

With respect to the stainless materials the tests show the 302 to be better than the 405 for high-temperature conditions. However, the tests also indicate that 302 might not be as good as 405 for the low-temperature portion of the temperature-time history. For example, Fig. 29 is a photograph of specimens made from 302 stainless after tests. The greater elongation indicating greater ductility is clearly observed at the higher temperature. The strength test data also indicate that the 302 is not as good in the lower temperature range. For example Fig. 30 shows that 302 is losing strength at the lower temperatures. This observation is supported by the known fact that 302 stainless steel is subject to intergranular attack when exposed to corrosive media after being held in the temperature range of about 800–1300 F. The general conclusion was thus reached that the 405 stainless is a better material for mufflers than the 302.

As a matter of interest tests were run on the 405 stainless coated with A19. As would be expected this resulted in improved life over the uncoated material and would seem to offer the maximum in muffler life.

With respect to the effects of lead, metallographic examination of the specimens always showed evidence of intergranular attack when lead was present in the fuel. It is undoubtedly this intergranular attack that caused slightly increased rates of corrosion, but in general the intergranular corrosion was not severe enough to affect the strength adversely.

A comparison between the laboratory tests and field results was made in two different ways. The first method used was to determine from the temperature-time records the fraction of time spent at high temperature for both the muffler and the laboratory specimens. This comparison showed that the laboratory specimens were at high temperature about three times as long as were the actual mufflers. A second comparison was made by noting the decrease in strength of the material used in one of the test mufflers whose temperature history was known. Fig. 31 shows the comparative strength of specimens taken from the indicated locations. This muffler had experienced a maximum temperature of 1180 F when used on a diesel engine and 1360 F when used on a gasoline engine. From the data of Fig. 31 the decrease in the strength of the muffler after 65 hr of use was found to be about 89%. From Fig. 11 the laboratory specimens under the same conditions (1360 F) were found to have decreased to about 64%. Or, putting it another way, under the same conditions it would only take about 20 hr using the laboratory test to produce a decrease in strength of 89% as compared to 65 hr in actual usage. Thus the 1000-hr laboratory tests should correspond to something in the order of 3000 hr (or 90,000 miles at 30 mph) of muffler tests.

As further confirmation of the tests, Fig. 32 shows a muffler which had a rolled aluminum-coated inlet tube and a noncoated mild steel outlet tube. Since both tubes were exposed to essentially the same service conditions, it is obvious that the coating was beneficial with respect to service life.

In summary, the following conclusions can be drawn:

1. The maximum metal temperatures encountered in truck mufflers are sufficiently high so that a small

[2] *Metallurgia*, Vol. 9, April, 1954, pp. 165–168: "Efficiency of Some Protective Treatments in Preventing Oxidation of Mild Steel at Elevated Temperatures," by A. H. Sully, E. A. Brandes, and R. H. Brown.

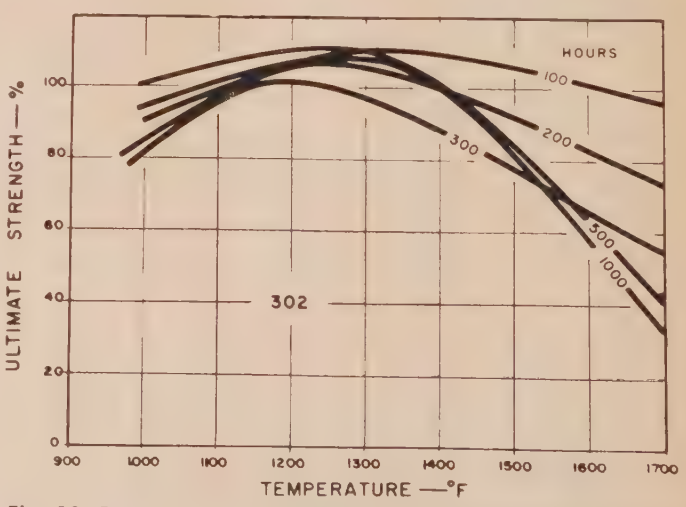

Fig. 30—Per cent ultimate strengh versus temperaure for 302 specimens

temperature increase will cause a rapid decrease in the strength of mild steel.

2. Although metallographic examination of failed mufflers suggests that other factors are present it is believed that high temperatures are primarily responsible for muffler failure under the test conditions.

3. Coated mild steels offer definite promise in extending either service life or permissible temperatures.

4. There is evidence to indicate that even in high-temperature service where condensation does not occur both lead and sulfur in the fuel tend to decrease service life.

Fig. 32—Muffler with coated inlet tube and uncoated outlet tube

Additional Comments by Authors

The following comments are answers to questions directed to the authors during oral discussion after the presentation of the paper.

It was intended that the text emphasize the fact that the temperature-time data taken on heavy-duty, over-the-road trucks clearly indicated that high-temperature corrosion was the chief source of muffler failure in this type of service. Thus, the laboratory tests were designed to give information on the ability of different materials to withstand temperature corrosion. Inasmuch as muffler failure in passenger-car service is most likely and primarily the result of low-temperature corrosion, the authors do not feel it wise to attempt to apply the results of these studies to passenger-car service.

With regard to the choice of materials, once again one must keep in mind the scope and intent of the tests. For heavy-duty trucks there is but little low-temperature corrosion and much more exposure to high-temperature conditions as well as violent thermal changes such as are encountered when a hot muffler is suddenly drenched with a spray of water thrown up by the wheels of the vehicle. The various materials in this study show the different degrees of resistance to prolonged exposure to high temperatures under conditions simulating actual truck operating conditions. Effective use of the charts in the selection of proper materials for use in the design of a muffler requires an intimate knowledge of the conditions under which the unit will be operated, tempered by a consideration of the economic factors involved.

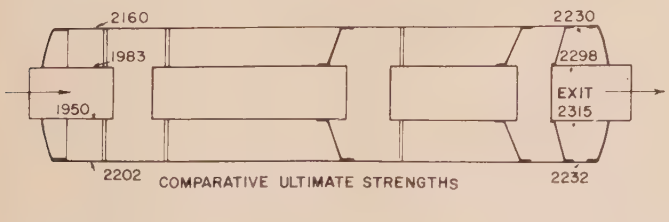

Fig. 31—Relative strengths of metal at different locations from muffler with known temperature history

With regard to thermal expansion and growth, a study of the photomicrographs will show the degree of disintegration of the crystalline structure of some of the best grades of stainless steel, particularly when operating in the critical temperature ranges. Because of this change in structure considerable elongation occurs and stresses in the muffler may result. This situation is aggravated by thermal expansion, and if means are not provided in the internal construction of the muffler for these parts to expand and contract, mechanical failure will occur in conjunction with chemical crystallization. Thus, some high-alloy steels will give a disappointingly low muffler life. Because ordinary mild steel is not so sensitive to intermolecular boundary-layer corrosion and consequent elongation over a wide range of temperatures there have been occasions where the mild steel gave better performance than the high-alloy steels.

Reference to the charts as well as field experience will show that some of the coatings give unexpectedly large protection to ordinary mild steel so that the net result is a greater total life for the coated mild steel mufflers than the uncoated stainless steels. The relative life of different materials is heavily dependent upon the conditions under which the mufflers operate, particularly as regards the chemical content of the gases and the temperature range.

In regard to electrochemical action between dissimilar metals in the muffler or to the use of dissimilar metal in the welding rods there has been nothing to indicate deterioration in actual field tests or in the microphotographs examined.

With regard to a comment on the effect of proper engine adjustment on muffler life, the authors are in complete agreement with the statement that a maladjusted engine can reduce muffler life to 30 days when normal expectancy is one year. As can be seen in the text, Fig. 12, at high temperatures comparatively small increases in temperature can cause almost catastrophic decreases in strength. Thus a maladjusted engine whose exhaust temperature may only be 50 F higher than normal may cause a decrease in muffler life of several times.

EFFECT OF FUEL ON STARTING

A COOPERATIVE study of the relationships between fuel volatility and the cold-starting and warmup performance of late-model passenger cars has been carried out by Standard Oil Co. (Indiana) and Ethyl Corp. Tests were made on seven 1956 passenger cars in an all-weather chassis dynamometer room at ambient temperatures of +40 to −20 F.

Fuel volatility was varied over a broad range. Warmup characteristics and sensitivity to fuel volatility varied widely among the cars tested.

A comparison of the results of this program with earlier test programs indicated that the modern V-8 engine is more critical to fuel volatility than its in-line predecessors.

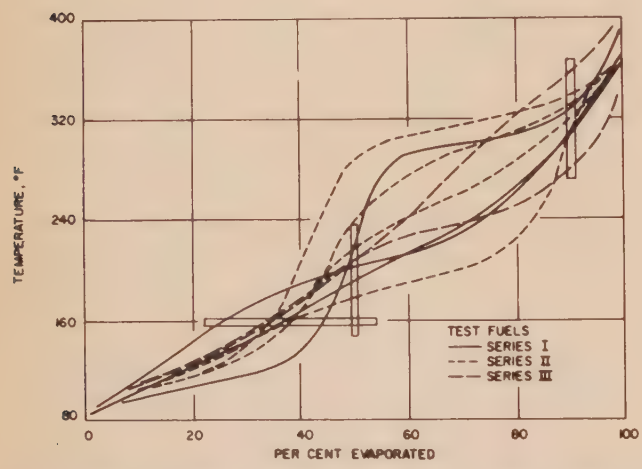

Fig. 1—ASTM distillation characteristics for volatility reference fuels. Bars represent range of per cent evaporated at 158 F, 50% and 90% points for fuels

Table 1—Properties of Test Fuels

Series	Fuel Number	Gravity, deg API	Reid Vapor Pressure	Distillation, ASTM D-86 Evaporation, % at 158 F	Temperature, F at 50% Evaporation	Temperature, F at 90% Evaporation
I	1	60.5	11.4	25.0	202	313
	2[a]	63.1	12.0	36.5	193	309
	3	57.0	11.6	43.5	220	329
II	4	65.8	11.2	36.5	178	312
	2[a]	63.1	12.0	36.5	193	309
	5	63.5	11.8	36.0	218	322
	6	57.7	11.8	37.5	240	332
	7	50.5	12.1	33.0	288	339
III	8	69.5	11.6	34.0	210	282
	2[a]	63.1	12.0	36.5	193	309
	9	62.5	11.6	34.0	207	357

[a] Base fuel common to three series of fuels.

THE cold-starting and warmup characteristics of engines are influenced by both fuel volatility and engine design. Continuous changes in refinery processing and automotive design necessitate frequent studies to establish winter fuel volatility specifications. Recent automotive trends to V-8 engines, automatic transmissions, multibarrel carburetors and automatic chokes, and the effect on fuel volatility of refinery processing to higher octane number created a need for this cooperative study between Ethyl Corp. and Standard Oil Co. (Indiana).

In this paper starting refers to the time to the first fire regardless of whether the engine continues to run. Warmup has reference to the performance of a car when accelerating at or near full throttle following a cold start. The warmup procedure was originally used by Taylor and Gibson [1] and has since been adopted by many other investigators. It has proved to be reproducible and sensitive to the effects of fuel and engine characteristics. Previous work [2] has shown cold starting correlated with front-end volatility as measured by the per cent evaporated at 158 F. Warmup performance correlated with both front-end vola-

[1] SAE *Quarterly* Transactions, Vol. 3, April, 1949, pp. 307–318: "New Approach to Evaluation of Fuel Volatility and Associated Engine Variables," by J. E. Taylor and H. J. Gibson.

[2] "Effect of Gasoline Volatility on Performance of Passenger Cars," by C. J. Domke, H. R. Taliaferro, and C. B. Tracy. Presented at meeting of API Division of Refining, New York, May 12, 1953.

VOLATILITY

AND WARMUP OF NEW AUTOMOBILES

G. T. Moore, R. D. Young, and H. A. Toulmin, Standard Oil Co. (Indiana) and Ethyl Corp.

This paper was presented at the SAE National Fuels and Lubricants Meeting, Tulsa, Nov. 8, 1956.

tility and the middle volatility as measured by the 50% evaporated temperature. Other investigators [1,3,4] have found some influence of the final volatility or 90% evaporated temperature.

Previously reported methods of evaluating warmup data have considered (a) the time to the first smooth acceleration, (b) the elapsed time to the first of three equal accelerations, or (c) the time to final warmup. This cooperative study uses average performance over a fixed time interval as a basis of evaluating car and fuel warmup performance relationships. This new system considers car performance during warmup, the warmup time, and performance at final warmup. Demerits are used to indicate the degree of performance below that of a perfect fuel.

Previous cold-starting and warmup programs,[1,3] have indicated that correlation exists between tests conducted on the road and on a properly controlled chassis dynamometer. Dynamometer data, however, were much more consistent because of limitations inherent in road testing, such as variation in atmospheric temperature, grade, wind velocity and direction, snow and ice, and traffic. Data were therefore obtained on a chassis dynamometer in an all-weather room. Three series of fuels were run in seven makes of 1956 cars at four temperatures. The test cars were in better mechanical condition than most cars on the road, and similar tests in "as-is" cars would probably result in poorer starting and warmup performance. A 5W-20 grade of oil was used throughout the program.

Test Fuels

Programs such as this require a series of fuels that permit studying one fuel volatility variable while holding constant other volatility factors. Volatility was measured by the ASTM distillation curve which is used for refinery control. Three series of fuels were chosen with each series having one control point variable while holding the other two constant. Table 1 contains the inspection data on each test fuel.

The three fuels of Series I varied in front-end volatility from 25.0 to 43.5% evaporated at 158 F. Series II fuels, a group of five, had a variable 50% point ranging from 178 to 288 F. The three variable 90% point fuels, Series III, varied from 282 to 357 F. One fuel, termed a base fuel, was common to all three series. For orientation purposes, Fig. 1 shows the ASTM distillation curves of the test fuels compared

3 "Cold-Starting and Warmup Performance of Late Model Passenger Cars," by R. K. Stone and K. C. Bier. Presented at meeting of API Division of Refining, New York, May 12, 1953.

4 SAE *Quarterly Transactions*, Vol. 1, July, 1947, pp. 441–447: "Engine Warmup with Present-Day Fuels and Engines," by J. G. Moxey.

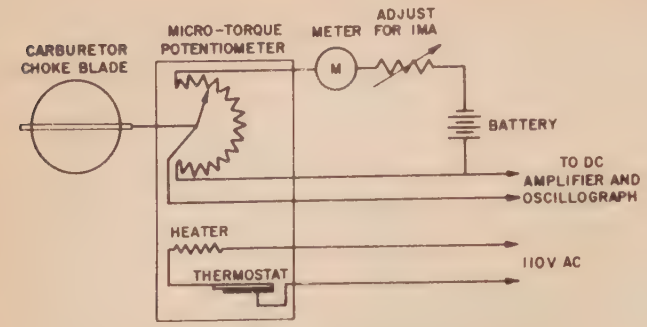

Fig. 2—Circuit diagram of choke position indicator

to the extremes of fuel volatility marketed in the north central area of the United States during January, 1956.

Test Cars

This investigation included tests on a Buick, Cadillac, Chevrolet, Chrysler, Ford, Lincoln, and Plymouth. All cars were 1956 models and were equipped with V-8 engines, automatic transmissions, and automatic chokes. The cars were placed in good mechanical condition since it was not the intention of this program to evaluate the effects of mechanical maladjustments. Calibrated carburetors of the latest production release were obtained from the automobile manufacturers and used for all tests. Since all the cars were relatively new, the manufacturer supplied intake manifolds containing deposits representative of 10,000 to 20,000 miles of operation because it is known that deposits in the intake manifold and in the exhaust crossover affect the transfer of exhaust heat to the mixture.

Test Equipment

The refrigerated chassis dynamometer room used in this program provides practically any climatic condition and is equipped with special dynamometer controls which closely reproduce both road load and full-throttle level road accelerations for any car. Cooling air for the test car is provided by a large blower which projects air at dynamometer room temperature against the front of the vehicle at a velocity equal to car speed. The blower may also be driven independently of the chassis rolls to provide an air blast for rapid cooling of an engine when the vehicle is not running.

A Brush single-channel, recording oscillograph was connected to the primary ignition circuit for the purpose of obtaining cranking speed, starting time, and total number of engine revolutions to start.[3] With this instrument each opening of the distributor breaker points created an impulse that was recorded on a paper tube moving at constant speed. This permanent record of the starting attempt gave an accurate indication of the time to first fire.

The automatic chokes on the cars were carefully set to the manufacturers' standard setting. In order to check the reproducibility of choke operation the choke blade angular position was continuously recorded. This instrumentation consisted of a micro-torque potentiometer connected to the choke linkage. The output of the potentiometer circuit was amplified and fed into a Brush recording oscillograph, as shown in Fig. 2. A record of choke position was recorded on a paper tape moving at constant speed.

Thermocouples were installed for measuring the following temperatures: ambient air, carburetor intake air, intake manifold mixture, jacket coolant, crankcase oil, and carburetor throttle blade. A multipoint potentiometer indicated and recorded these temperatures during each run and during the cool-down period.

A large burette was used to measure fuel consumed in each test.

Test Procedure

After installation of a test car on the dynamometer rolls and cooling to ambient temperature, the general test procedure for each run was to make a cold start and then proceed directly into the warmup phase of the test. Following the warmup test a new fuel was introduced into the fuel system and the engine was shut down to "soak" for the next run. The oil was changed between each run to eliminate oil dilution effects on the test results.

To facilitate rapid chilling between tests, the fresh crankcase oil was withdrawn from the crankcase through the drain, pumped through an external heat exchanger, and returned through the oil-fill pipe. At the same time the jacket coolant was circulated through the other side of the same heat exchanger. The cold room blower air blast chilled the coolant as it passed through the car radiator; and the coolant, in turn, cooled the oil in the heat exchanger. Forced circulation of the coolant through the engine block also speeded chilling of the engine. By using this method of cooling, engine temperatures were reduced to ambient in 2 hr, whereas cooling with the air blast alone required about 4 hr, and outdoor storage required from 6 to 12 hr.

Starting tests were made whenever a fuel was run for warmup performance, but specific starting tests were not made. In all tests the manufacturers' recommended starting procedures were employed. Current for the starter was supplied in the normal manner from the car battery. If the engine failed to start after 10 sec of cranking, it was started by any means available.

The warmup test procedure consisted of running the car at 20-mph road load and measuring its accelerating ability at the end of each minute of operation following the cold start until it was fully warmed

up. All of the cars except the Cadillac were equipped with torque-converter automatic transmissions. During the accelerating portion of the cycle all but the Cadillac were operated with the greatest throttle opening that could be obtained without downshifting to a lower gear ratio. The Cadillac was operated in fourth gear during the road load portion of the cycle and third gear at the detent during the accelerating portion. The measure of car performance was based on the speed attained at the end of 6-sec acceleration. Following each acceleration the car speed was returned to 20 mph by mild braking. The cycle was repeated at 1-min intervals until three successive accelerations gave the same final warmed-up performance or until operation was stabilized. Any evidence of rough operation, flat-spots, or backfiring of the engine during the warmup period was also noted. Readings of the fuel burette were taken at idle after the initial start and at idle at the end of each five cycles of operation.

Cold Starting

In Fig. 3 the per cent evaporated at 158 F is related to starting time for the average car at each test temperature. At 20 and 40 F little difference of either fuel volatility or ambient temperature is evident. Starting was satisfactory with all fuels, being accomplished in less than one engine revolution. This approaches the minimum possible starting time and increasing front-end volatility could show little, if any, improvement.

Fuel volatility proved to have a definite effect on starting at 0 and −20 F. At 0 F the fuel with the most volatile front end gave starting times that approached the lowest possible. The base fuel of 36.5% evaporated at 158 F gave adequate starts at 0 F and

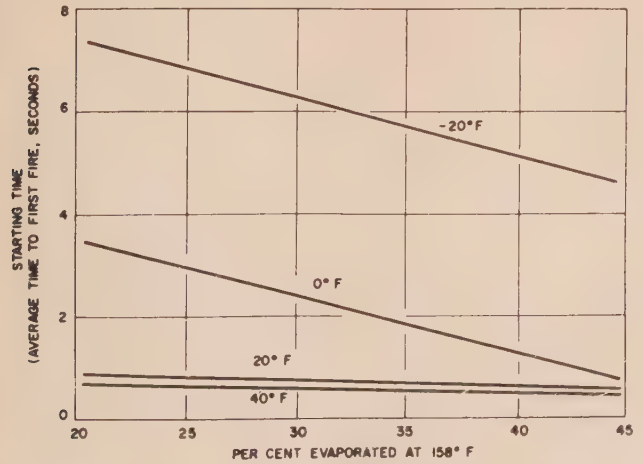

Fig. 3—Effect of fuel volatility on average cold-starting performance for seven 1956 passenger cars

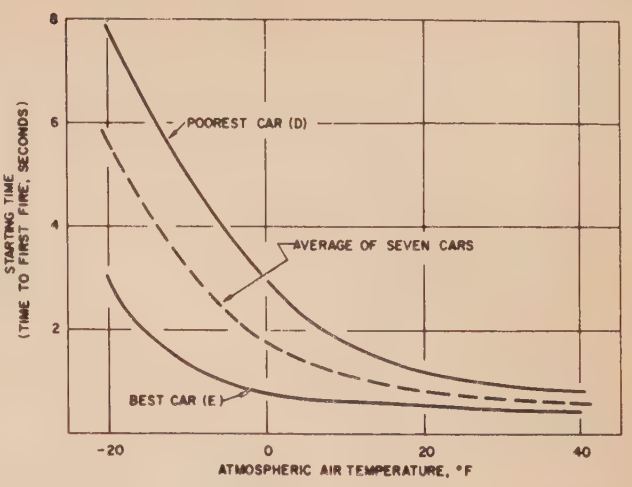

Fig. 4—Effect of temperature on cold starting of late-model passenger cars for 35% evaporated at 158 F fuels

did so quite dependably. At −20 F any start is a good start, and probably most failures in the field are not the fault of the fuel. At the lower temperatures several factors make starting more difficult. Cranking speed is lowered due to an increase in oil viscosity and a decrease in battery capacity. Ignition voltage is lowered because of the demand of the starter for more power, as well as the decrease in battery capacity. Also there is greater difficulty in evaporating the light ends of the fuel at low temperatures.

The effect of ambient temperature on starting is shown in Fig. 4 for the nominal 35% evaporated at 158 F fuels. Curves for the best (E), average, and poorest car (D) are indicated. At 20 and 40 F the starting ability of all cars was very satisfactory, and little difference between cars was observed. At 0 F some difference between cars was apparent although all were able to start and would continue to run dependably. At −20 F cars C, E, F, and G gave satisfactory starting times and continued to run. However, cars A, B, and D, although generally firing on one or more cylinders in from 6 to 9 sec, required priming with isopentane in several tests to make the engines run.

Warmup Performance

The test technique provided data relating car speed at the end of each 6-sec acceleration to time in minutes. Plots of such data present a vivid picture of car performance during warmup as affected by fuel volatility, air temperature, and engine variables.

An example of such a plot is shown in Fig. 5 for car G operating on the base fuel at 0 F. The first and second accelerations gave the same speed of 35½ mph. On the third acceleration a speed of only 30 mph was attained, probably as a result of the auto-

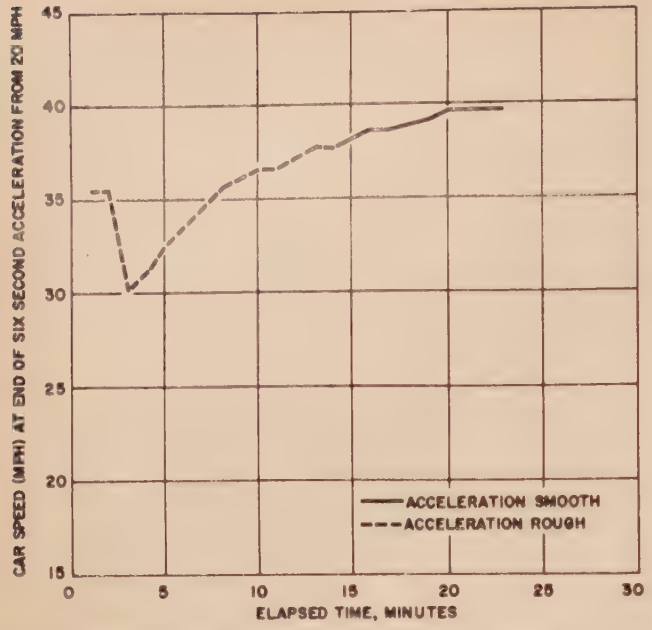

Fig. 5—Vehicle performance during warmup period for car G at 0 F atmospheric temperature with 50% evaporated at 193 F test fuel

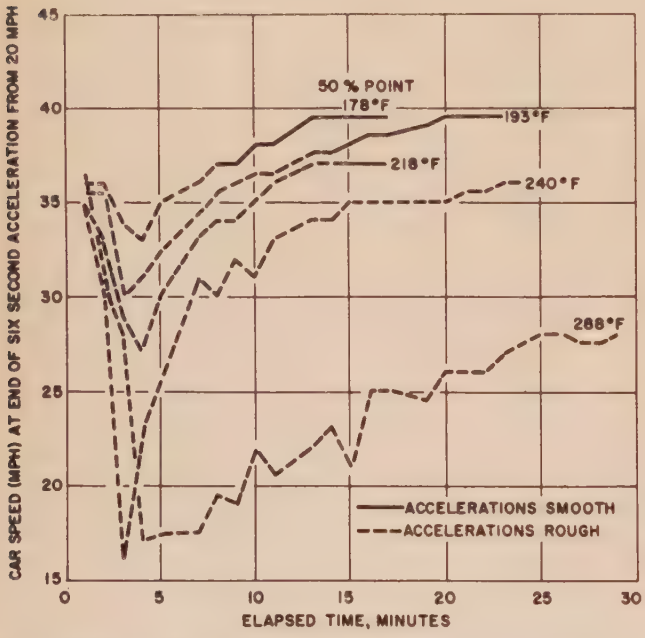

Fig. 6—Effect of fuel volatility on performance during warmup period for car G at 0 F atmospheric temperature using test fuels 35% evaporated at 158 F and 90% evaporated at 312–339 F

matic choke opening too rapidly. The fourth and succeeding accelerations resulted in recovery of speed until at 20 min the speed had reached equilibrium. Noticeable roughness in the accelerations was noted up to 14 min of operation, after which the accelerations became smooth.

In a similar manner the influence of fuel volatility on performance during warmup can be shown. Fig. 6

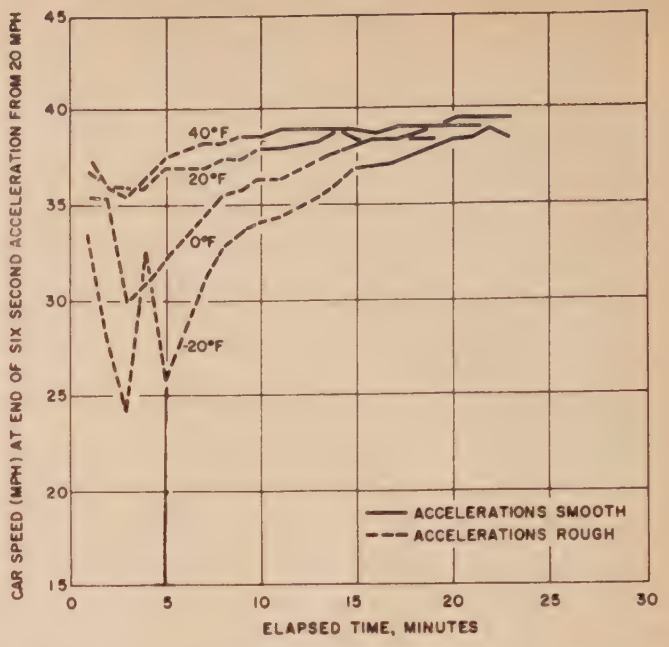

Fig. 7—Effect of atmospheric temperature on performance during warmup period on car G with 50% evaporated at 193 F fuel

shows the results of the tests on car G at 0 F on the series of fuels having variable 50% evaporated temperatures. A definite influence of the 50% evaporated temperature on vehicle warmup performance is evident as well as on the final performance after the vehicle temperatures reach equilibrium. Performance during warmup became increasingly poorer as the 50% point increased. Even after equilibrium was reached the fuels with 50% points of 218 F and higher gave increasingly poor performance with increasing 50% point.

The effect of ambient temperature on performance during warmup is shown in Fig. 7 for car G on the base fuel at the four test temperatures. The same pattern of results may be noted due to changes in air temperature as for changes in the 50% evaporated temperature.

Differences in car performance during warmup are shown in Fig. 8 for three of the cars and for the average of the seven cars when operating on the base fuel at an air temperature of 0 F. It is quite evident that the cars differed widely in warmup performance. Car D was the best and car C the worst. Car G was about midway between D and C but still poorer than the average of the seven cars. In the following discussion car C has been deleted because its performance raised doubts as to whether or not it was representative of later production.

Evaluation of Warmup Data

A perfect fuel, considered to have a uniform performance throughout the warmup period, was as-

sumed. Its performance for a car and test temperature was set 2 mph above the average final warmed-up speed of the four best fuels. The amount in miles per hour by which the average acceleration speed obtained with a fuel, during 23 min of operation, failed to equal the perfect fuel was termed its demerit rating. The 23-min interval was chosen because the performance of even the poorest fuel was generally stable by this time.

As an example of the demerit scheme, the two performance curves of Fig. 9 may be examined. Fuel 5 gave a slight deficiency in acceleration ability for a long interval, while Fuel 1 gave a large initial sag in performance but recovered rapidly. Choice of a warmup time would be difficult for Fuel 5. Use of the demerit scheme rates both fuels approximately equal and, as each fuel is less than ideal in a different way, it would be difficult to choose between them on any basis.

The warmup data were corrected for unavoidable variations in the "fixed" fuel variables using the base fuel as a standard; that is, the Series II fuels were corrected for variations in per cent evaporated at 158 F and 90% evaporated temperature from the base fuel values.

Demerits were used for warmup comparisons of fuels, ambient temperatures, and cars. The effect of the per cent evaporated at 158 F for the six cars is shown in Fig. 10. Its effect is small; a reduction of 10% (from 37% to 27% resulted in an average change of only 0.7 demerit. The spread between the best (D) and poorest (A and G) cars was 2 to 2½ demerits.

In contrast, Fig. 11 shows that the 50% evaporated temperature is effective in warmup control. Demerits increase rapidly with increasing 50% evaporated temperatures. On the average an increase of 25 F (from 185 to 210 F) gave a 1.25 increase in demerits. Going from 185 to 250 F gave a 5-demerit increase. The spread between the best (D) and poorest (A and G) cars was about 3 demerits at the more volatile 50% points and about 6 demerits at a 50% point of 250 F. The fact that the demerit curves tend to flatten out as volatility increased indicates an optimum 50% point beyond which no further improvement in performance should be expected.

The effect of the 90% evaporated temperature of

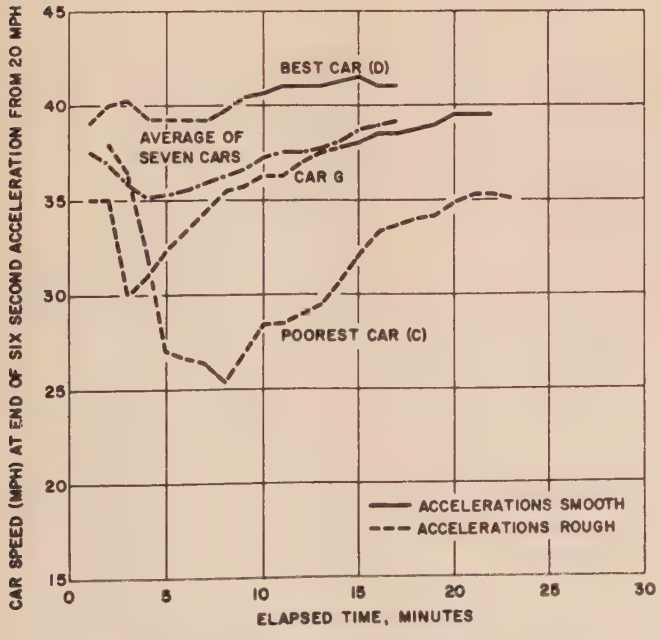

Fig. 8—Variation in warmup performance for seven 1956 passenger cars at 0 F atmospheric temperature and using 50% evaporated at 193 F test fuel

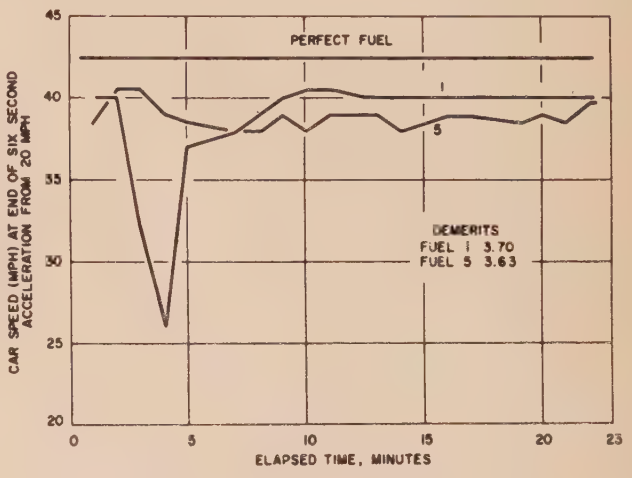

Fig. 9—Example of demerit ratings for test fuels

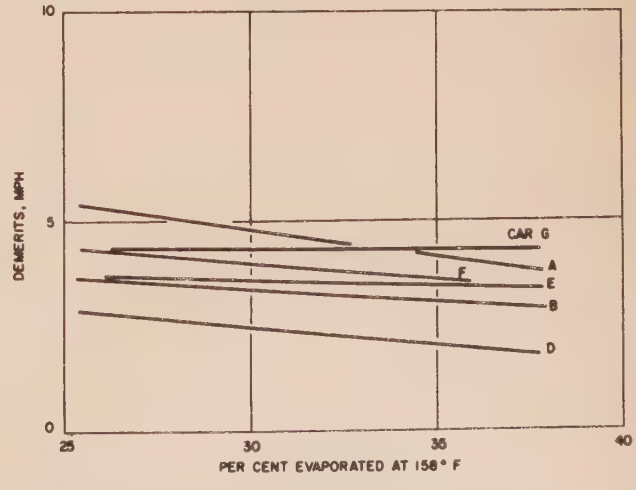

Fig. 10—Effect of per cent evaporated at 158 F on warmup performance (0 F atmospheric temperature with 50% evaporated at 193 F and 90% evaporated at 309 F test fuels)

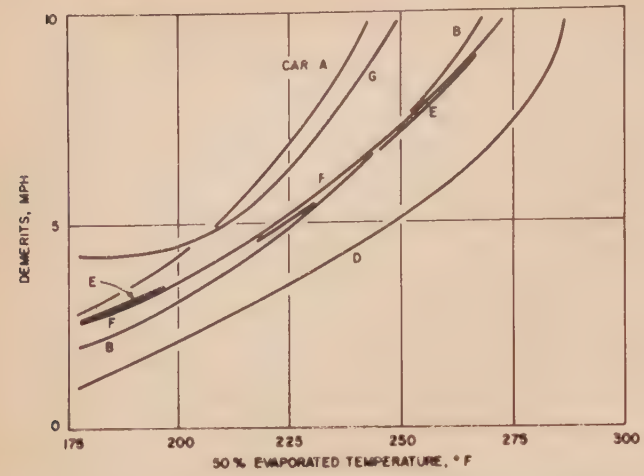

Fig. 11—Effect of 50% evaporated point on warmup performance under conditions of 0 F atmospheric temperature and using 35% evaporated at 158 F and 90% evaporated at 309 F test fuels

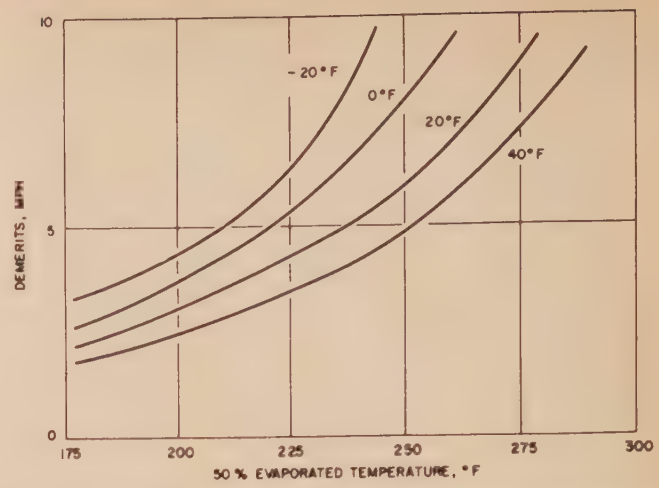

Fig. 13—Effect of fuel volatility and atmospheric temperature on warmup performance using test fuels 35% evaporated at 158 F and 90% evaporated at 309 F

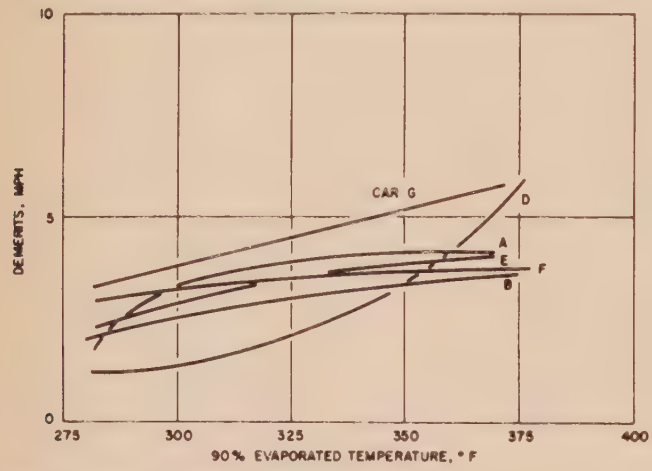

Fig. 12—Effect of 90% evaporated point on warmup performance at conditions of 0 F atmospheric temperature and using 35% evaporated at 158 F and 50% evaporated at 193 F test fuels

final volatility is shown in Fig. 12. Its influence is intermediate between that of the front and middle volatilities. The 90% effect on the average was quite uniform throughout the range investigated, increasing about 0.7 demerit for an increase of 25 F in the 90% point. At low and high 90% points the best and poorest cars gave a spread of about 2.5 demerits. Car D showed the greatest influence of the 90% point.

Fig. 13 shows the influence of ambient temperature on the performance of the average car, using the Series II fuels. The change of 60 F gave a spread of about 1.5 demerits at the lower 50% points and 6 demerits at a 50% point of 250 F.

The preceding data were used to establish nomographs as in Fig. 14 to simplify the data application to specific problems. The nomograph relates 50% point, 90% point, and ambient temperature to demerits. The influence of front-end volatility has been omitted because of its small effect.

Effect of Distillation Curve Shape

The fuel having the highest front-end volatility, 43.5% evaporated at 158 F, gave very poor performance during warmup. This fuel had a normal 50% point (220 F) and a normal 90% point (329 F). The only unusual part of this fuel was the sharp 154 F rise from the 40% point to the 60% point from 137 to 291 F. It was first assumed the poor performance resulted from an overrich mixture. Since a check fuel of high front-end volatility, Fig. 15, gave good performance, it is reasonable to conclude the volatile front end *per se* was not harmful but that the 60% portion of the distillation curve caused the poor performance.

This portion of the distillation curve was investigated by utilizing, in a fully warmed-up car (a) a series of binary blends having sharp temperature rises at different percentages evaporated, and (b) pure hydrocarbons covering a range of boiling points. Using the binary blends in decreasing order of volatility determined a go and no-go pair of fuels. This procedure located a narrow band of per cent evaporated separating a good from a poor fuel. Operation with pure hydrocarbons determined the critical temperature on the distillation curve.

Car G at 0 F gave satisfactory performance with isooctane and the 60% binary blend. Toluene and the 50% blend did not run satisfactorily. In Fig. 16 the shaded area formed by the boundaries of these four fuels locates the region of the distillation curve that controls fully warmed-up performance of car G at 0 F.

These results explain why the fuel with the most volatile front end failed to give satisfactory performance. The "hump" in the distillation curve of this fuel lay to the left of the critical area; insufficient fuel was vaporized for good performance at the low ambient temperatures.

Fuel Trends

Distillation characteristics of winter grade gasolines have evidenced only minor changes over the last 10-year period, according to Bureau of Mines national motor gasoline surveys. Refinery changes necessary to produce higher octane quality will tend to hold 50 and 90% points at their present levels or at most increase them slightly. Because warmup performance of new cars is closely related to these distillation points, any increase is significant.

A reduction of 50, 70, or 90% temperatures is expected to be costly. For example, removing the heaviest fraction of reformate from motor fuel to decrease these temperatures would represent a substantial penalty in octane number. This material is nearly pure aromatics and may rate over 105 octane (unleaded) and over 108 octane with 3-cc tel. Other means developed to lower these distillation points would probably cost about as much.

Refiners may be faced with a serious dilemma. At the very least, closer control will be desirable to obtain optimum warmup and maximum octane number. However, any decrease in the 50% point and above aimed at improving warmup would be expected to compound the already severe costs of raising octane level.

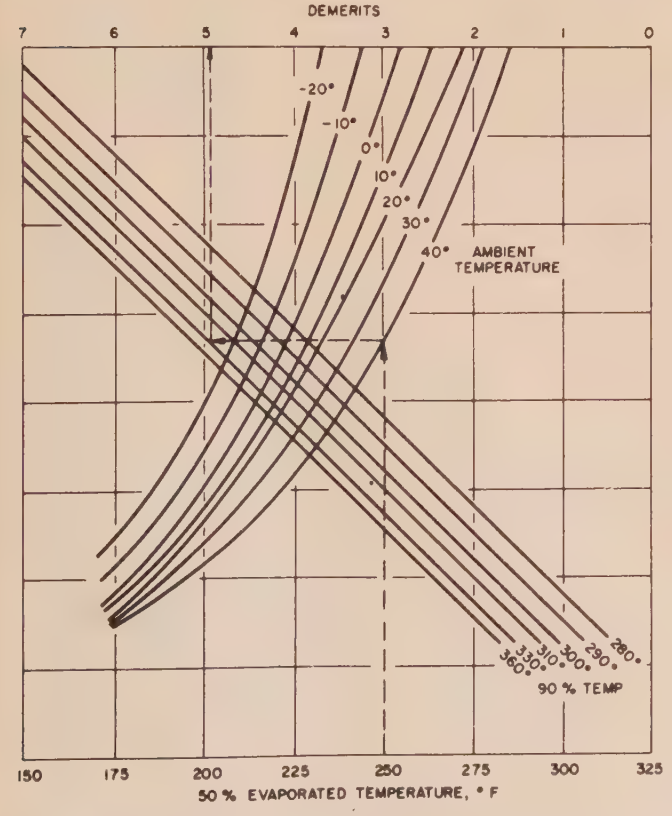

Fig. 14—Relationship of fuel volatility and atmospheric temperature to warmup performance for car B

Car Trends

The automatic choke operation of the cars tested varied considerably. In the interest of maximum economy some car manufacturers allow the choke to open rapidly, even though car performance may fall off rather badly after the first 2 or 3 min of the warmup period. Other manufacturers use slow-opening chokes to improve warmup, even though fuel economy suffers. Fig. 17 shows the operation of the automatic choke mechanisms of six of the cars tested at an ambient temperature of 0 F.

Fuel economy was measured for each five cycles of

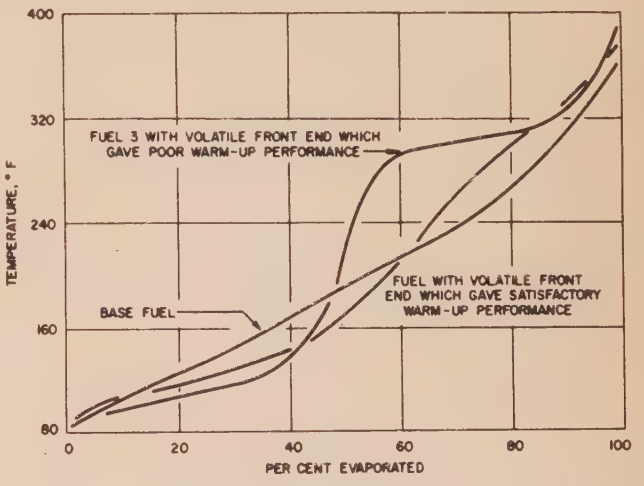

Fig. 15—ASTM distillation curves for three test fuels

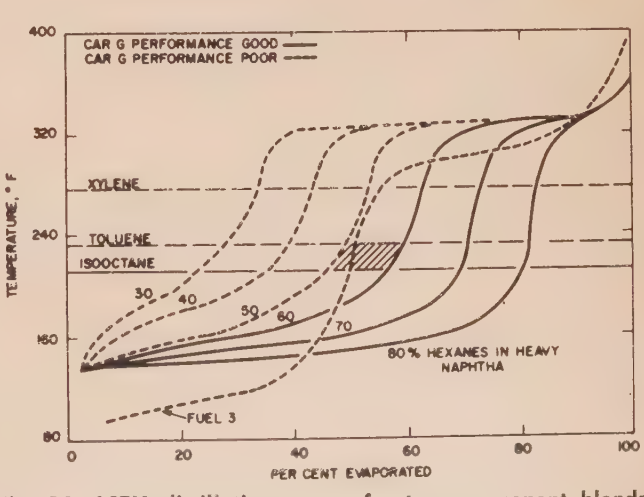

Fig. 16—ASTM distillation curves for two component blends and pure hydrocarbons at 0 F atmospheric temperature. Two component fuels made by blending mixed hexanes (135–137 F) and heavy naphtha (314–330 F)

operation during this test program, and the average results for each car at ambient temperatures of 0, +20, and +40 F are shown in Fig. 18. The results can be considered to typify low-temperature operation for short trips. The fuel economy obtained after six miles of operation was essentially the same as the 4- to 6-mile increment. It is interesting to compare the fuel economy of cars E and G, which had fast-opening chokes, with that of car F, which had a slow-opening choke. When all three cars were essentially at equilibrium warmup conditions (four to six miles of operation), the fuel economy was very nearly the same. However, car F suffered a loss in fuel economy relative to E and G during short-trip operation due to its long period of choke operation.

The fuel economy data were analyzed to determine if there was any measurable effect of fuel volatility on fuel economy. It was found that even though the vehicle performance was greatly improved with the more volatile fuels, fuel economy remained the same regardless of fuel volatility.

A comparison of engine warmup characteristics reveals that the modern V-8 engine requires a longer period for warmup than the in-line engines used in past programs. In the in-line engines both the intake and the exhaust ports were located on the same side of the cylinder block, and all exhaust heat was diverted to the manifold hot-spot during warmup operation. In the modern V-8 engine only the exhaust from one bank of cylinders is available to heat the mixture during warmup. It is believed the lower mixture temperatures prevailing in today's engines have made them more critical to the portion of the ASTM distillation curve between the 50 and the 90% points.

In order to compare warmup trends over a period of years, the data from this program were compared to warmup data obtained from previous tests conducted by Standard Oil Co. (Indiana) and by Ethyl Corp. When the base fuel for this program, 193 F 50% point, was used as standard, it was found that at 0 F the average 1956 car required 14 min and the best car required 9 min for warmup. In three previously conducted programs [1,2,3] using 1946–1947, 1949, and 1951–1952 cars the average warmup time was found to be approximately 10, 4½, and 9 min, respectively.

Figs. 19, 20, 21, and 22 show results of a comparison of the warmup performance of 1956 model cars with earlier models of the same make. Some makes have improved considerably in warmup performance in recent years, some have remained about the same, and others have shown a definite reduction in warmup performance. Since the engineering design factors which tend to improve warmup, that is, richer full-throttle carburetor mixtures, longer periods of choke operation, and more heat on the intake manifold are compromises which tend either to reduce fuel economy or lower the engine output of a warmed-up engine, these factors are weighed carefully by the engine designer. However, in some makes of cars cold weather performance could be greatly improved

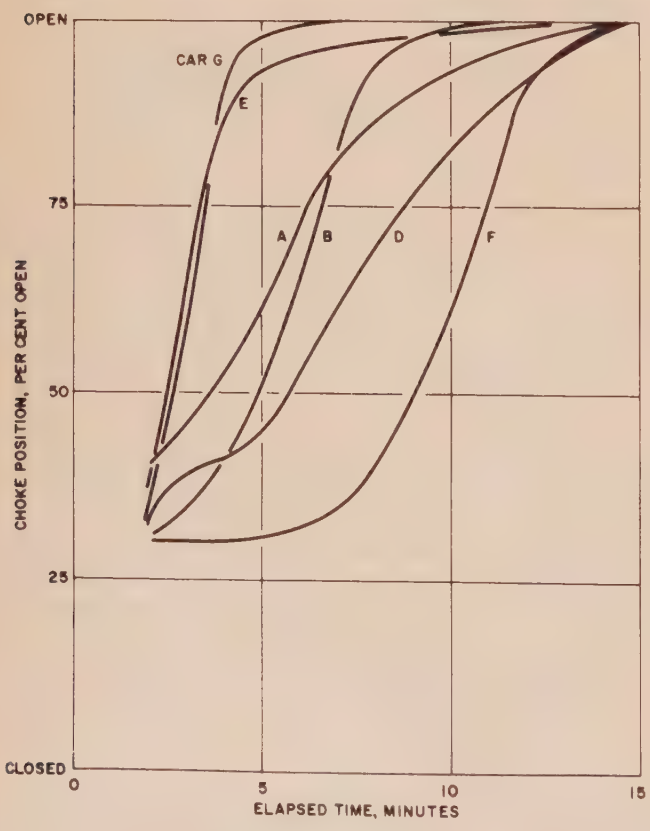

Fig. 17—Operation of automatic choke during warmup under conditions of 0 F atmospheric temperature. Choke opening during 20-mph road load portion of cycle

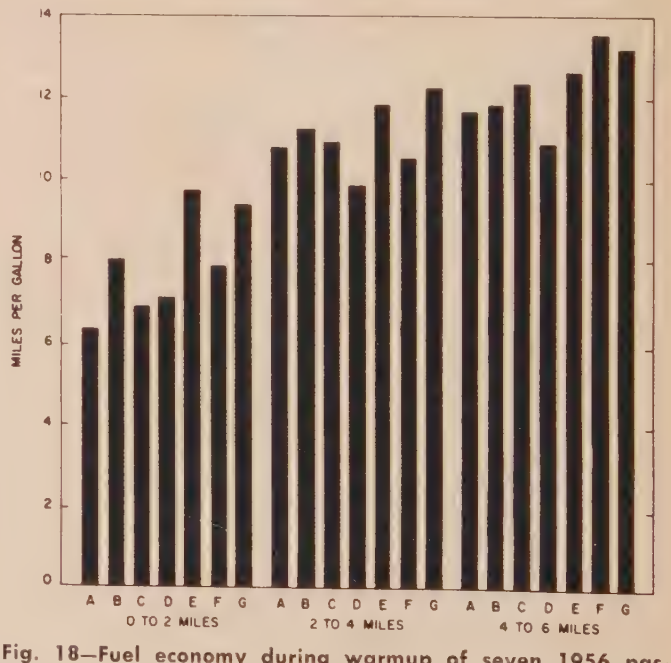

Fig. 18—Fuel economy during warmup of seven 1956 passenger cars. Each bar represents average of runs made at 0, 20, and 40 F

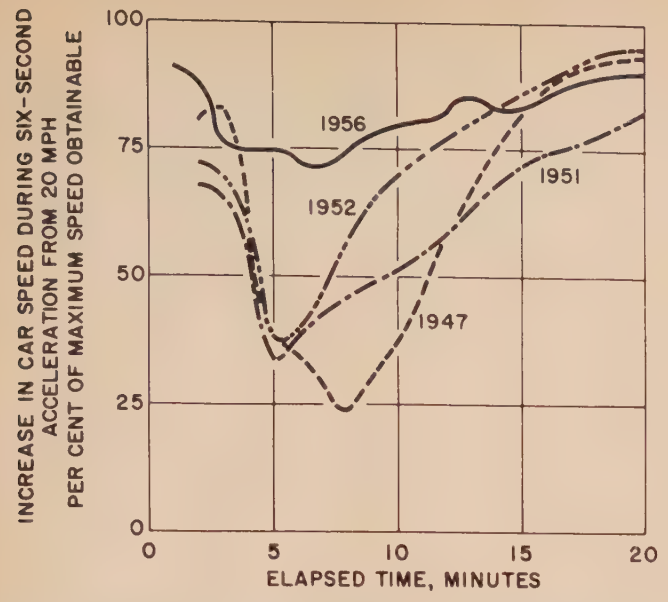

Fig. 19—Performance during warmup of four models of same make automobile at 0 F atmospheric temperature. All cars equipped with automatic transmission

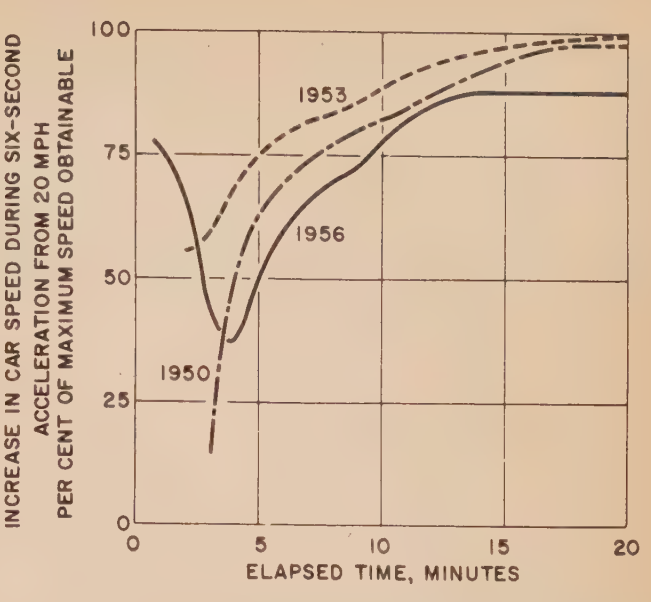

Fig. 21—Performance during warmup of three models of same make automobile at 0 F atmospheric temperature. All cars equipped with automatic transmission

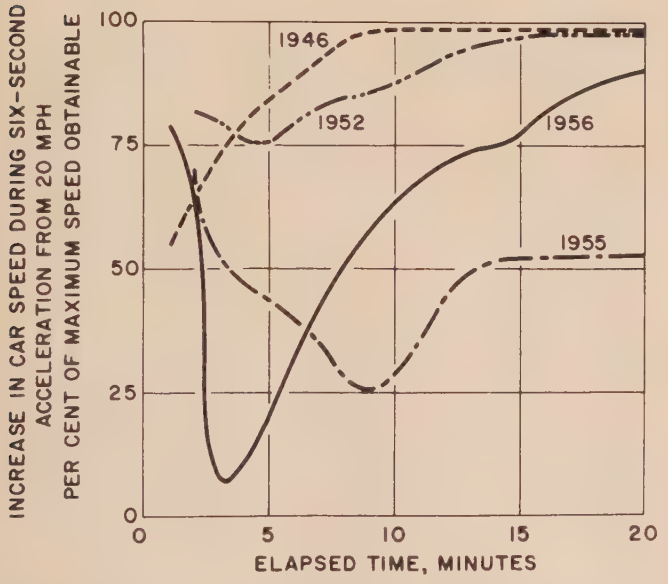

Fig. 20—Performance during warmup of four models of same make automobile at 0 F atmospheric temperature, 1946 model run at 10 F. All cars equipped with standard transmission

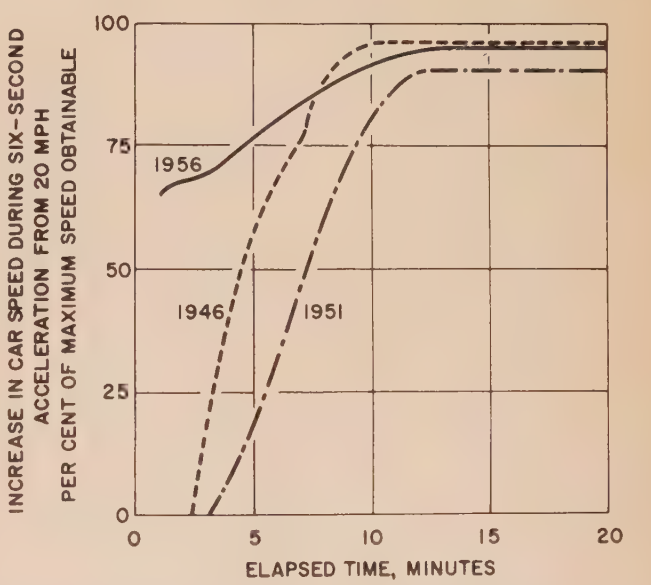

Fig. 22—Performance during warmup of three models of same make automobile: 1951 and 1956 models equipped with automatic transmission, 1946 model run at 10 F, and 1951 and 1956 models run at 0 F

by raising mixture temperatures slightly during warmup operation.

Summary

Cold-starting and warmup data for late model passenger cars in good mechanical condition have been presented. Cold-starting performance is independent of ambient temperature or fuel volatility within the range of normal winter grade fuels at 20 F and above. Below 20 F cold starting improves with increasing front-end volatility. Warmup characteristics of the cars tested and their sensitivity to fuel volatility varied widely. Warmup performance was controlled to the

greatest extent by the 50 to 70% evaporated portion of the ASTM distillation curve and only slightly by the final volatility and the front-end volatility. A comparison of the test results of this program with earlier warmup programs indicates that the modern V-8 engine is more critical to fuel volatility than its in-line predecessors. It seems evident from the results presented that the automotive and the petroleum industries have a mutual responsibility to provide the customer with satisfactory cold weather performance. Close cooperation is necessary to prevent conflicts between automotive design and fuel volatility trends.

Acknowledgments

The authors wish to express their appreciation to the automobile manufacturers for their cooperation and to R. A. Gustafson of Ethyl Corp. and C. J. Domke and J. E. Getz of Standard Oil Co. (Indiana) for their participation in the program.

DISCUSSION

Wants Relationships Between Engine Design and Warmup Performance

—R. E. Steinke

California Research Corp.

IN 1952 we completed a joint program with Ethyl Corp. testing 1950–1952 model cars. This work was summarized in a paper by Messrs. Stone and Bier in the 1953 midyear meeting of the API. There were two important conclusions resulting from this work:

1. Warmup characteristics and sensitivity to fuel volatility varied widely among cars.

2. The modern V-8 engine is more critical to fuel volatility than its predecessor.

In subsequent years, we have tested an additional group of 15 cars including 1953, 1954, 1955, and 1956 model cars. Data on 1953 and 1954 model cars substantiated the 1952 conclusions that there was an increase in severity of the warmup problem with the higher output engines. However, in testing 1955 cars, we found two cars of different makes which had excellent warmup characteristics. Only one car previously tested in our past programs bettered the warmup performance of these two cars. Warmup characteristics of the two 1955 cars compared to the 1950–1952 model cars tested in 1952 are shown as cars B and C in Fig. A.

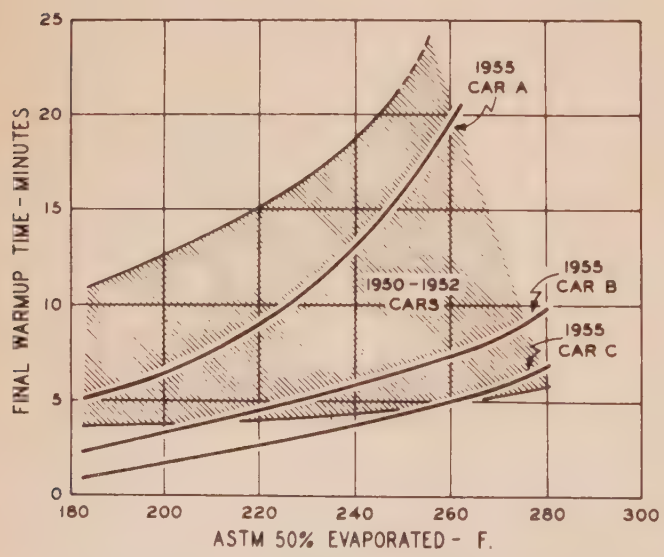

Fig. A—Effect of fuel volatility on warmup performance of late-model cars at 30 F atmospheric temperature

The engine of one of these two cars (car B) was almost identical in type and dimensions to that of another car tested of the same year but of a different make (car A, Fig. A) which had much poorer warmup performance. In an effort to determine the cause for the difference in warmup characteristics, several measurements were taken. Fuel/air ratios were measured, and manifold stove section, fuel-air mixture, manifold casting, and oil and water temperatures were measured during warmup. None of these measurements was sufficiently different to account for the warmup difference.

Intake manifold design was also studied, and manifold and port volumes were measured. Manifold designs were identical in appearance, and their volumes varied less than 1%. The port volumes varied considerably. The car with poorer warmup performance had port volumes about 75% as large as the car with faster warmup. This would seem to vary in the wrong direction for creating turbulence and does not explain the warmup difference. The chokes of both of the cars were almost completely opened at 5 min.

It seems to us that the relationship between engine design and warmup performance would be a most fruitful field of research for equipment manufacturers.

Discusses Devices To Aid Warmup Performance

—R. I. Potter

Ford Motor Co.

THE authors have shown rather convincingly that 50% evaporated temperature of a fuel is the most important single criterion in cold warmup. They have also shown that the 90% evaporated temperature, if varied over a range of 100 deg, is intermediate, though considerably less important; finally, they have shown that the least important criterion in the range tested is the per cent evaporated at the 158 F point.

In arriving at these conclusions, the authors have devised an ingenious scheme of demerits to express these performance parameters, and have presented a very usable nomogram clearly relating fuel volatility and ambient temperature to warmup performance. Automotive engineers and fuel technologists alike will benefit by this nomogram.

The authors have devoted a portion of their paper to fuels of

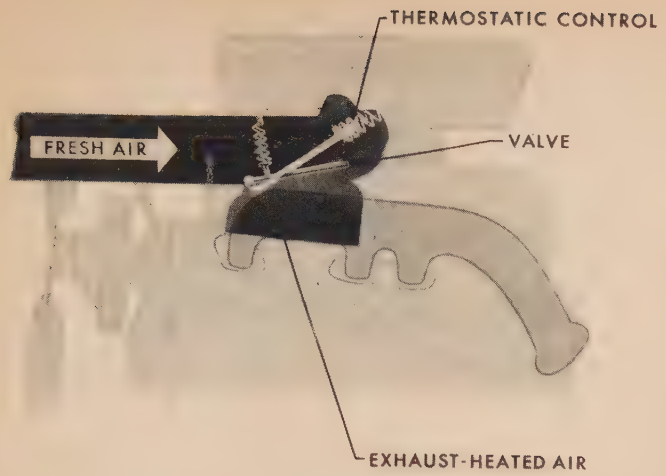

Fig. B—Automatic heater for carburetor inlet air during warmup

unusual or "gapped" distillation ranges, pointing out that they do not accelerate or warmup properly. The analogy to these fuels is the fireplace kindled with only newspaper and logs and minus kindling and intermediate wood—under these conditions combustion is slow and erratic. Much the same effect is achieved with gasoline by gapping or cutting out a 75-deg section of the distillation range in the midpoint, or pressurizing with propane instead of butane.

Continuing with the subject of warmup, the automotive industry also can assist good performance at low temperatures with regard to warmup by equipping engines with an additional mechanical device for temperature control. Such a device is an automatic heater for carburetor inlet air during warmup periods. A picture of this device is shown in Fig. B. From the diagram it will be observed that as soon as the engine starts and the exhaust manifold begins to heat in cold winter operation the carburetor intake air is drawn over the exhaust manifold, thereby warming the inlet charge. This device is controlled by a built-in thermostat spring, which operates to heat the inlet air at temperatures below approximately 70 F. The effect on inlet air temperature of this device with regard to warmup is shown in Fig. C.

Not only does the device aid engine warmup by supplying heat and assisting in taking advantage of today's responsive gasolines with built-in warmup volatility; but also, at the other end of the scale, it tends to improve the octane requirement and the power output of the engines under road load conditions. It does this by collecting the cold air from directly in front of the vehicle and supplying it immediately to the carburetor, before it becomes preheated by passing through the radiator and fan and over the hot engine.

Another benefit to the petroleum industry from devices of this type that control the temperature of carburetor inlet air lies in their ability to prevent carburetor icing with either normal or very volatile fuels without deicers. Carburetor throttle-plate temperatures during warmup at 0 F are shown in Fig. D. Temperatures not only rise much more rapidly during the 30-mph part-throttle test, but the initial downward temperature plunge at the throttle plate occasioned by evaporative cooling was reduced from 6.5 to 3.5 min. Similar data exists for throttle-plate temperature response in the 20 to 30 F ambient warmups.

If this device or devices of similar nature receive more widespread adoption by the automotive industry, the petroleum industry will be freed from the considerable expense of including "deicers" in gasoline for use during the few days of the year when weather conditions demand them, and at the same time be freed from possible penalties on octane requirement often occasioned by high underhood inlet air temperatures. Even further advantages would accrue to fuel purchasers who would pay no penalty in terms of gasoline mileage reduction and hot stalling and starting for overly volatile winter gasolines (Fig. E).

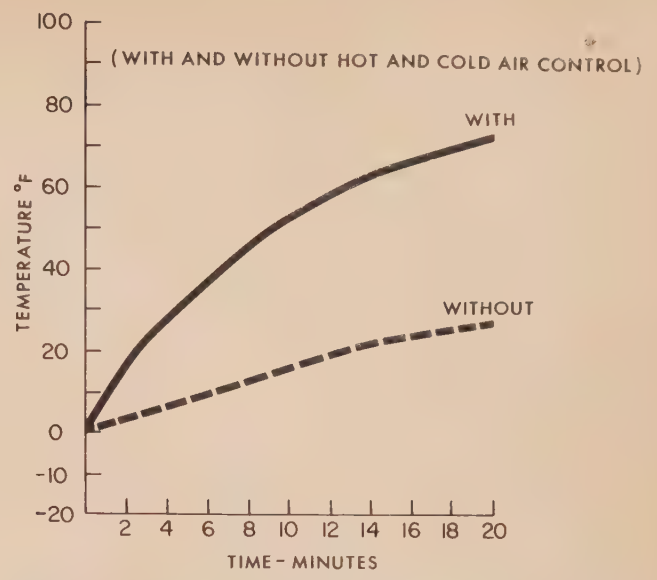

Fig. C—Comparison of carburetor intake air temperatures at 20-mph warmup and 0 F ambient temperature

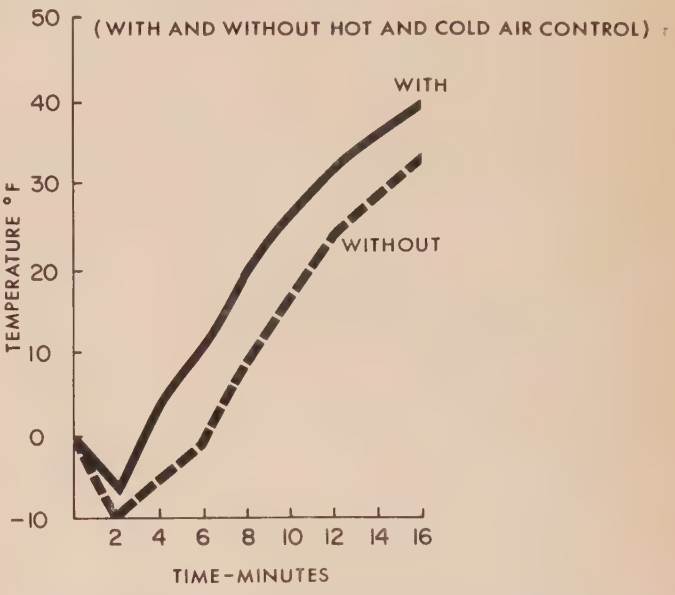

Fig. D—Comparison of carburetor throttle plate temperatures at 30-mph warmup and 0 F ambient temperature

In 1956 this combination air heater and fresh air inlet with thermostatic control was standard equipment on the Lincoln cars. In 1957 it is being added to the Mercury automobiles.

Compares Effect of Choking on Car Performance

—J. P. Hamer
Esso Research and Engineering Co.

The authors have pointed out that future gasoline manufacturing operations will probably result in gasolines with relatively high 50% distillation temperatures. They also state that gasoline cost will increase if present volatility levels must be maintained. We believe that this is an important point which would certainly merit the attention of the car manufacturers. The objective of such attention might be the improvement of warmup character-

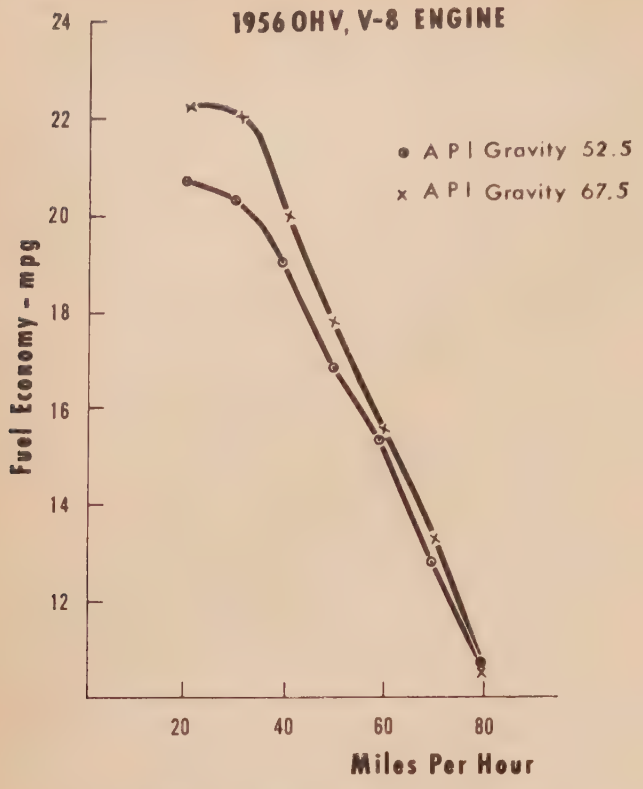

Fig. E—Effects of API gravity on fuel economy

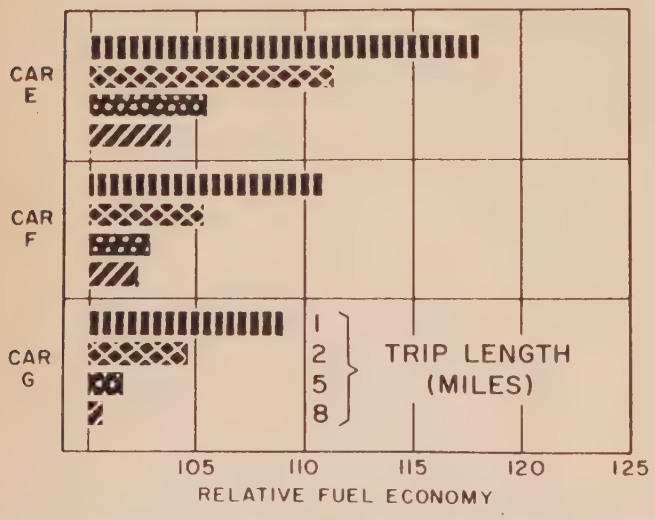

Fig. F—Relative fuel economy of Fuel A versus Fuel B for best choke adjustment at 30 F

istics of automobiles so that these expensive gasoline manufacturing operations can be held to a minimum.

Another interesting question raised by these papers is concerned with fuel economy. The authors point out that there was no effect of fuel volatility on fuel economy at manufacturer's recommended choke setting. This is probably true, but the question arises as to the validity of this conclusion if the chokes had been adjusted to match fuel volatility. Esso Research and Engineering Co. has carried out some tests on two fuels whose properties are shown in Table A.

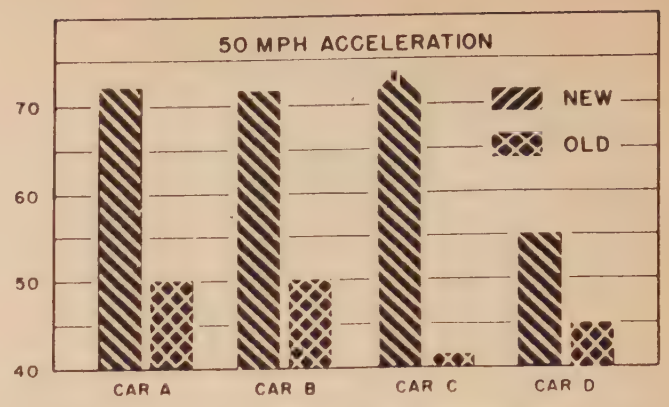

Fig. G—Comparison of intake manifold temperatures, F above ambient

Table A—Fuel Economy Studies

	Fuel A	Fuel B
Rvp	12.6	9.8
% D + L at 158 F	33.0	23.0
% D + L at 212 F	56.0	47.0
90% pt., F	305	345

These fuels were tested at 30 F in three cars operated on a chassis dynamometer. The operation was designed to simulate city-type driving in the New Jersey metropolitan area. Two cars had automatic transmissions with automatic chokes, and the other was manually choked.

The chokes were first adjusted by trial and error to the leanest setting on each fuel which would give satisfactory full-throttle accelerations at the test temperature. The settings were, of course, different with each fuel. After soaking at 30 F, the cars were operated under a strictly repeatable driving regime. In each car, a significantly better fuel economy was obtained with the more volatile fuel, as shown in Fig. F. This improvement tended to decrease with longer length trips. This latter behavior has been shown by the authors. The car manufacturer has to set his choke to take care of the average—or poorer than average—fuel volatility on which his cars will operate. These data show that improved fuel economy can be obtained by leaning out the choke to match the properties of the more volatile fuels available in the field.

Recent model cars give poorer warmup performance than their 1946–1949 counterparts, and the reason for poorer behavior of current model cars is that chokes open quicker than in the earlier models. In some instances, in fact, they may open so quickly that poor car performance results. For lack of a better explanation, this seems logical to us. Messrs. Moore, Toulmin, and Young have also suggested that the poorer performance in current model cars may also be due to their lower fuel-air mixture temperatures. Some temperature measurements which have been made at our laboratories do not seem to agree with this view. Admittedly, the determination of accurate air-fuel mixture temperatures in the intake manifold of a passenger car is a difficult problem, but these data will be presented for possible interest in connection with these papers. Fig. G shows that during a 50-mph acceleration, 1955 and 1956 model cars give higher mixture temperatures in the manifold than 1946–1949 models. These measurements were made under fully warmed-up conditions, but the same relative effects would probably be noticed under operating conditions in which the engine has not yet reached a warmed-up condition.

THE AUTOMOTIVE GAS TURBINE — Today and Tomorrow

G. J. Huebner, Jr., Chrysler Corp.

This paper was presented at a Detroit Section Meeting, Detroit, Oct. 8, 1956.

THIS paper discusses the progress of research on the automotive gas turbine and predicts its future potentialities.

Comparison of gas turbines and presently used engines shows the possibilities of the gas-turbine applications.

Design, construction, and testing of gas turbines are discussed, especially in the light of economy and performance.

MOST of the ideas presented here have been known for years, some of them for centuries, but the ability to achieve concrete embodiment of such ideas today in useful and economically sound form can take place only with the progress achieved by many men and many groups in many fields. The classification and combination of these ideas and elements into useful and workable form is the function of what we call research. Presented here is an interim progress report of the functioning of a research team in action using the methods and facilities made available to science by corporate organization and supported and encouraged by a sympathetic and alert management.

The next 10 years will see radical changes in the automobile. Powerplants will probably see as many changes as any other part of the car. I am not unfamiliar with the future of the piston engine or of the possibilities offered by the free-piston–gas-turbine combination. In each of these fields, we will see many interesting and valuable improvements, but I do not feel that there will be fundamental changes beyond known principles in the foreseeable future. Without fundamental discoveries in reciprocating or free-piston engines, the gas turbine shows the greatest promise of any new type powerplant for American passenger cars and trucks in the next decade.

At Chrysler Corp. we have been actively and continuously working on gas turbines for more than 10 years. Many of the same people that started out 10 years ago on turbines are working on them today. They represent an accumulated backlog of experience which can be brought to bear with great effect on turbine problems. But equally important to us is that most of these people spent the earlier portions of their professional lives in the design and development of automotive and aircraft reciprocating engines, and some of them still do so as the occasion arises. The corporate activity within which the turbine group operates is also equally concerned with research on future automotive piston engines, including free-piston engines, so that the necessities of automotive powerplant requirements are as vital to the turbine effort as they are to the reciprocating engine effort. These data although optimistic are not

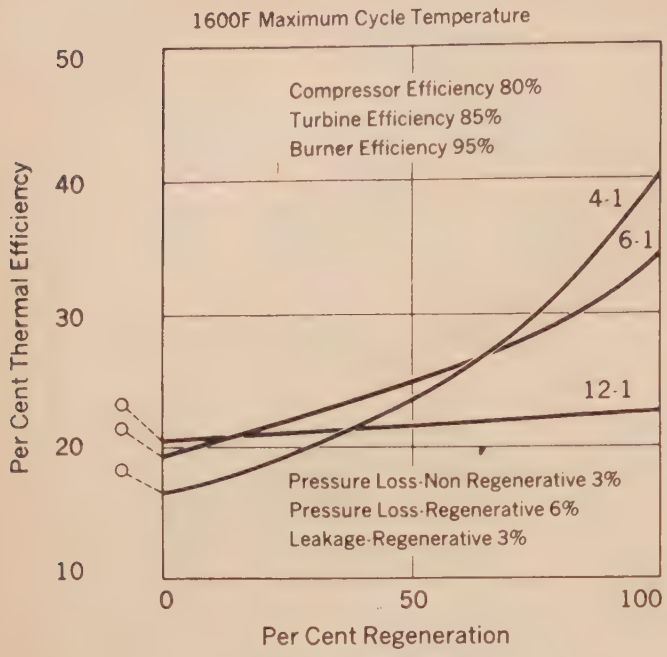

Fig. 1—Effect of regeneration and pressure ratio on thermal efficiency of engine

complete because our work is not finished, but they represent down-to-earth engineering and are as scientific as we know how to make them.

A gas turbine is important since it has the greatest potential for both near and long-range future development. It rivals conventional piston engines today in its efficiency, and there are many known methods of improving the cycle efficiency still further! It will be possible to get greater and greater efficiency from the engine as time goes by. Nor would the introduction of a gas turbine in production see an end to the improvement in the cycle. A thermal efficiency of 40% in the next decade seems possible, and this compares with the 28% which is about as good as we can do today with automotive piston engines.

The gas turbine has a torque curve which is the rival of the steam engine. It is possible for it to match the power requirements of an automotive vehicle much more closely than a reciprocating engine.

In addition to having greater future potential thermal efficiency and better output characteristics for an automobile, the turbine is a durable engine. In all our studies so far, we have not found anything that contradicts the experience of the USAF; namely, gas turbines last longer than piston engines. I cannot claim that we have now solved all the problems necessary to give us optimum durability, but we can foresee solutions to the problems that confront us. Our studies have been carried out on a large enough scale to give us confidence that we have already been confronted with most of the problems. At the present time we could probably design a gas turbine to last indefinitely. This could be done by increasing the mass of rotating parts and decreasing the temperature and speed at which they operate. Such an engine would last nearly forever, but it would also take nearly forever to get away from a stop light. We are glad that such steps are not necessary, and that advances in the field of high-temperature materials and refinements of our design techniques are increasing the durability of the few remaining troublesome parts of the engine.

There are no mechanical vibrations originating from the gas-turbine engine. It is a surprising experience to drive a turbine car because the smoothness of the engine makes apparent any defect elsewhere in the chassis that has formerly been masked by the vibration of piston engines, smooth as they are.

A gas turbine is about as simple as an engine can get. The total number of parts is less than a fifth of that in a piston engine and less than half that in a free-piston engine. (This is the latest estimate, but it keeps going up.)

If properly tooled and produced in right quantities, there is no reason why there should be a cost penalty on the gas turbine. I will not pretend though that our program is far enough along to predict just how much turbines will cost when produced in volume. The situation is this: the engine has only a few parts, but the parts are strange and will require new and as yet undeveloped production approaches. We are under no illusions about the problem that confronts us here. Just because we are convinced that the gas turbine is mechanically and thermally superior does not mean that for these reasons alone it must supplant the piston engine. The turbine is a challenge to engineers, but it is just as much a challenge to management. If turbines are ever to be produced in quantity, radical manufacturing changes will be necessary. Planning for this requires a sound basis in fact.

In the past, many predictors have said that gas turbines will first be produced in limited quantities. However, we can't afford to build them in small quantities or the cost will be too high! When they have been tooled for mass production, we cannot produce only a few each day or we will lose millions of dollars because our market won't be large enough for us to recover our tooling cost. My prediction would be that when the gas turbine has been fully accepted by the automotive companies, it will be put into production in properly tooled plants and will be produced at an economically sound volume.

Another reason why we think gas turbines will be used in future automobiles is that the major problems of fuel consumption have been overcome. No longer do they give only 4 mpg. Engines of today are giving economy equal to that of piston engines. Furthermore, the gas turbine has no octane requirements. Properly developed, it will operate on almost any fuel that will flow through a pipe. However, one exception

is necessary: a few years ago much was said about the savings of operating with a cheap fuel, but the cheap-fuel story is myth. If we sold water at filling stations and taxed and handled it as we do gasoline, we would pay nearly ⅔ as much money for it as we do for gasoline today. The ability to operate on any reasonable fuel should help in the conservation of our natural resources, but it will not likely reduce the cost of operating a car.

We had four overall objectives when we started our gas turbine research program:

1. Obtain economy equal to that of automotive reciprocating engines.

2. Determine problems peculiar to our type of engine.

3. Study manufacturing problems.

4. Get experience operating cars of this type.

Our original work on automotive gas turbines started before World War II, but our efforts were restricted to studies of cycles and design concepts. We decided that the gas turbine would be a major postwar effort and started on it immediately after the war. However, our automotive objectives were postponed for a while in the early stages by the design and successful development of a regenerative aircraft gas turbine which we undertook for the Navy Department.

From the very start on both aircraft and automotive gas turbines, all our efforts have been concentrated on engines with exhaust heat recovery. We have tested simple-cycle engines but these might be better described as component test fixtures. We have tried many types of heat exchangers and have found many types that didn't work. Our aircraft gas turbine had a tube bundle through which the compressed gases made three passes while the tubes were surrounded by exhaust gases. We found that this type of heat exchanger was satisfactory for aircraft, but that its effectiveness for reasonable size and weight was limited to about 70%. Unfortunately, automotive heat exchangers have to be above 85% effective if we are to compete successfully with the piston engine.

There have been so many conflicting statements on the effects of regeneration that although you realize what regeneration does for the gas turbine it might be interesting to look at some of the results on which we have based our thinking. All of these curves are based on the same component efficiency where applicable. These are: turbine 85%, burner 95%, and compressor 80% and include all entrance and exit losses chargeable to these components. In addition, the results shown have been reduced by pressure losses of 3% in the case of the nonregenerative engines and a pressure loss of 6% in the case of regenerative engines. The regenerative engines have further been charged with a leakage rate of 3% of total airflow. We think that these figures are fair and reasonable and certainly possible of attainment without too much development pain.

Fig. 1 shows the effect of regeneration and pressure ratio on thermal efficiency at 1600 F nozzle inlet temperature. The lines to the left of the ordinate line represent a nonregenerative engine, and the solid curves represent the performance of a regenerative engine; both are drawn for the maximum pressure ratio shown. We think that the conclusions are quite marked. Increased pressure ratio within reasonable limits is no substitute for high regenerator effectiveness. This fact has dominated our thinking in gas-turbine design.

However, the opinion has often been advanced that a few hundred degrees gain in nozzle inlet temperature would render obsolete the necessity for regeneration. We cannot fully agree with this point, and Fig. 2 shows the effect of regeneration and cycle temperature on the thermal efficiency. These curves are all drawn at a pressure ratio of 4/1. It is to be noted that the regenerative effect is enhanced at the higher cycle temperatures. The two effects are therefore not incompatible. Higher cycle temperatures and higher regenerator effectiveness go hand-in-hand to increase thermal efficiency and point the way to future development on the gas turbine.

The theory has also been advanced that regeneration can cover up component troubles and that its effect can actually substitute for poor compressor efficiency for example. Fig. 3 shows the effect of

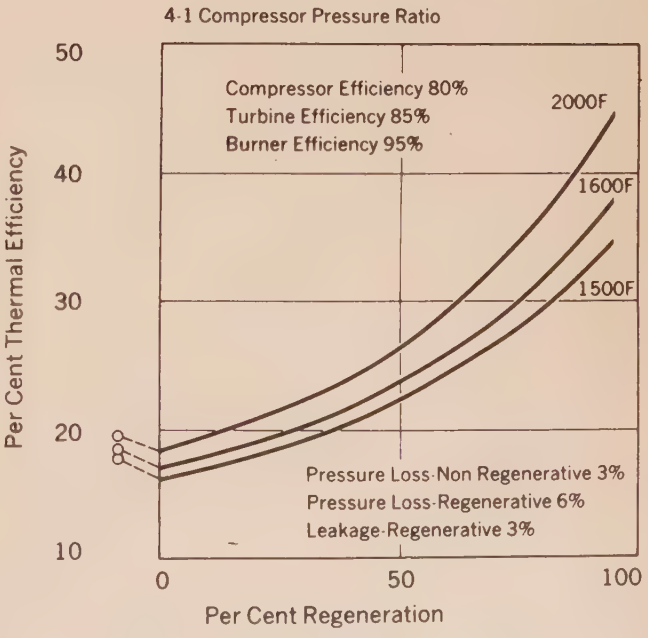

Fig. 2—Effect of regeneration and cycle temperature on thermal efficiency

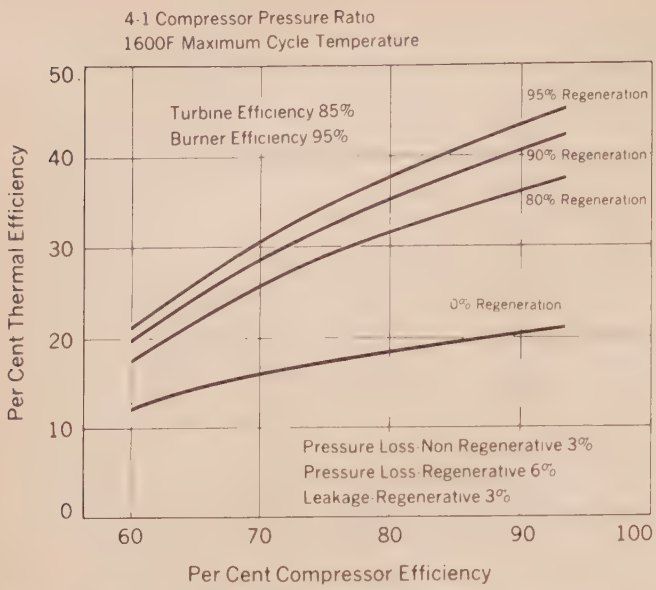

Fig. 3—Effect of compressor efficiency and regeneration on thermal efficiency

compressor efficiency and regeneration on thermal efficiency. Increased compressor efficiency is even more important with regeneration than without it as the curves show. Of course with some truth, a bad compressor could get by for a while on a regenerative engine, but regenerator size and weight would soon catch up with it and force the same attention to careful and detailed compressor development that is necessary without heat recovery. To a slightly lesser degree the same conclusion is valid for all the other components.

There were many other considerations in addition to the necessity of regeneration which controlled the configuration of our first engine. Obviously it had to go into a conventional car, or road test data would have no useful basis for comparison. Further, it had to be an engine which would allow us to make changes easily, for we never fooled ourselves into believing that the engine as originally designed would be far enough advanced to be other than the laboratory development tool which we intended it to be. Therefore, we decided that the engine design should give the greatest freedom in reworking the major components. In our first engine we also decided to experiment with the idea of using a cast housing. This would show us additional problems that might be encountered in mass production and would give us great freedom in the installation of instrumentation, even though it would substantially increase the weight.

Reasonable fuel consumption, of course, also meant a small engine. Speaking from experience, small gas turbines are much more difficult to make efficient than are large ones. A leakage path of only a few thousandths of an inch becomes a major leak when the total area of useful flow is only about 20 times as great as the leakage path. We knew that there would be many extra problems involved in small gas turbines if we wanted to develop them for automotive vehicles, but nevertheless they must be capable of being used in every size of automotive application. Therefore, we set out to do the most difficult job: design the engine for a small car and make it behave in that application.

As far as engine development is concerned, I think you will all agree that much more useful work and a larger store of data can be compiled more economically in dynamometer tests. Unfortunately, dynamometer tests would not make apparent the handling characteristics of the engine. Neither would dynamometer tests emphasize the control problems that will probably concern the driver. They tend to conceal the problem of lag on acceleration and do not take into account many types of failure that can occur in driving. For these reasons we felt that our research program had to include car operating experience.

Fig. 4 shows the result of our initial efforts. The air intake at the left of the engine is annular, and the air flows uniformly from its circumference to the compressor inducer which has two rotating axial stages integral with the cast radial compressor. The intake passages are lined with insulating material to prevent heat transfer to the incoming air. The radial diffuser and collector delivers the air at low velocity to the engine top cover which directs it downward through the regenerator to a plenum chamber communicating with the space surrounding the burner combustion sleeve. Introduced into the burner, the flow after heating is directed downward through the burner sleeve and a conductor pipe to a vortex chamber communicating to the first stage nozzle. After flowing through the two turbine stages the exhaust gas passes upward through the regenerator and is again directed downward, by the top cover, to the exhaust pipe (not shown since it is located diametrically opposite to the burner in this drawing). The housing is cast iron. Both the engine top cover and the burner cover are cast aluminum. Accessories are driven from a reduction gear mounted at the front of the engine and meshed with a pinion on the gas generator shaft. The accessory reduction gear ratio is 8.3/1, and drives a shaft which runs parallel to the gas generator shaft, driving the fuel pump, regenerator, and electric starter-generator. Final reduction of 12.5/1 is through a 2-step reduction gear at the rear end.

The regenerator is a circular built-up disk 18 in. in diameter and 3 in. thick and is currently operating with a heat-recovery effectiveness of 83% at full load and 87% at quarter load.

This effectiveness was obtained only through careful research and component development. However, it by no means represents what we consider to be the pinnacle of regenerator development. We have already exceeded this figure by three percentage points

and are working with considerable success on even higher heat recovery.

The compressor used in this engine operates at 4.25/1 pressure ratio, and measured adiabatic efficiencies of 78% have been obtained. Construction of this original compressor impeller was quite interesting. The impeller itself is an aluminum casting with the vanes fully shrouded. The inducer section consists of two rows of cast-aluminum blades loosely placed in slots in the shaft and retained radially by a steel shroud shrunk to the eye diameter of the impeller shroud. The shaft is steel and is of the built-up type with a calculated natural frequency of 77,000 rpm. A roller bearing in the front end and a ball bearing at the rear are jet-lubricated. Shaft seals may be seen directly behind the impeller and on the hub of the compressor drive turbine. Operating speed of this unit is in excess of 51,000 rpm and, therefore, components and the entire assembly are balanced to quite close tolerances. The entire assembly is made by stacking the parts from either end and is retained by a through bolt which passes down the center of the shaft, the drive of the assembly being taken by friction between the various parts.

The rear shaft is assembled similarly to the power turbine and is supported at its forward end, behind the power turbine, by a ball bearing. The rear end of the shaft is supported by the reduction gear drive pinion which is straddle-mounted in roller bearings. The ball bearing takes the turbine wheel thrust since there is no bearing thrust component in the herringbone reduction gear.

The construction described above is not one which we would recommend to anyone. The herringbone reduction gear is unfortunately noisy, and we were constantly plagued with drive pinion bearing troubles and with power-turbine shaft-alignment problems. This reduction gear has since been replaced by a single-step helical which is better in every respect.

The engine as shown in Fig. 4 is the type first operated. It was later shown publicly in March, 1954. With changes not shown in this figure, it is the engine which we used in our transcontinental run last March and is the same engine which powers the car which I drive.

During the entire development period substantial progress was made on this engine. Component efficiencies were increased, many of the mechanical problems were solved, and sufficient car operating experience was obtained to point out in a very practical way the deficiencies and the advantages of a gas turbine in a passenger car.

The basic advantages of the turbine in this application have already been discussed, but we recognize

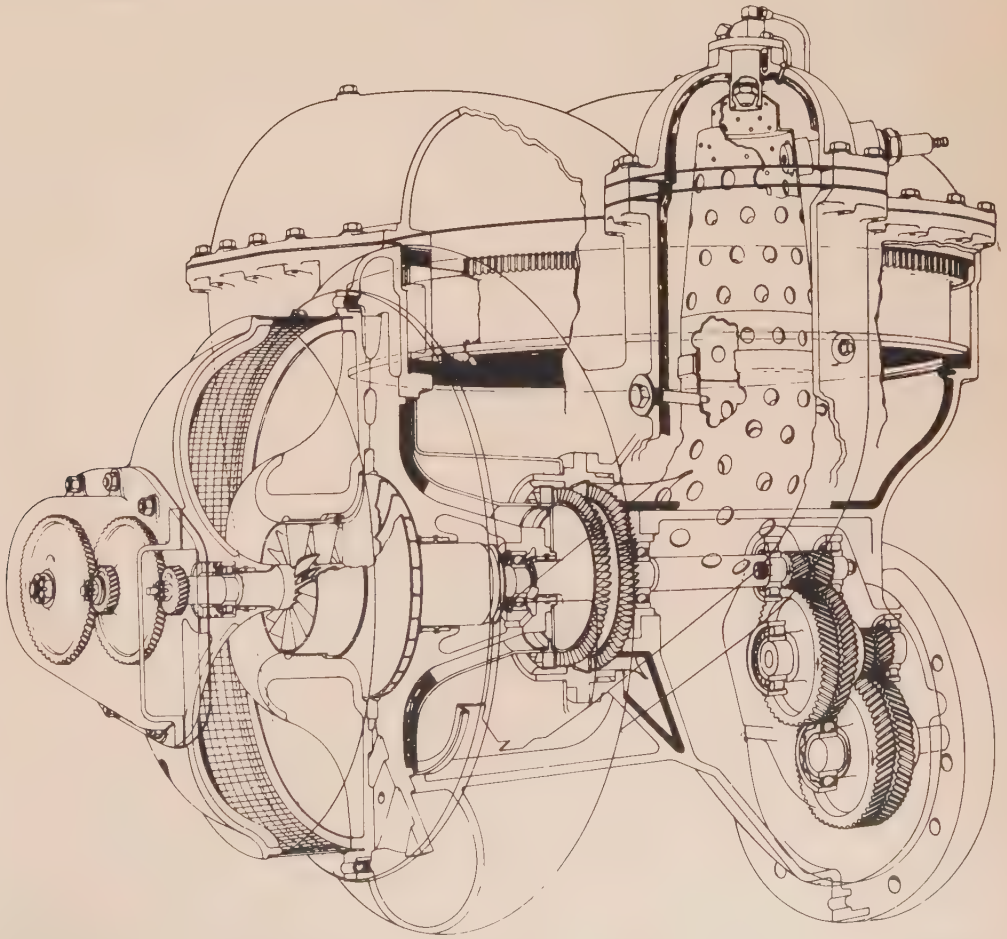

Fig. 4—Schematic diagram of initial engine design

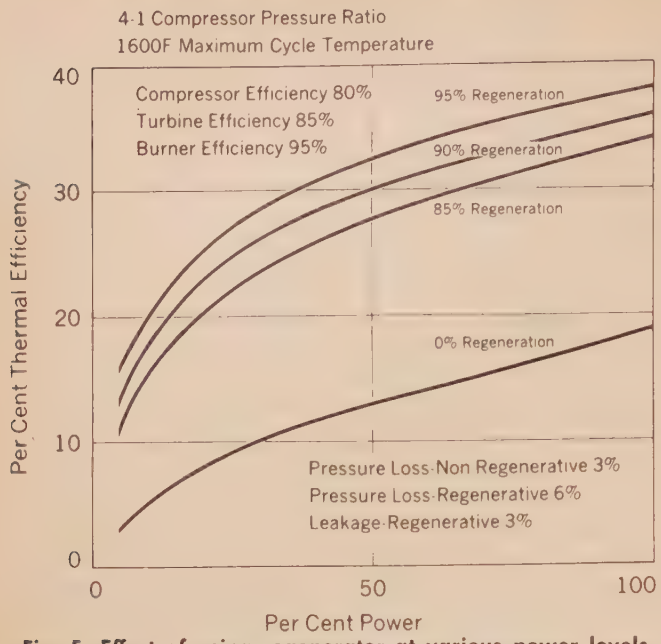

Fig. 5—Effect of using regenerator at various power levels

that the turbine also has certain characteristics which are either unlike the reciprocating engine or inferior to it. Our continued interest in the turbine, however, is indicative of the confidence we have that these problems can and will be solved and that the lead established by 60 years of reciprocating engine progress can be overcome. The two deficiencies most usually associated with the automotive application of the turbine are acceleration lag and lack of engine braking. We have made substantial progress on both of them, and if we continue on the same line of development that has been effective so far we see no reason why our solution to these problems should not be perfectly satisfactory. We do not know what other people are doing in this direction and can only assume that they have made equal or perhaps greater progress than we have.

In the latter half of 1955, we began endurance testing on this engine in earnest. Our experience is not much different from that experienced with any highly experimental piece of machinery. It took us several weeks to just get off the ground, but once past this initial point, endurance test hours began to build up. Today endurance testing is getting to be a problem at Chrysler. Tying up a test cell for hundreds of hours has necessitated the building of special endurance setups in order to keep our test cells free for development work.

When our confidence in the engine had increased to a satisfying degree we started endurance runs with the turbine-powered Plymouth. Out of this was born the idea of a transcontinental run. From the endurance standpoint we had little doubt of our ability to make a successful crossing since we had behind us severe Proving Ground test runs of considerably greater length. Car operation had run the gamut of traffic, high speed, hill climbing, and long periods of idling but we felt that we had much to learn from the effects of road dust, widely fluctuating ambient temperatures, altitude, and general abuse. In addition, the specter of a schedule to be maintained at all costs applied a continuous spur to our efforts.

The trip was marred by two incidents. The failure of a reduction gear bearing due to loss of oil caused by cracking of an oil line in the reduction-gear housing, a design long since abandoned along with its reduction gear in favor of a new single-stage helical box; and the fatigue failure of a casting in the intake environment. Neither of these should have occurred, and both have since been corrected. In both cases, however, the repairs were easily effected on the spot without otherwise altering the engine, and the test continued.

A few weeks ago an expert offered the opinion that the regenerator was bypassed on the transcontinental run because it was not needed at the power level being used. I can assure you that this was not the case. We ran the whole way with the regenerator in full operation all the time. Fig. 5 shows the effect of a regenerator at various power levels. The regenerator has a marked effect on efficiency, even at full power output.

Overall, economy on the transcontinental trip ran between 13 and 14 mpg. This includes the frequent and extended periods of idling which occurred because people wanted to look at the car. The engine was not shut off from New York to San Bernardino, where we got a few hours sleep, except for the two incidents mentioned above.

From the standpoint of the results obtained on the transcontinental test we were well satisfied with the performance of the engine on this first turbine-powered transcontinental road test. We have had more trouble with newly designed reciprocating engines on their first cross-country runs, and the test results and the experience in turbine operation which we obtained were well worth the trouble.

Since our objective is a successful automotive powerplant, much of the work which we have done in the past 2½ years has been directed toward achieving the same degree of reliability of the automotive gas turbine as we have grown to expect in the automotive reciprocating engine. The major qualifying consideration in this work has been price. Cost reduction has been achieved throughout much of the engine. We have found that we are able to take much of the cost out of our components by intensive effort. One example of cost reduction is the improvement in turbine-wheel fabricating methods. Many of you are familiar with the costly way in which aircraft turbine wheels are manufactured. Each blade is individually machined, weighed, and selectively fitted to the hub of the wheel. We felt that we could have no part of such a manufacturing technique and, therefore, there was no reason to attempt to build such a wheel. In addition, our turbine blades are too small to be as-

sembled on an individual basis. Aircraft turbine blades are frequently about 3 in. long; ours are about the size of a thumbnail. Even the turbine wheel in our first engine was made by a much simpler technique. At present all 53 blades are cast in one operation, and no further machining is done on them other than removing a riser on the tip of each blade. Several techniques have been used in assembling the wheel itself, but the one with which we have been most satisfied is welding the cast ring of blades to a lower alloy hub and then machining. Balancing is then done on the assembled wheel in a single operation.

Materials used in our early engines were the best that could be obtained. Aircraft quality was maintained throughout, and some of these alloys are still being used as the occasion arises in our present work. However, production in automotive quantities is simply just not possible with aircraft-type superalloys if we use the present world supply of alloy elements as a base. The use of these alloys has allowed us to do the mechanical and thermodynamic job that had to be done, but at the same time work has been going forward to obtain materials which could be made from elements readily available and noncritical in nature.

Progress in this activity has been fully as substantial as that made in the mechanical and gas dynamic problems. Although high-temperature alloy work of this type is not done overnight, our progress to date has indicated that current operating temperatures can be maintained for satisfactory operating life without the use of any cobalt and with nickel contents low enough to be well within a practical range from the standpoint of availability.

In the range of materials not requiring superalloys fully as startling results have been obtained. One example is in the case of our burner liner. The best material we have found yet is 1010 steel aluminized, and on our engine at least it was found to be much superior to stainless steel in this application.

Because the field of automotive gas turbines is so new, we have had to develop test equipment to go along with our engine program. You have never seen an engine so completely instrumented as these. Even in cars, we are able to measure temperature at some 60 different points in the engine and, through a switching system, read the temperatures on the instrument panel of the car. In our dynamometer testing we have facilities for measuring over 120 different temperatures on a single engine. And we have banks and banks of manometers to measure pressures at every conceivable point of the engine. Special test fixtures have been built for testing heat exchangers, burners, turbine stage wheels, fuel nozzles, and in fact every part of the engine.

We could not come as far as we have without the use of today's modern digital computers for the necessary data reduction. For that matter there probably wouldn't be any data to reduce if we weren't able to use the computer for our design studies and stress analysis work.

What I have said so far satisfies the requirements for the factual, interim progress report. From this point on everything is in the realm of careful, thoughtful prediction. We can close that progress report with the two curves shown in Fig.6. The ordinate is brake specific fuel consumption and the abscissa is per cent power. The top curve shows the specific fuel consumption, with gasoline, of our original engine. The lower curve represents the fuel consumption of the engine with which we are now working, the engine which has resulted from what we learned on the first one.

Fig. 7 shows this new turbine engine compared to

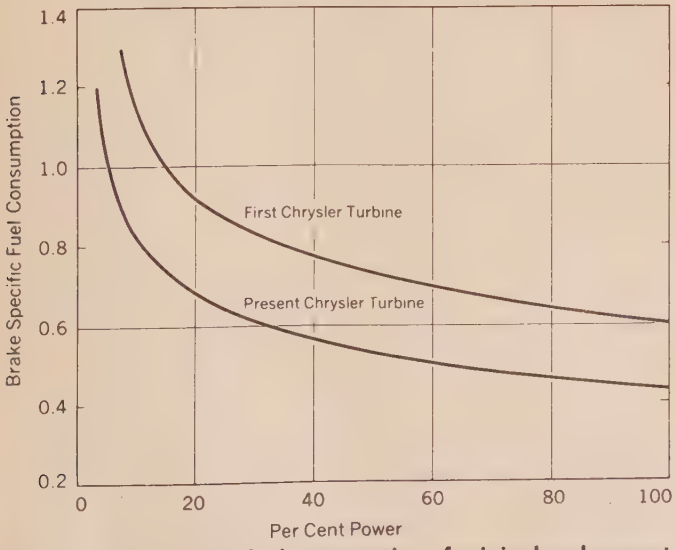

Fig. 6—Comparison of fuel consumption of original and present engine

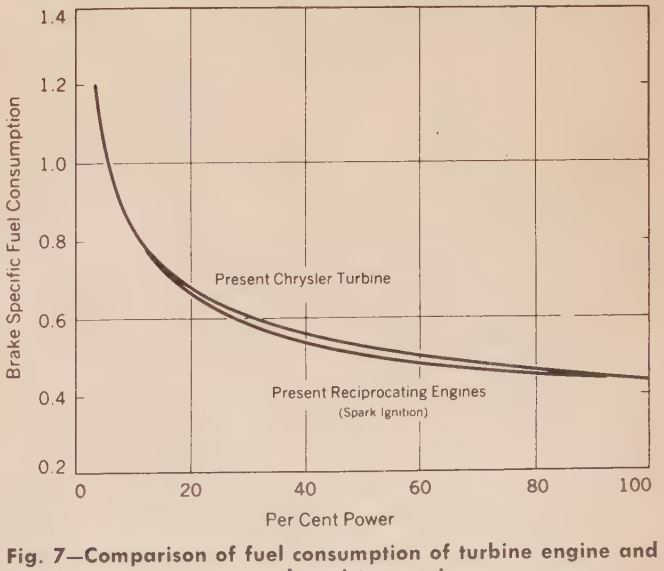

Fig. 7—Comparison of fuel consumption of turbine engine and present-day piston engine

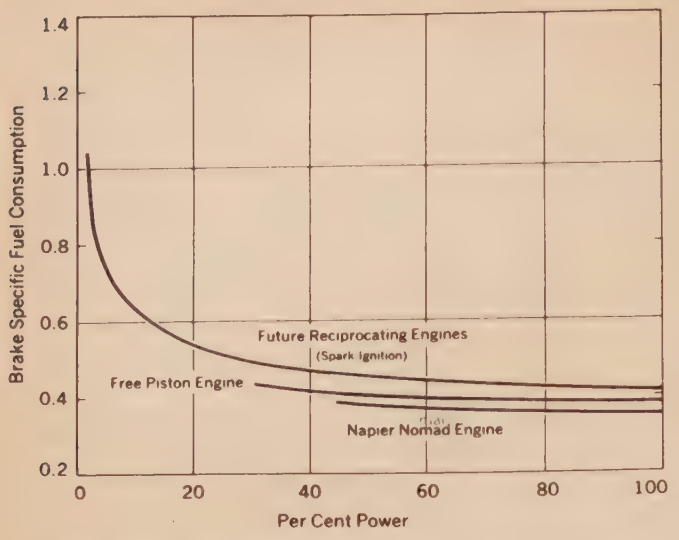

Fig. 8—Fuel consumption comparison for three engines

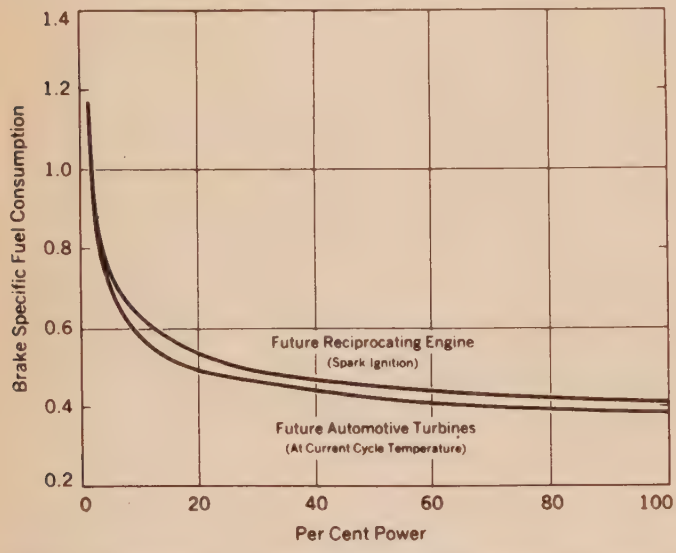

Fig. 9—Comparison of future turbine and reciprocating engine fuel consumption

a modern automotive piston engine. Not good enough yet but getting close. Fig. 8 shows three very interesting fuel consumption curves. The lowest curve is plotted from data released on the Napier Nomad compound engine. As they say in Oklahoma, "they've gone about as fur as they can go," in compounding an engine, and I don't think that we want it in an automobile. The curve just above it is the estimated fuel consumption of a free-piston compressor and turbine combination. We think this is a reasonable estimate although we may be giving the free-piston engine a little more of a break than it deserves from the time standpoint. The top curve shows a generous prediction for an advanced automotive reciprocating engine of not precisely today's conventional type.

A progress report would stop here, but engineering does not, and all of our professional lives we have all been working in and gambling with the future. If our faith in the gas turbine were confined to the engine with which we are now working and if we could only hope to equal or slightly exceed the efficiency of the automotive reciprocating engine, there would be merely the incentive of novelty to spur us toward greater efforts on the turbine. But we have a much greater incentive than this. Fig. 8 shows our concept of the fuel consumption of an automotive piston engine in the not too distant future, and we will be very happy if we achieve this degree of efficiency. Plotted with it is our concept of the automotive gas turbine of the not too distant future. We are working with the components of this gas turbine today, and our objectives call for approximately 85% compressor efficiency and 92% regenerator effectiveness. The compressor efficiency we have already achieved in past programs, the regenerator effectiveness we believe we know how to achieve. It now remains to repeat it in the package size and cost that makes it reasonable for automotive use.

The thermal efficiency shown for this turbine is obtained at cycle temperatures readily obtainable with today's materials. We hesitate to guess at what the near future will bring forth in high-temperature materials, but there are materials available to us today that will allow operation at 200 deg higher than the nozzle inlet temperature shown here. The turbine curve in Fig. 9 can therefore be taken as showing presently available techniques and current cycle temperatures. From this point the turbine really takes off, and with future increases in cycle temperature which are believed possible, leaves the reciprocating engine and the free-piston-gas-turbine combination far in the background, since they are already using nearly the peak cycle temperatures which will ever be available to them.

For every 100 F increase in nozzle inlet temperature over this curve we can expect an increase in specific output of about 10% and in thermal efficiency of about 5%. We do not think that even this represents the end of the line for automotive gas turbines. With confidence we look forward from this point, not as a goal but as a starting point for future automotive powerplant development.

Most thought-provoking is the realization that the automotive gas turbine, although still in its infancy, has its greatest development possibilities yet to come. These possibilities can only be realized by the same inspired and earnest scientific effort that has marked over a half century of reciprocating engine development. It can only be done, to repeat, by many men working in many groups and in many fields. The opportunities are there, and it remains for us and for many others not yet aware of their role in the engineering and production of future automotive powerplants to achieve the ultimate of which this versatile machine is capable.

WHERE DOES ALL THE POWER GO?

Carl E. Burke, Larry H. Nagler
American Motors Corp.

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General Motors Corp.

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Chrysler Corp.

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This symposium was presented at the SAE Summer Meeting, Atlantic City, June 6, 1956.

The Engine . . . The Power Source

by

Carl E. Burke and Larry H. Nagler

A DISCUSSION today involving the modern automobiles is seldom complete without some reference to maximum horsepower. We may also say, without danger of contradiction, that the horsepower in question will probably appear, at first glance, to be more than ample to do the job. The very logical question then presents itself—"where does all the power go?"

The writers of the five papers comprising this symposium will endeavor to answer this question. The papers are, in a sense, complete in themselves, yet they are distinctly interrelated and as a whole give a combined story of the use and expenditure of the power designed into the engine of the automobile.

The logical beginning for this symposium is undoubtedly a study and analysis of the power source. The engine is unquestionably the origin of the mechanical power to be analyzed in these papers, yet we feel that we should not proceed into that phase of the discussion without first examining the real origin of the power—the fuel.

Many fuels have been tried or proposed in motor vehicles—kerosene, naphtha, liquefied petroleum gas, fuel oils, benzine, and alcohol being among the better known. Gasoline has proved itself best considering performance in an engine, cost, ease and safety of handling, and availability in huge quantities required by today's automobiles.

The automobile engine is primarily a mechanism for converting chemical energy into useful mechanical power. The engine obtains chemical energy from

AS a basis for the analyses of this symposium, a hypothetical car has been used to evaluate the engine power distribution in performance.

Effects of fuel, engine accessories, and certain car accessories are evaluated. The role of the transmission in making engine power useful at normal car speeds is also discussed.

Variables encountered in wind and rolling resistance determinations are reevaluated by improved test techniques.

Net horsepower of the car in terms of acceleration, passing ability, and grade capability are also summarized.

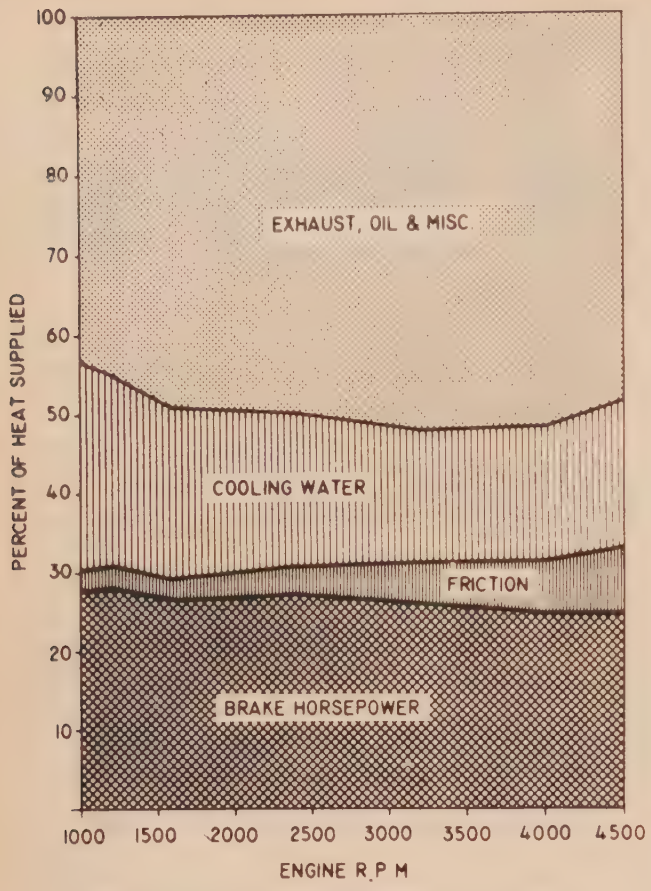


Fig. 1—Heat balance at wide-open throttle

the fuel, which is fundamentally a source of heat. Heat is measured in British thermal units (Btu) which is defined as the quantity of heat required to raise 1 lb of water 1 F. Each gallon of gasoline contains around 114,000 Btu, and each pound approximately 19,000.[1]

Heating value of gasoline can be directly converted to useful energy or work. Each Btu of heat energy theoretically can be considered the equivalent of 778 ft-lb of work. Therefore, 1 gal of gasoline is capable of developing about 89,000,000 ft-lb of work, which is equivalent to 2700 hp for 1 min, or 45 hp for 1 hr.[1] This energy can be visualized as a force of about 8½ tons against a movable object (like a railroad train) for one mile, or lifting an aircraft carrier (45,000-ton class) upward 1 ft.

Despite the tremendous potential energy of gasoline, only a small fraction is effectively utilized in an automobile engine. Most of the fuel energy is wasted because of the inherent nature of the complex process of converting chemical energy first into heat energy and gas pressures, next into mechanical rotation, and finally into forces to push the car. The basic characteristics and natural laws of these changes are an intriguing subject. A careful analysis of the problem will enable us to use the available power more effectively.

Fig. 1 shows a heat-balance chart of the modern automobile engine at wide-open throttle illustrating that, at best, a recovery about ¼ of the available heat energy of the gasoline burned. Fig. 2 is a similar chart indicating that at cruising speeds, the results are not quite as good as at wide-open throttle.

What is the nature of the fuel, gasoline? Gasoline is prepared from the world's second most plentiful liquid (petroleum, or crude oil). This is an important but frequently overlooked factor favoring its universal popularity as a motor fuel. Tremendous amounts of energy are packed into the molecules of which gasoline is composed. These molecules are almost entirely combinations of hydrogen and carbon, in various proportions. Hydrogen and carbon in themselves are excellent fuels. They combine readily with oxygen to form, respectively, water and carbon dioxide with the subsequent release of comparatively large quantities of heat energy. For example, 1 lb of hydrogen when mixed with oxygen or air releases 52,000 Btu of heat energy; each pound of carbon releases 14,100 Btu.

A typical gasoline is composed of approximately 15% hydrogen and 85% carbon—combined, as stated previously, in slightly varying proportions. There are thousands of individual hydrocarbon compounds making up any particular gasoline. Each of the compounds contributes an individual characteristic as to heat energy, density, antiknock value, stability, volatility, and other overall characteristics of the final product. In general, the higher the ratio of hydrogen to carbon in a particular chemical compound composing gasoline, the more heat will be released. However, the hydrogen-carbon range of all of the chemical compounds present in appreciable quantities is comparatively limited.

Four main types of hydrocarbon compounds are to be found in gasolines. These are identified by chemists as paraffins, naphthenes, aromatics, and olefins. In addition, refiners add small quantities of substances such as antiknock fluid (tel plus scavengers), inhibitors to prevent corrosion and gum-forming tendencies, de-icer compounds, and the like. The characteristics of the four major types of hydrocarbons are briefly described in the Appendix.

Gasoline when mixed with air is one of the most powerful explosives, pound for pound, known today. Other explosives may be more violent because they burn at more rapid rates; however, as a means of producing controlled heat energy gasoline has few equals. It is commonly considered that some of the more violent explosives, such as TNT or dynamite, are more powerful than gasoline, but actually they have a very low heating value per pound. Assuming a car can obtain 24 mpg of gasoline, it would run

[1] Values are considered approximate, since gasoline is not a fixed chemical substance, but varies slightly in its chemical composition and density.

only four miles on a gallon of dynamite. The explanation basically is that the gasoline is concentrated fuel (only carbon and hydrogen). Oxygen from air is used to complete the explosive mixture. Such explosives such as TNT and dynamite carry their own oxygen to burn their carbon and hydrogen. This oxygen is combined chemically with other elements in a rather expensive form. Oxygen from air is free and does not have to be purchased by the motorist when he fills the fuel tank.

Nevertheless, as many stock-car racers and big-time race drivers already know, it is possible to add certain elements and compounds to gasoline to produce more power in an engine. In general, the additive compounds contain added oxygen which supplements the oxygen from the air in burning comparatively larger quantities of fuel in a specific engine, and thereby producing more power. However, when compared with the cost of the gasoline, the added compounds are relatively expensive.

Modern gasolines have many important characteristics, but probably one of the most significant is octane rating. Octane rating is simply the measure of a fuel's ability to operate at high compression ratios, high compression pressures, and/or high temperatures without resulting in uncontrolled combustion known as detonation or fuel knock. These disturbances are ordinarily associated with uncontrolled burning of the mixture ahead of the flame front which was initiated by the timed ignition spark.

Fig. 3 shows the rise in fuel octane rating over the past 30 years. The improved fuels have had a tremendous effect on engine design—it has made possible the use of higher compression ratios and more efficient engines.

The subject "power source" for this paper, the typical 1956 engine, with over 50 years of refinements, designing, and work by thousands of engineers is opinioned to be a very efficient mechanism for the conversion of the chemical energy of the gasoline into mechanical horsepower. A general description of the engine is shown in Table 1.

The typical V-8 engine described is the basis for investigation of the question, "Where does all the

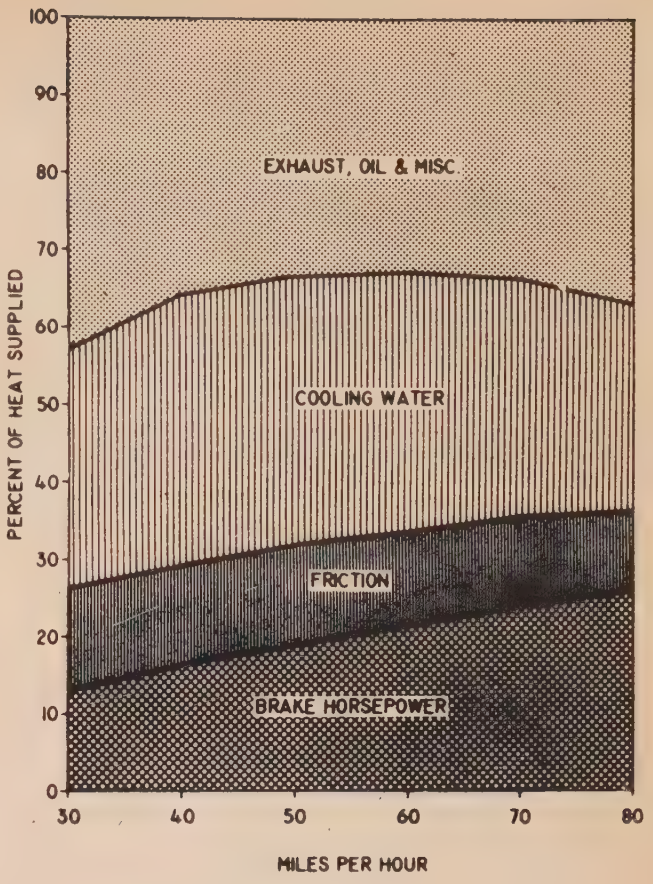


Fig. 2—Heat balance at cruising speed

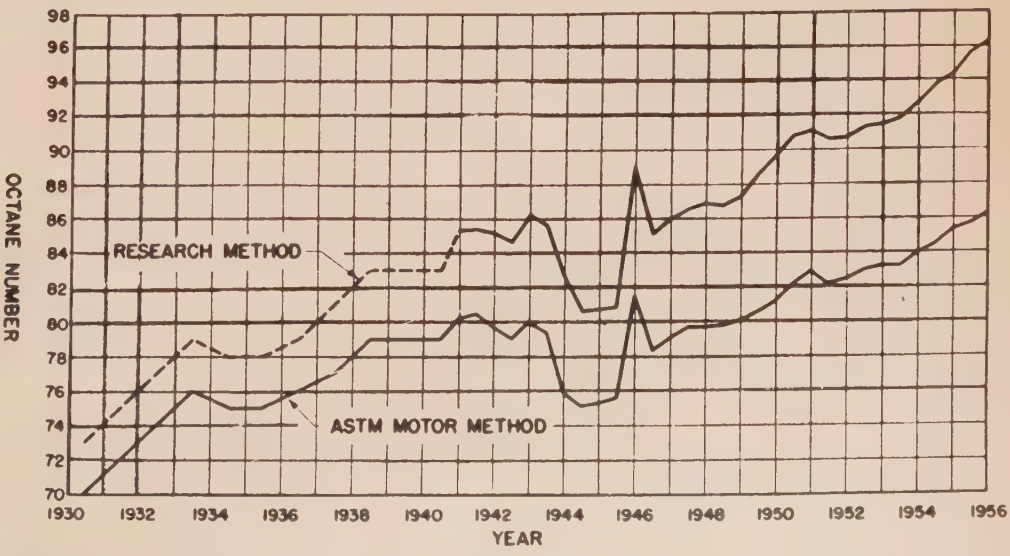


Fig. 3—Trend in antiknock quality of premium gasolines sold in United States

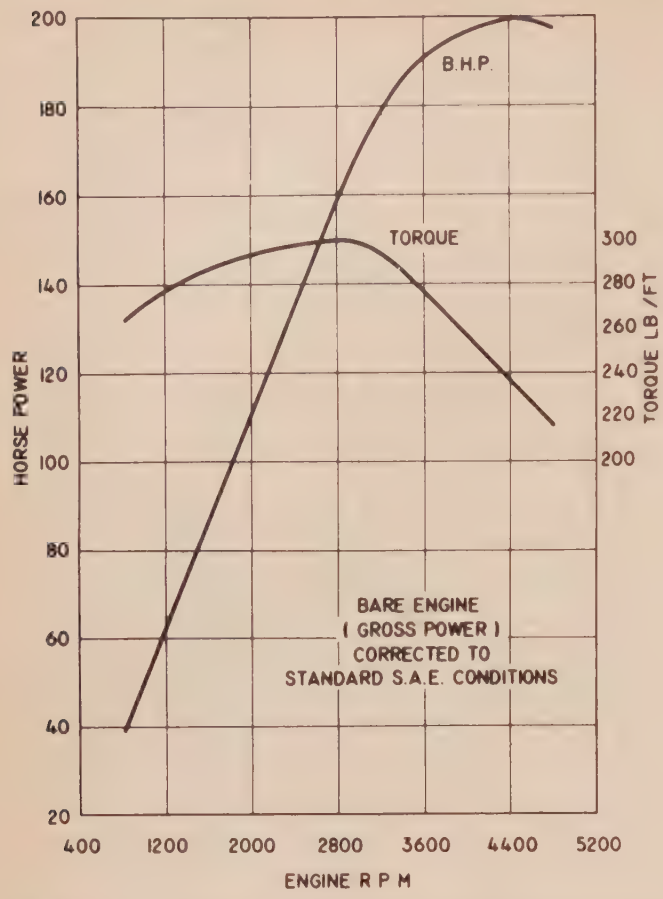


Fig. 4—Power characteristics at wide-open throttle for hypothetical engine

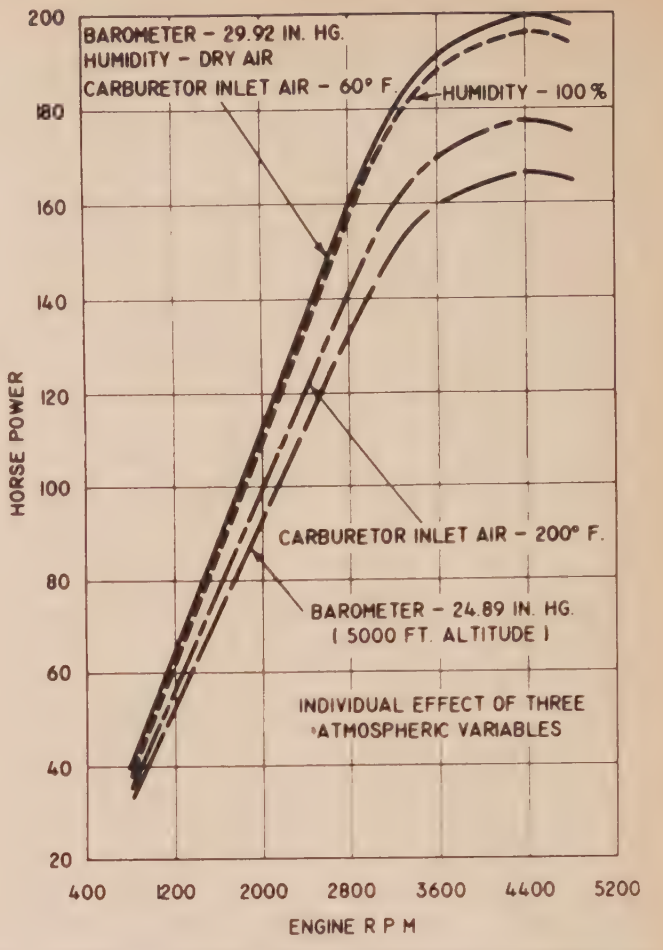


Fig. 5—Effect on engine power of atmospheric conditions for hypothetical engine

Table 1—Engine Description

Type	90 deg, V-8, Overhead Valves
Total Displacement	300 cu in.
Stroke to Bore Ratio	9 to 1
Compression Ratio	8.5 to 1
Fuel Octane Requirement	93 (Research)
Maximum bmep	150 psi
Maximum hp per cu in.	67
Engine Weight	600 lb
Gross Horsepower	200

Table 2—Standard Conditions

Barometer	29.92 in. of Hg
Vapor Pressure	0 (Dry Air)
Air Temperature	60 F

power go?" Fig. 4 shows the wide-open-throttle power characteristics of the hypothetical engine. This, in the engine test field, is referred to as the bare engine power or gross power and is the starting point, from the mechanical horsepower standpoint, of this paper. The conditions of this test are generally as follows:

1. No air cleaner.
2. No generator.
3. No engine fan.
4. Spark and fuel adjusted to mean best power at each speed (every 200 rpm).
5. No exhaust heat to intake manifold.
6. Exhaust system is dual piping from exhaust manifold to evacuated laboratory exhaust system with back pressure not less than atmospheric or more than 1 in. of Hg pressure.
7. The water pump and the fuel pump of the engine are included in the bare engine horsepower.

The bare engine power is corrected for the atmospheric variables of barometer, vapor pressure, and ambient temperature in accordance to procedure set up by the SAE. The set of standard conditions are given in Table 2.

The basic formula to obtain the correction factor as developed from a modification of Charles Law of Gases, $\frac{PV}{T} = \text{Constant}$, is shown in Table 3.

The use of a correction factor in dynamometer test work is an absolute must due to variations in atmospheric conditions that affect power output of the engines. With the ordinary control of carburetor inlet air temperature in a dynamometer room to a maximum spread, for example, of 20 F the normal day-to-day variations in humidity and barometric pressure can in total produce daily variations in engine output of 10%. This variation in power output in the dynamometer room where atmospheric conditions are controlled to a certain extent, brings up the question as

to what happens in the car. Underhood air temperatures in the car rise to 200 F under certain conditions of operation. Barometric pressure, throughout the country, will vary from 30 to, for example, 24.90 in. of Hg in higher altitudes. Humidity, of course, is a variable in all parts of the country. Fig. 5 shows the effects of the variables mentioned on engine power output. The effects are shown individually; they are not plotted to show their possible cumulative effect.

To review, Fig. 4 represents the corrected, bare engine, gross horsepower. Prior to installation in the automobile, a group of components and calibrated parts are added, thereby contributing certain power losses in accomplishing each necessary function. The power losses to installed engine components and conditions are shown in Table 4.

Fig. 6 shows the installed engine power corrected to the standard SAE conditions. This curve includes the losses shown in Table 4. In addition, on Fig. 6, the installed engine power adjusted to a set of atmospheric conditions which have been found to be an average for the Detroit, Mich., area. These conditions are barometer 29.40 in. of Hg, vapor pressure 0.40 in. of Hg, carburetor inlet air 100 F.

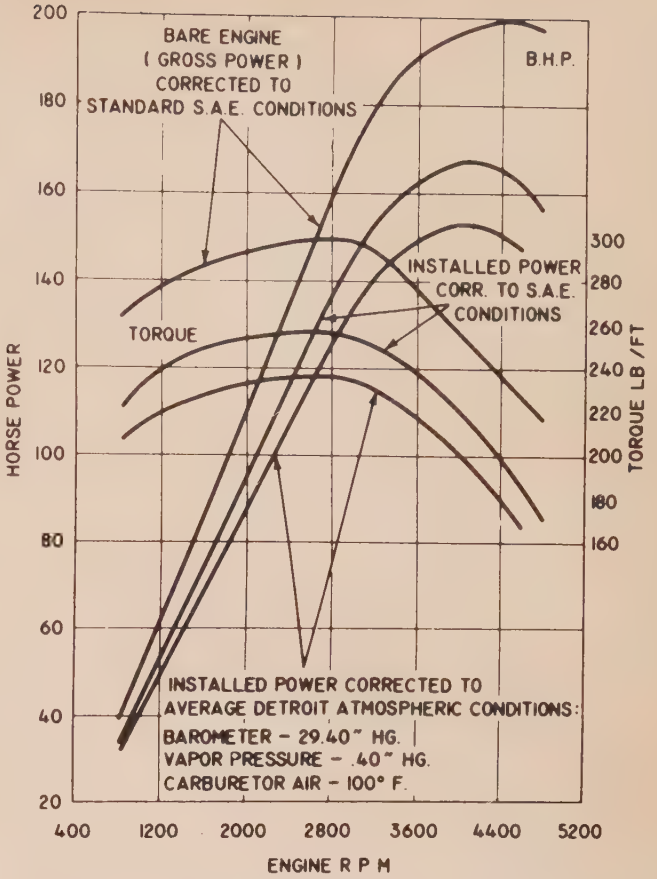


Fig. 6—Power characteristics at wide-open throttle for hypothetical engine

APPENDIX

Paraffins ($C_n H_{2n+2}$)

The paraffins are present in gasoline in relatively large quantities. These are generally classified by the chemist as "straight chain" compounds. They contain the highest heat energy of all the hydrocarbons, primarily because the paraffins have a slightly greater percentage of hydrogen. Some fuels run as high as 60% of paraffin compounds. Individual chemical compounds of this series include triptane (C_7H_{16}) which is a hydrocarbon having an exceptionally high antiknock characteristic, and heptane (also C_7H_{16}) but arranged differently) which has an exceptionally poor antiknock characteristic. Pennsylvania crudes are relatively high in paraffins.

Naphthenes ($C_n H_{2n}$ and $C_n H_{2n-2}$)

The naphthenes are also known as cyclic paraffins. The heat content is somewhat lower than for the paraffins, and they differ from both paraffins and olefins as to antiknock quality and stability. Crude oils from California and parts of Texas and Russia are relatively high in naphthenes—hence the term naphthenic crudes.

Aromatics ($C_n H_{2n-6}$)

The aromatics have slightly less hydrogen in their make-up than the paraffins. Usually the aromatics make up less than 20% by volume of a typical gasoline. One of the best known aromatics is benzol (ben-

Table 3—Engine Power Correction Factor

Corrected bhp = [(Observed bhp + fhp) × Correction Factor] − fhp

$$\text{Correction Factor} = \frac{29.92}{B-E} \times \sqrt{\frac{460+t}{520}}$$

where:
B = Observed Barometer
E = Vapor Pressure (from humidity), in. of Hg
t = Intake Air Temperature F
The above corrects to a standard barometer of 29.92 in. of Hg, dry air at 60 F (520 R)

Table 4—Power Losses to Installed Engine Components and Conditions

Muffler System	
Dual Muffler	7% at 4400 rpm wide-open throttle, diminishing down speed range
Single Muffler	14% at 4400 rpm, diminishing down speed range
Air Cleaner	3% at 4400 rpm wide-open throttle, diminishing down speed range
Combustion-Chamber Deposits	4% over entire speed range
Manifold Heat	Maximum 5% to 1600 rpm, diminishing to 1% at 4400 rpm
Automatic Distributor	6% at 800 rpm 4% at 1600 rpm 2% at 2400 rpm 1% at 4400 rpm
Total Loss	16% at 4400 rpm

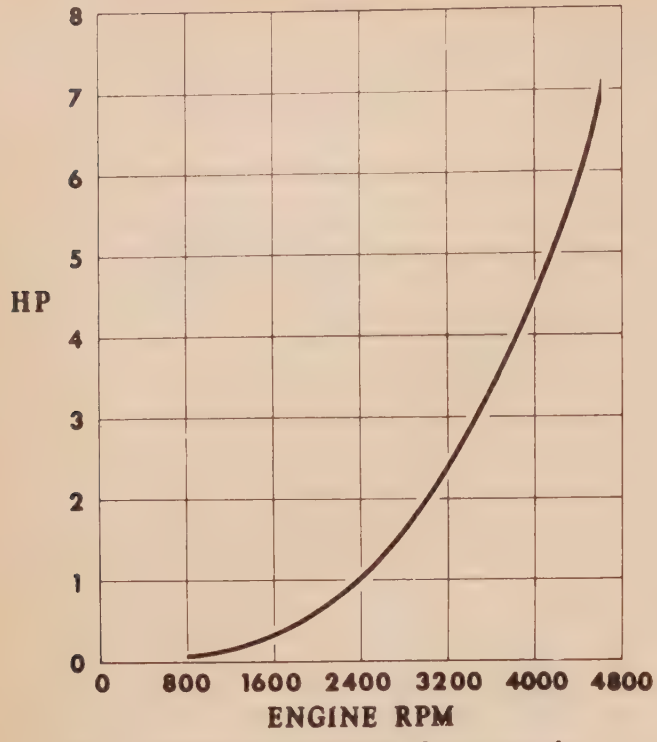


Fig. 7—Horsepower requirement for average fan

zine) which is ordinarily produced as a byproduct from manufactured gas. (It can also be obtained directly from petroleum.) Toluol (toluene), xylenes, and cumenes are also aromatics.

Olefins ($C_n H_{2n}$)

The olefins are found in motor gasoline to some extent, although they do not occur naturally in crude oil—olefins result from the refining processes. Olefins tend to be less stable than paraffins—they have a somewhat greater tendency to react with oxygen, other compounds, and other olefins. Some of the olefins have excellent antiknock qualities.

The Accessories—The First Bite

by

E. C. Campbell

THE basic engine and its inherent power characteristics have been well covered, and we have a pretty good idea what to expect the engine crankshaft to deliver in the way of usable power. Before installing a transmission and putting this power package in an automobile, consideration must first be given to the power going out the front end of the crankshaft to the engine-driven accessories which are now common to most automobiles. For the purpose of our hypothetical engine we will consider only the fan, generator, and power steering pump. The first two have been required on cars for years, and the power steering pump is rapidly becoming standard because of customer demand. Our hypothetical engine will have to supply the power to drive these accessories.

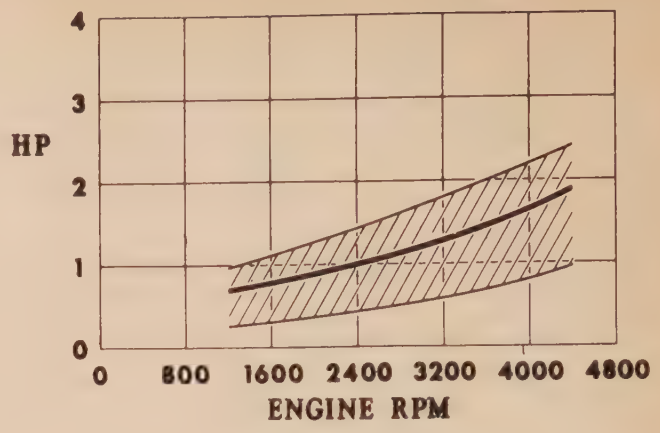


Fig. 8—Power requirement of generator

Fan

The fan is the most demanding of these accessories and shall be considered first. The fan is basically part of the engine since its primary purpose is engine cooling. This is accomplished by pulling air through the radiator and then blowing it back over the engine, exhaust pipes, and the like. Years ago most engines were rated with the fan installed on the engine so that it was not necessary to run separate fan tests and then deduct the power requirement from the engine output. Engine rating is primarily a dynamometer laboratory procedure, and the whirling fan presented a very definite safety hazard to personnel conducting engine tests and also presented a serious noise problem. For this reason most engines are now rated with the fan removed and the fan tested separately in free air on a small accessory dynamometer.

There are several factors affecting the fan power requirements. Among these are grill styling, heat rejection of the engine, radiator efficiency, and last but not least, the efficiency of the fan itself. Fig. 7 shows the horsepower requirement of an average fan for the size engine under discussion here. The curve indicates that the faster the fan turns the greater the increase in power requirement becomes; so that up in the speed range of the horsepower peak it is eating up considerable power, requiring over eight times as much power as it does at 2000 rpm, or approximately 50 mph.

Generator

Our next accessory is the generator, a device of varying appetite which converts energy according to

the loads other equipment is demanding of it. The generator supplies the electrical requirement of the entire automobile, serving to keep the battery charged, operate the ignition system, lights, radio, heater and defroster fans, and other miscellaneous accessories the owner might decide to install. Typical of rotating electrical apparatus, it has its own blower equipment which adds to the load. Fig. 8 shows the load range within which the generator has to operate, with the load being automatically controlled by the voltage regulator. For purposes of our hypothetical engine we have selected about ⅔ of its peak requirement for an average power diversion. This is the heavy line appearing in the center of the band.

Power Steering Pump

The third accessory drawing power from the engine ahead of the flywheel is the power steering pump. This is another accessory which draws on the engine in proportion to the load placed on it. It can idle along taking care of the friction losses of the system it serves or it can actually go to work. To operate the power steering gear, the pump is required to provide pressure as high as 700 to 800 psi instantaneously. However, these loads are intermittent and generally occur at very low engine speeds while parking or turning corners, and the engine will only be charged with the pumping losses of the entire system. Fig. 9 shows the horsepower requirement of the power steering pump.

Now that the individual power requirements of the basic accessories are known, we can total them up and see what they do to the net horsepower available at the flywheel. Looking at the engine power and torque graph (Fig. 10); the lower curve is the net horsepower available at the flywheel. At 50 mph the accessory demand is relatively low, in the order of only 2 hp, but up in the speed range of the horsepower peak the demand is high, dropping the peak horsepower available at the flywheel to 145 hp. It is now noted that although the engine has a rating of 200 hp, more than 25% of the gross engine power has been used up in the area ahead of the flywheel. It must also be remembered that so far only the average operating conditions of standard accessories have been considered.

Air Conditioning

It is essential to consider another accessory which is becoming very popular in the warmer climates, and that is air conditioning. Since this is a system which appreciably affects engine power, it must be

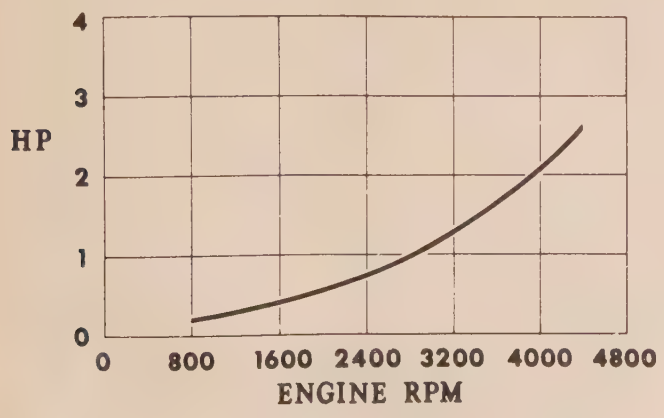


Fig. 9—Power requirement of power steering pump

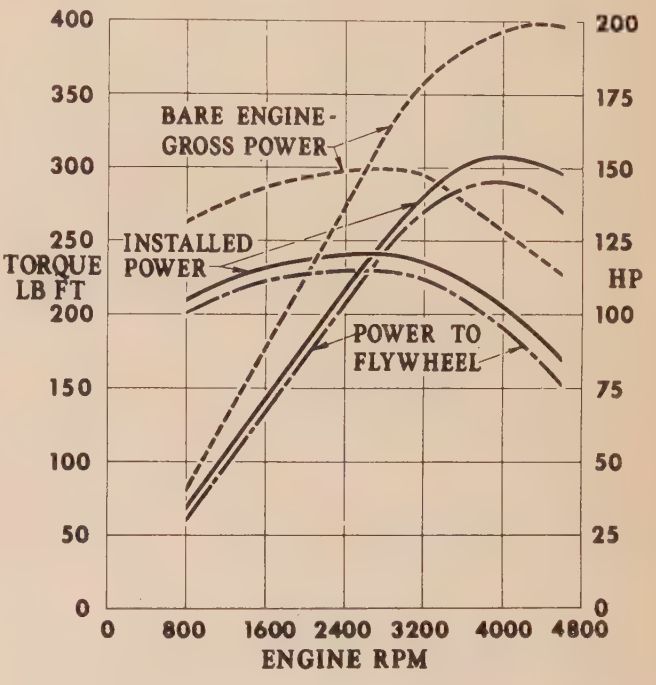


Fig. 10—Net power to flywheel under operating conditions

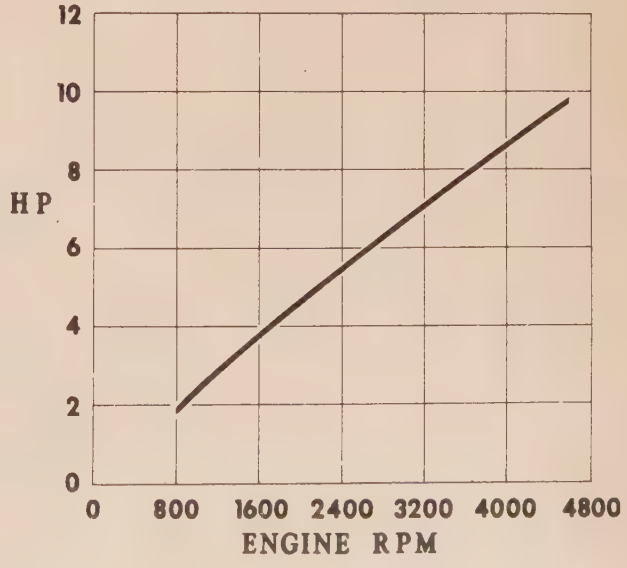


Fig. 11—Power requirement of air conditioning compressor

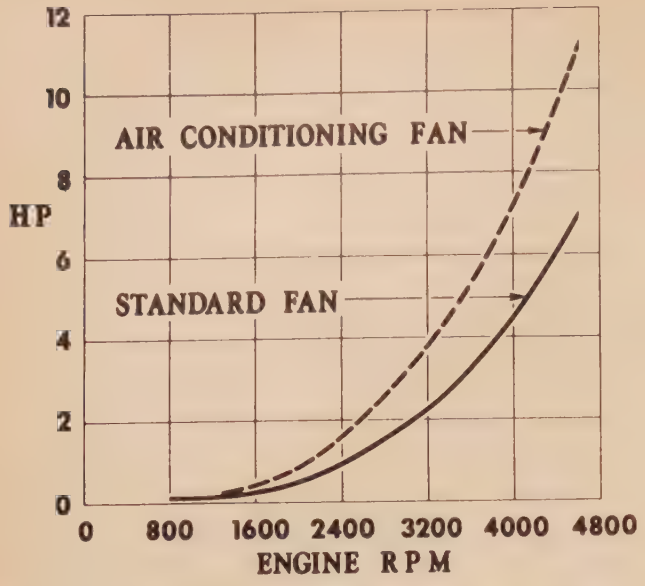


Fig. 12—Comparison of power requirements of air conditioning fan and standard fan

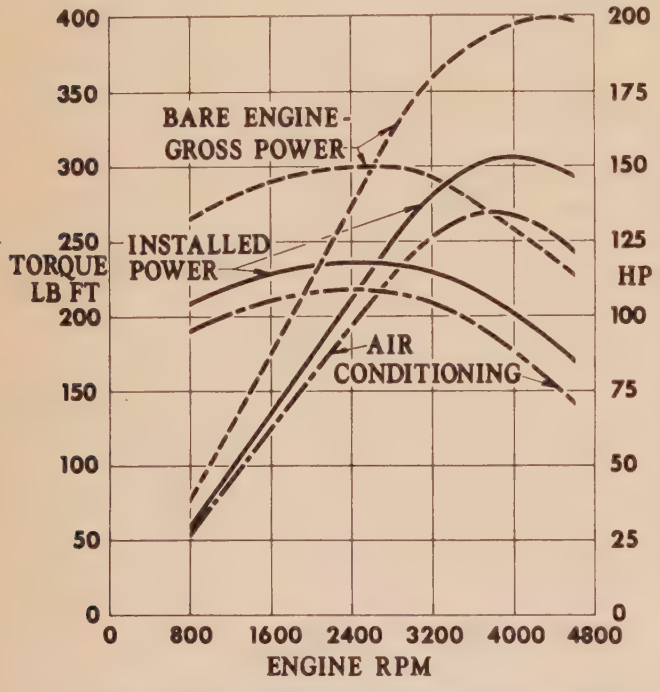


Fig. 13—Net power of flywheel under operating conditions of air conditioned car

given careful consideration by the engineers responsible for overall operation and performance of a particular automobile. As in any refrigeration system, it is necessary to expend a certain amount of energy to transfer heat from one space to another. This energy is expended in two ways in an automobile air conditioning system. The first additional power used is the compressor (Fig. 11). The automobile refrigeration compressor varies from most refrigeration pumps which run at steady speed. The compressor is belt-driven from the crankshaft front pulley,

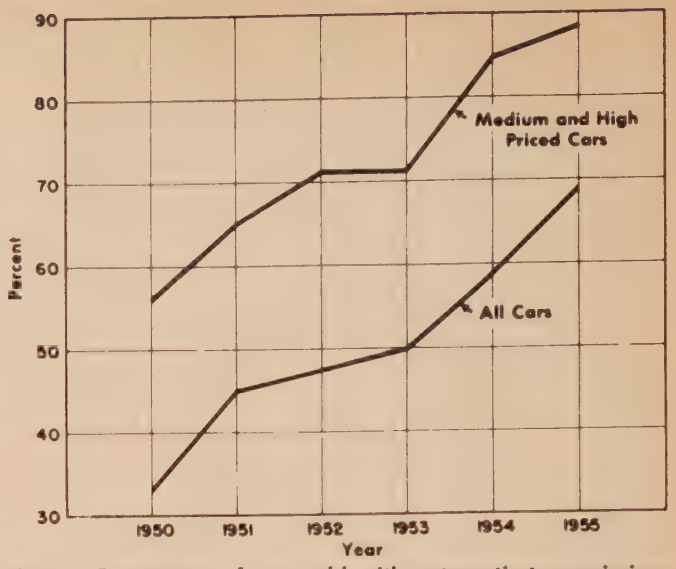


Fig. 14—Percentage of cars sold with automatic transmissions

and its speed is directly proportion to engine speed. The power requirement curve is similar to other accessories showing that the faster the engine turns, the greater the power required. The second power drain brought on by the air conditioning system is through the fan. The compressed refrigerant is condensed (cooled by a second radiator which is usually placed in front of the engine radiator). This, then, requires a larger fan to pull additional air through the two radiators. Fig. 12 shows the air conditioning fan imposed on the standard fan curve.

Deducting the compressor power and the net difference between the fans from the previous engine power curves develops another curve for power at the flywheel (Fig. 13). Again the drain has not been too great in the lower speed ranges, but at the horsepower peak another 11 hp or 7% of the net available has been used up. Although air conditioning is far from being a standard accessory, it is becoming popular enough that engineers must consider its effect on net engine power. It is not feasible to install a special engine to maintain a constant power output to the flywheel. Therefore, the engine designer must weigh all the possibilities and then effect a compromise as to the amount of power in a standard engine to be provided for accessory use while still maintaining good performance under all operating conditions.

Effective Power Transmission

by

W. E. Zierer and H. L. Welch

IT is generally recognized that cars having a high degree of activity, along with comfort and convenience, are most desired by American car buyers. While it is the engine's horsepower that determines the ultimate performance level of a particular car, the attainment of that performance is dependent upon the

optimum application of the power to the rear wheels. It is the province of the transmission to fulfill the latter requirement, and its degree of accomplishment may be considered a measure of overall transmission effectiveness.

The internal-combustion engine is essentially a constant-torque device with practical limits as to how slowly or how fast it can run. Within those limits the engine develops power almost proportional to speed, and its maximum output, therefore, is developed at high engine speeds.

Present-day automatic transmissions, however, combine with the engine to provide the driver with near maximum power for acceleration and hill-climbing ability, over a wide range of car speeds, by merely depressing the accelerator. Control of the transmissions has been simplified to the point where it is accomplished by the movement of a single operating lever or by simply pushing a button.

Trend to Automatic Transmissions

It is not surprising, therefore, that automatic transmissions have become the overwhelming first choice in American automobiles. This is illustrated in Fig. 14, which shows that the percentage of cars sold with automatic transmissions has more than doubled during the past five years. In 1950, 11 years after the introduction of the first commercially successful automatic transmission, 33% of all cars sold were equipped with automatic transmissions. By 1955, the percentage had increased to 68%. In medium- and high-priced cars, the percentage increased from 56 to 88% during this same period.

It is significant also that taxicab and police cruiser applications are showing a growing trend toward automatic transmissions. Safety, freedom from driver fatigue, and reduced maintenance costs are recognized as further advantages of the automatic transmission in heavy traffic operation.

Each type of automatic transmission in common usage today employs a planetary gear train in combination with either a torque converter or fluid coupling. When a fluid coupling is used as the slipping element, a 4-speed planetary gearbox is used. In the case of those using a torque converter as the slipping element, either a 2- or 3-speed planetary gearbox is used. For the purpose of analyzing the performance characteristics of a representative automatic transmission, a 3-speed planetary gearbox with torque converter will be considered, having gear ratios of 2.45 and 1.45 in first and second speed, respectively. The torque converter gives a maximum torque multiplication of 2.34 at a stall speed of 1630 rpm.

Fig. 15 shows a transverse section through the transmission and serves to illustrate the components required to perform automatically the function of providing the proper gear ratio for performance or economy as well as neutral and reverse. With friction bands and clutches and hydraulic pumps to energize them, it is not surprising that additional power is required for their operation. Then there is the efficiency of the hydrokinetic torque converter, which is such an important element of the transmission.

Output from Transmission

Fig. 16 shows that with an input horsepower of 145 at 4000 rpm which might be expected from an engine developing 200 gross hp at 4400 rpm, the

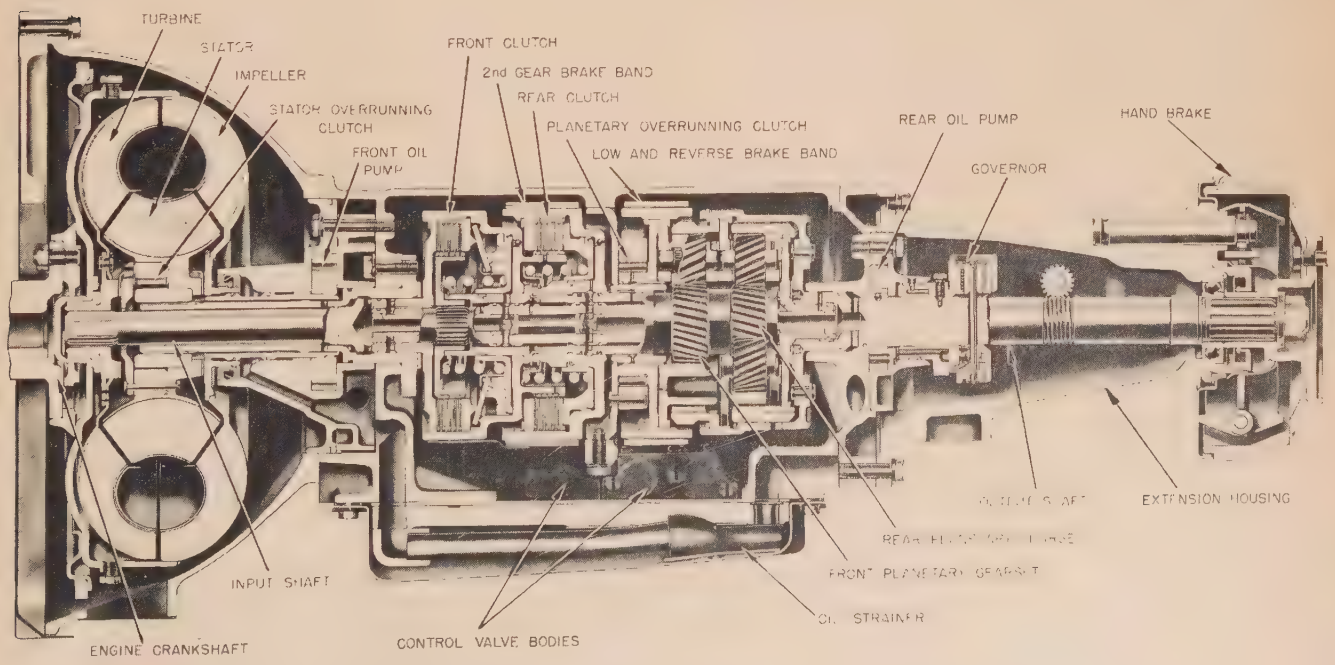


Fig. 15—Three-speed automatic transmission (Torque-Flite)

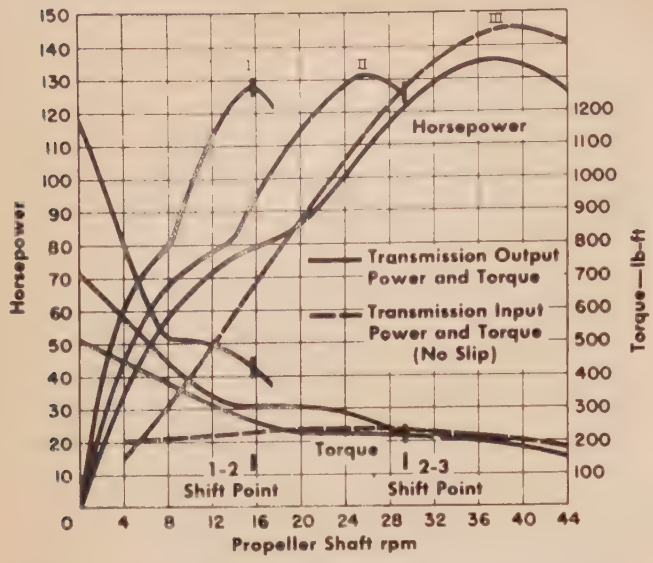


Fig. 16—Automatic transmission output power and torque

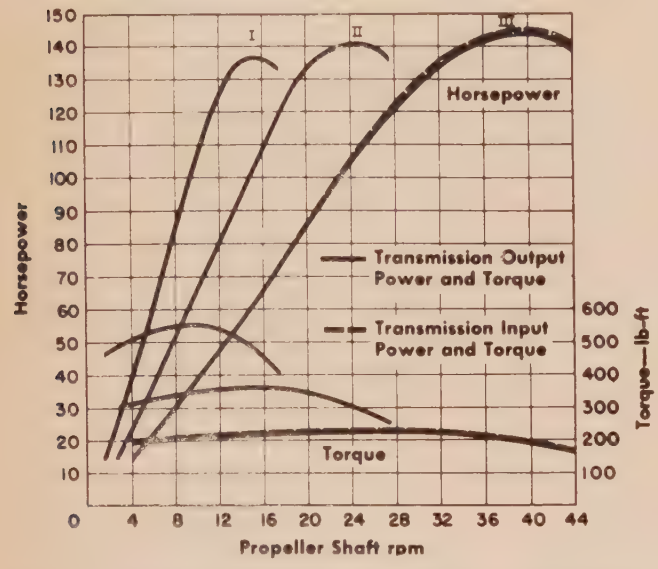


Fig. 17—Manual transmission output power and torque

automatic transmission delivers 135 hp at 3800 output rpm in direct, 131 hp at 2600 in second, and 127 hp at 1600 in low gear. These peak values represent 93, 90, and 87% of the input horsepower, respectively.

The torque curves shown here are interesting since torque is the form of output that permits an evaluation of performance irrespective of speed. Note that the automatic transmission multiplies the engine torque nearly six times for initial starting and greatly enhances performance throughout the normal speed range.

Fig. 17 shows the power and torque output from a 3-speed manual transmission, with the same 145-hp engine supplying the input. For this analysis, a typical gearbox of the reverted gear train type was used. Gear ratios of 2.55 and 1.60 for first and second gears were selected as typical, with high being direct.

Power delivered to the propeller shaft would be 143.8 hp at 4000 rpm in high, 141 hp at 2400 rpm in second, and 137 hp at 1500 rpm in first gear. These values are, respectively, 99, 97, and 94.5% of the input power. The mechanical efficiency of the manual transmission is indeed excellent, but its good performance potential at low propeller shaft speeds is limited in usefulness by the necessity for manual shifting.

The torque curves in Fig. 17 exhibit a notably different characteristic at low speeds from those obtained with the automatic transmission. Less than half as much torque is available for starting with the manual transmission, with a corresponding loss of initial acceleration and gradeability.

Transmission Losses

In considering the power output from the transmission in relation to the power input, it was evident that losses of some significance were present. A better understanding of the amount and character of those losses throughout the speed range can be obtained by referring to Fig. 18. There are shown the wide-open-throttle power losses of the automatic and manual transmissions in direct gear. The total power loss in the manual transmission is very small, being only 1.4 hp at 4400 rpm and is attributable to bearing and seal friction and lubricant churning losses.

Power losses in the automatic transmission are in two categories: namely, drag and slip. The drag losses are composed about equally of pump power requirement and general gearbox friction. The front pump is driven at engine speed and performs the function of supplying oil pressure required for engaging clutches and bands at low car speed. This pump also circulates oil through the torque converter and keeps it under desired pressure. At approximately 1500-rpm propeller shaft speed, the smaller rear pump takes over the pumping duties, and the front pump is unloaded. The rear pump is driven at propeller shaft speed. The front pump is the larger user of power until it is unloaded, after which the rear pump power requirement exceeds it in high gear. Total power absorbed by the two pumps reaches 4.4 hp at 4400-rpm propeller shaft speed in direct. At this same speed, gearbox friction absorbs 3 hp and the band drag a low 0.3 hp.

The slip loss is a manifestation of torque converter efficiency. At wide-open throttle this is the major loss in the transmission up to 3400-rpm propeller shaft speed. The very large power loss indicated at low speeds reaches a maximum at standstill or "stalled" condition where torque multiplication is maximum and output power is zero. The entire engine output of 69 hp at the stall speed of 1630 rpm is lost since no external work is accomplished. However, the very high torque developed is of great utility for

quick starting. In practice, this loss at stall speed is a transient condition of very short duration. The entire range of high power loss is of relatively little significance in regard to fuel economy under most operating conditions because of the time factor.

The sharp peak in the torque converter loss curve at 2000-rpm propeller shaft speed corresponds to the converter output to input speed ratio where torque multiplication stops and fluid coupling action begins. This drop in efficiency at the transition from converter to coupling is present in all torque converters to a greater or lesser degree. Those having less of a drop in efficiency at the coupling point generally suffer in efficiency at lower numerical speed ratios. At higher speeds, the torque converter steadily improves in efficiency, reaching 97% at 4400-rpm propeller shaft speed. The corresponding power loss at that speed is 4.3 hp.

The transmission power losses in first and second gear at wide-open throttle are compared with high gear losses in Fig. 19. It is apparent that at low speeds the effect of the converter loss is minimized due to the gear ratios. As might be expected, however, the drag friction increases when the power is delivered through the planetary gears. Thus the wide-open-throttle power losses in the automatic transmission are appreciably higher than in the manual transmission, whether in first, second, or high gear.

For comparison with the wide-open-throttle losses, a bar chart has been superimposed in Fig. 19 showing the road load loss in the automatic transmission at a car speed of 50 mph. The total loss of 2.6 hp shown is comprised of 1.8-hp friction and 0.8-hp converter loss.

The effect of this amount of power loss at 50 mph would be to reduce fuel economy by approximately 1.8 mpg. Of this economy loss, the gearbox friction would account for 1 mpg while the torque converter would account for 0.8 mpg. Fortunately, there are compensating features of the automatic transmission which offset this loss completely. First, the enhanced performance with the automatic transmission makes it possible to use a rear axle ratio about 12% lower numerically than is desirable with a manual transmission. At this speed, the 12% reduction in axle ratio should result in an improvement in fuel economy of 1.5 mpg. Secondly, with an automatic transmission it is usually possible to modify carburetor and distributor calibrations to gain an economy increase of from 0.1 to 0.3 mpg without noticeable surging. Furthermore, while this analysis is being made with a given engine, it is possible that the elimination of low-speed lugging operation with the automatic transmission would permit an increase in compression ratio with a resulting gain in fuel economy and performance. Thus, it may be seen that any loss in fuel economy that may result from the use of the automatic transmission is not the effect of equilibrium or constant-speed driving but is the result of warmup effects or other driving conditions where unusually high slippage occurs.

Effective Power Transmission

Although the convenience of "no shift" driving might justify the power absorbed in an automatic transmission it would be wrong to think of it as another power-consuming luxury. The function of the transmission takes on greater significance when it is recognized that it makes the high-speed power potential of the engine readily available at all car speeds. It also provides road load cruising with relatively

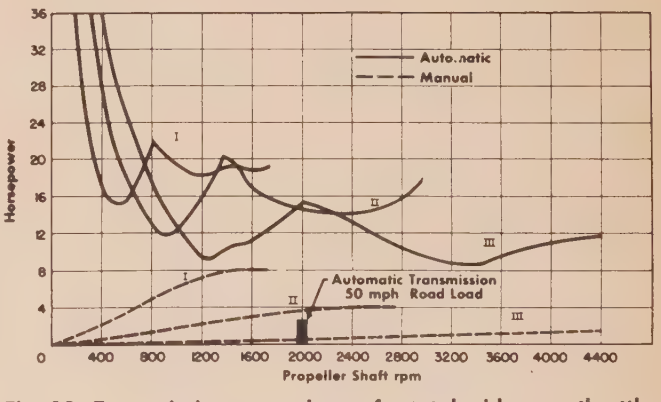


Fig. 19—Transmission power losses for total wide-open-throttle losses in all gears

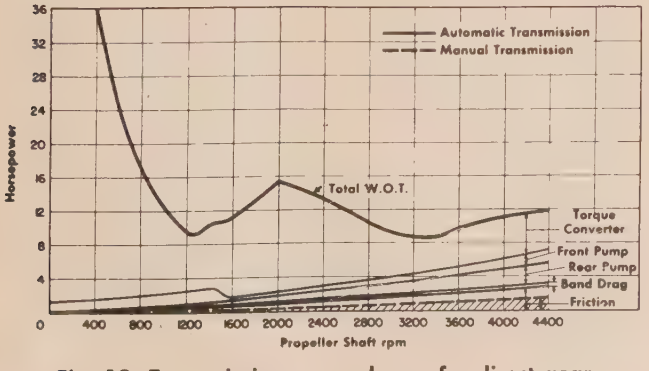


Fig. 18—Transmission power losses for direct gear

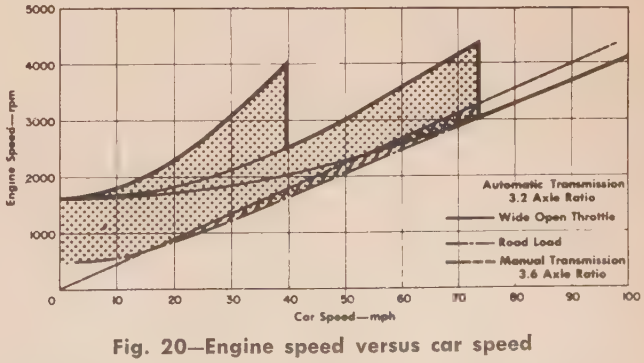


Fig. 20—Engine speed versus car speed

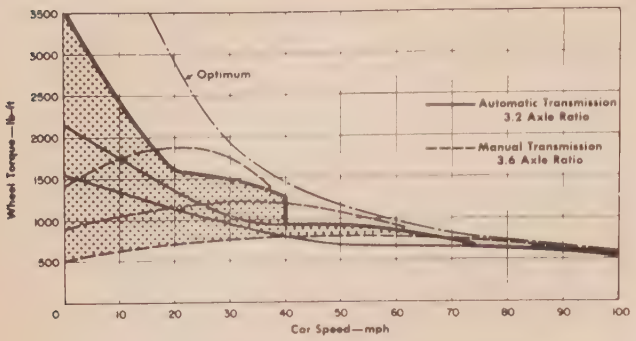


Fig. 21—Rear wheel torque at constant speed

low engine speed, for maximum benefits in the form of smoothness and quietness.

A better understanding of the contribution of the automatic transmission can be obtained with the aid of two illustrations. In Fig. 20, the engine speed versus car speed relationship is shown for both automatic and manual transmissions. The double-shaded area shows the reduction in road load engine speed accomplished by using a lower numerical axle ratio with the automatic transmission. In this case a 3.2 axle ratio has been selected for the automatic transmission and 3.6 for the manual. These values are in line with current industry practice and represent a difference of 12% referred to previously.

The single-shaded area shows the range of engine speed that is automatically available whenever additional engine output is desired. As a result, the available engine power is greatly increased by means of running the engine faster at normal car speeds whenever high performance is called for.

Fig. 21 shows a similar comparison of maximum rear wheel torque with both transmissions. Here the shaded area represents the large measure of extra torque that the automatic transmission makes available without driver effort at all speeds up to 68 mph. Although the manual transmission also makes higher torque available in first and second gears it, of course, can only be achieved by manual shifting. In practice it has been found that American drivers shift gears for starting and hill climbing but seldom downshift to augment acceleration where the vehicle is traveling along in high gear.

With the automatic transmission at speeds below 20 mph in low gear, the extra torque multiplication is accomplished by the converter. In the application of this type of transmission to an actual car, the torque converter characteristics would be carefully tailored to achieve maximum low-speed acceleration within the limitations of wheel slip, rotating inertia, and general noise level considerations.

The converter also provides increased torque for instantly responsive acceleration in high gear at speeds up to 50 mph.

In order to appreciate the degree to which the automatic transmission approaches the optimum application of power to the rear wheels an optimum limit curve is also shown in Fig. 21. This curve is based on the maximum input power being applied to the rear wheels at all car speeds. While this bogey is, of course, impossible of achievement, the close similarity to the actual rear wheel torque curve with the automatic transmission shows how effectively the engine power is applied at all operating speeds. From this it is apparent that in combination with the automatic transmission, further improvements in engine output can be very useful in improving car performance at normal operating speeds.

Summary

We have seen that there is very little power loss in the manual transmission, being only 1½ to 8 hp at maximum engine output depending upon the gear used. In the automatic transmission the losses are considerably higher (10 to 18) at maximum horsepower. At low speeds the losses can be even greater. The comparison of the automatic and manual transmission therefore leads to a paradox:

1. From an efficiency standpoint the manual transmission has the higher efficiency and can therefore transmit more horsepower to the propeller shaft.
2. From the standpoint of effectiveness, the automatic transmission gives better performance at normal operating speeds and smoother, quieter cruising.

The explanation for this is that because the automatic transmission permits the engine to develop more power at low and moderate car speeds, it is possible to deliver more usable power to the propeller shaft with lower transmission efficiency. Also, from the practical standpoint, the lower efficiency is significant only during the small percentage of the total operation, that is at high performance level. For road load operation, the slight efficiency loss is almost entirely balanced by the practicability of using a lower axle ratio. Expressed otherwise, an investment of a few horsepower expended in the automatic trans-

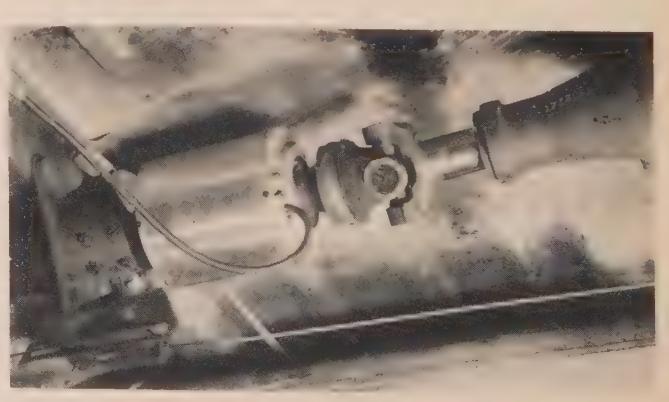


Fig. 22—Propshaft torquemeter assembly for measuring torque input to axle

mission yields handsome dividends in terms of better car performance at the most used car speeds.

Acknowledgment

This paper was written with the help of a number of our colleagues in the Engineering Division. Their valuable contribution and willing assistance are gratefully acknowledged.

Wind and Rolling Resistances

by

L. C. Lundstrom

AUTOMOTIVE engineers have been intrigued for years by the many problems associated with the wind and rolling resistances of automotive vehicles. One of the earliest references to the measurement of rolling resistances is found in E. H. Lockwood's SAE paper, "Chassis Friction Losses," dated 1922. Early measurements of wind resistances were made by L. E. Conrad, and reported in a 1925 issue of "Public Roads."

A cursory examination of wind and rolling resistance data published through the years by these and many other engineers leads one to the opinion that many of the observations and calculations were erroneous and conflicting. As our knowledge of the subject has been strengthened by improved instrumentation and technique, we now consider that possibly the largest error in the early measurements was in the identification and recording of the test variables. Rolling resistance was often assumed to vary only with load, but speed, tire composition, and ambient air temperature may have even greater effects. Similarly, the effects of ambient air temperature on wind resistance never have been properly recognized. It is the purpose of this discussion, therefore, to reevaluate the variables encountered in wind and rolling resistance determinations and to show how they affect the theoretical car described in this symposium.

Fig. 23—Wheel torquemeter for measuring output torque

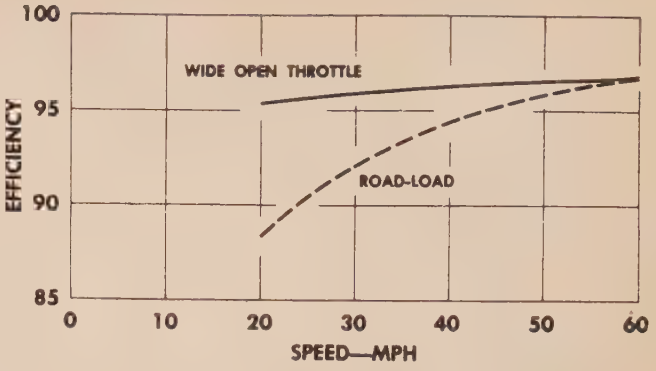


Fig. 24—Rear axle efficiency at wide-open throttle and road load

As many of the determinations of wind and rolling resistance have been based on coasting tests and towing tests, the rear axle resistance has been included conventionally as a part of rolling resistance. In recent years, however, torquemeters have been installed on road test vehicles to determine axle efficiency directly. Fig. 22 shows a propshaft torquemeter to measure torque input to the axle, and Fig. 23 shows the wheel torquemeter to measure output torque. Observations of the rotative speeds of the shafts are obtained by the use of electronic counters and timers.

As the axle ratio for the theoretical car is 3.2/1, the efficiency of the overall axle is determined under any condition by:

$$\frac{\text{Output Torque}}{3.2 \times \text{Input Torque}}$$

Very little data has been released on axle efficiencies under all load conditions, partly owing to the difficulty in obtaining sufficient accuracy when the results depend upon the small differences between two large quantities. The data shown in Fig. 24 illustrate that the overall efficiency of a typical axle is 95% to 97% under full throttle but may drop below 90% under low-speed road load conditions. Seal friction, churning losses, and bearing losses probably account for the lower efficiency under light load conditions.

The available or maximum axle torque and axle power plotted in Fig. 25 were obtained by applying the rear axle efficiency of Fig. 24 to the propshaft output torque and power of Fig. 26. Also plotted on the same graph are the assumed road load axle torque and axle power curves for the theoretical 4000-lb car. This road load torque supplies the force required to propel the car at constant speed on a level, hard-surfaced road, under ideal conditions. As the vehicle is in equilibrium, the force is, of course, equal to the sum of wind and rolling resistances. When a complete vehicle is available, axle or wheel torquemeters can be installed to measure the road load torque directly,

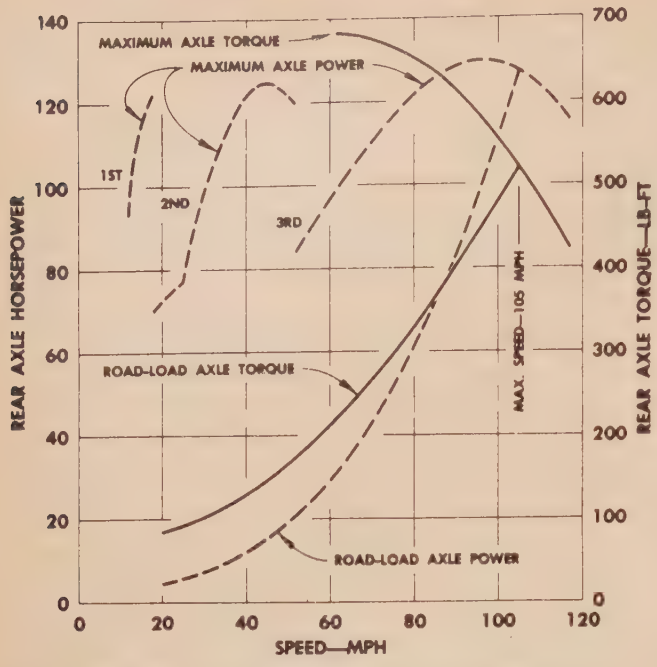


Fig. 25—Maximum and road load axle torque and power of theoretical car

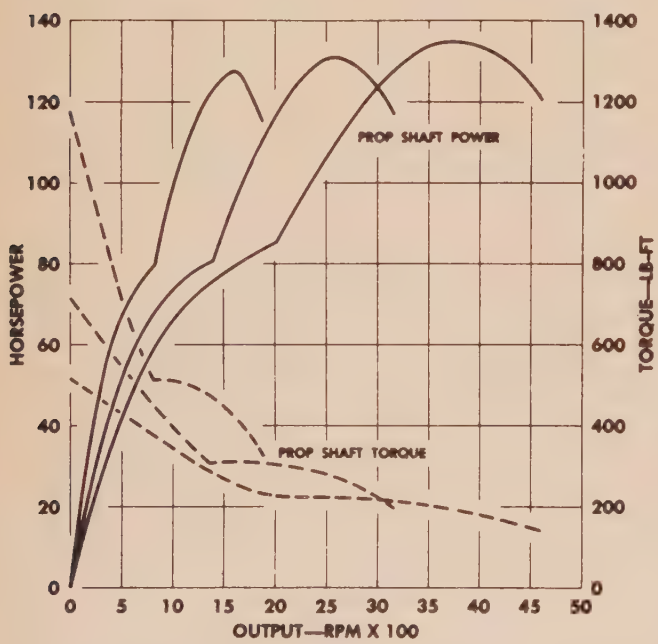


Fig. 26—Transmission output torque and power for theoretical car

and the corresponding road load power is computed from the simple formula:

$$\text{hp} = \frac{\text{TN}}{5252}.$$

The intersection of the road load axle power and maximum axle power curves defines the theoretical maximum speed of this theoretical car. As plotted, the vehicle would have a top speed of 105 mph, but this value could vary considerably if the wind and rolling resistances are above the ideal minimum. In the high-speed range for this car, nearly 4 hp are required to increase the car speed 1 mph.

However, when the road load torque and power requirements must be computed for a vehicle still in the design stage, the opposite approach is required, and the effects of the various wind and rolling resistances must be considered. The physicist and the mathematician can derive basic equations for these resistances, but the engineer must supply the constants. Unfortunately, the wind and rolling resistances involve many variables: friction in the wheel bearings and seals, friction within the tire, friction at the tire-road contact, wind or air drag of the body and chassis, and churning of air by the wheels. If the new design deviates from conventional practice, the engineer can make only educated guesses in the selection of constants for the computations.

The equation commonly used [2] for the axle power to overcome wind and rolling resistances under road load conditions is:

$$\text{Road Load hp} = \frac{V}{375}\ (K_1 W + K_2 A V^2)$$

where:

W = Weight of car, lb
A = Frontal area of car, sq ft
V = Velocity, mph
K_1, K_2 = Rolling and air resistance constants

This equation does not include all of the variables but does provide a simplified computation that is certainly helpful in initial design studies. For the assumed road load curves of Fig. 26, K_1 is 0.0149 and K_2 is 0.00149; these values are within the ranges observed experimentally and give the performance characteristics assumed for the theoretical car.

We must further qualify the calculation, as the constants are based on the ideal test conditions of level, hard-surfaced road, 70 F air, and 0 wind. The theoretical car was assumed to have conventional body styling with 25-sq ft frontal area and current production 6.70 x 15 tires at standard inflation pressure. These rather ideal conditions are necessary for consistent computations and tests, but the customer seldom is operating his car under ideal conditions. The effect of the variables are overlooked at times by the engineer in his appraisal of a new design feature or evaluation of performance under less than ideal conditions.

We expect the constants for rolling resistance to be slightly different for tires of different compositions, construction, sizes, loads, and inflation pressures. The tire companies may be able to assist in the selection of proper rolling resistance values by supplying data from laboratory tests of new products.

As an example, the torque and power differences

[2] GM Automotive Engine Test Code.

between current production and natural rubber tires are shown in Figs. 27 and 28, respectively. Production tires with both synthetic and natural rubber have some advantages over 100% natural rubber tires, but the differences in the road load power curves of this illustration are due entirely to the differences in composition. At 60 mph this amounts to 2 hp, or 14-lb-ft torque. Higher inflation pressures will reduce the tire deflection and flexing and, consequently, the rolling resistance. For high-speed driving, the reduction of resistance with higher inflation pressures also shows up in terms of lower tire temperature.

Precision road tests now have confirmed earlier opinions [3] that air temperature has a very definite effect on road load torque and power requirements. Figs. 29 and 30 illustrate the variations in torque and power that can be expected for the theoretical car when operating in 30, 50, and 70 F air. The 70 F curves are identical with the road load axle power and torque curves shown on Fig. 26 and are consistent with the proposed road load power equation presented in the preceding paragraphs. Note that at 60 mph 5½ more hp are required to propel the car at 30 than at 70 F. This amounts to approximately 0.5% change in power requirement per degree of temperature change, or 20% change between 30 and 70 F. In the normal temperature range, this change or variation with temperature is approximately linear.

The sizeable temperature effect is probably the principal reason for the wide variation in results published by different observers over the years. Their data probably were accurate for the conditions under which the tests were performed, but the air tempera-

[3] "Passenger-Car Wind and Rolling Resistance," by K. A. Stonex. Paper presented at SAE Passenger Car, Body, and Production Meeting, Detroit, March 10, 1949.

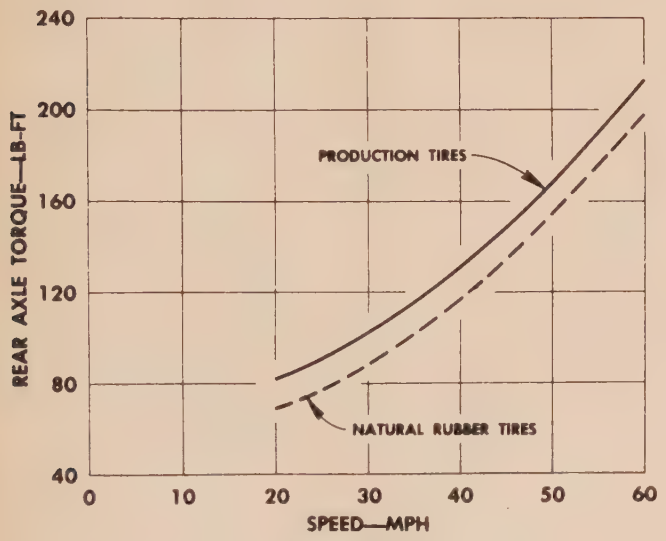


Fig. 27—Effect of tire composition on road load torque at 70 F air temperature

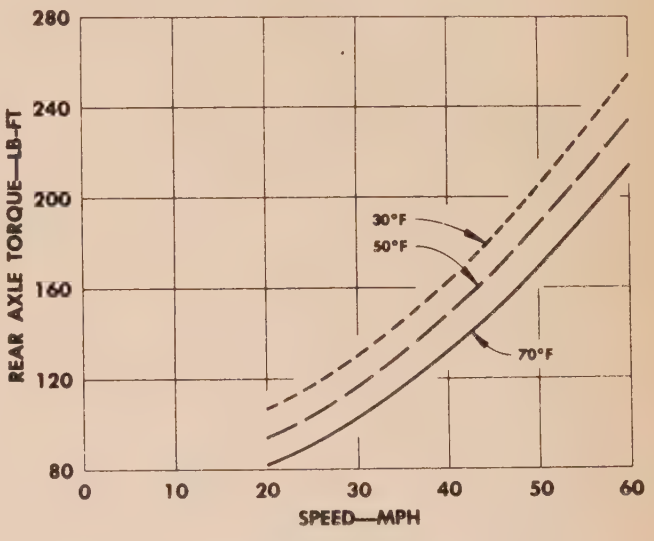


Fig. 29—Effect of air temperature on road load torque

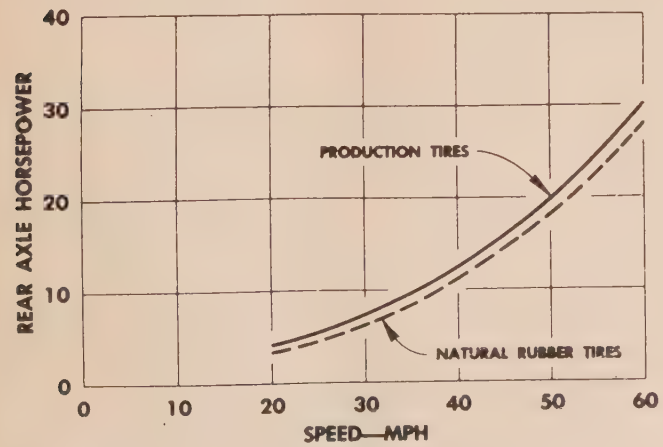


Fig. 28—Effect of tire composition on road load power at 70 F air temperature

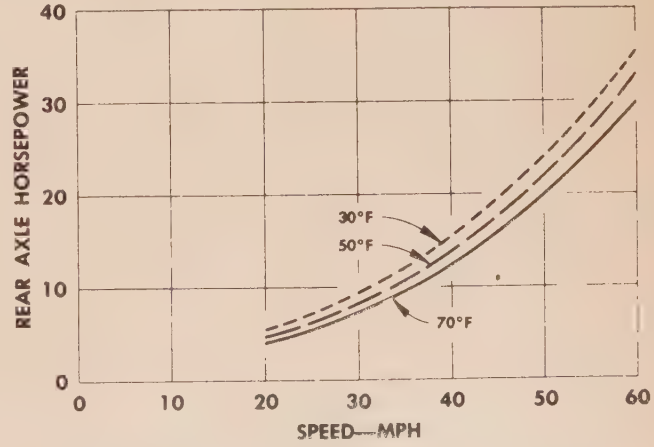


Fig. 30—Effect of air temperature on road load power

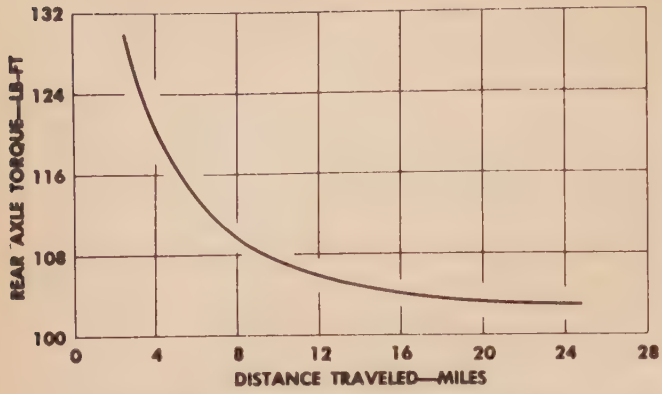


Fig. 31—Change in road load torque during warmup at 30 mph and 70 F air temperature

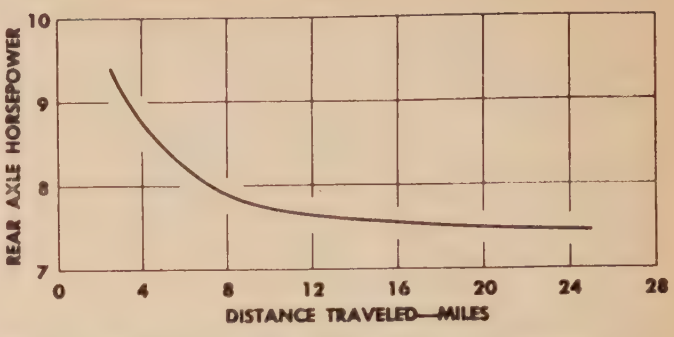


Fig. 32—Change in road load power during warmup at 30 mph and 70 F air temperature

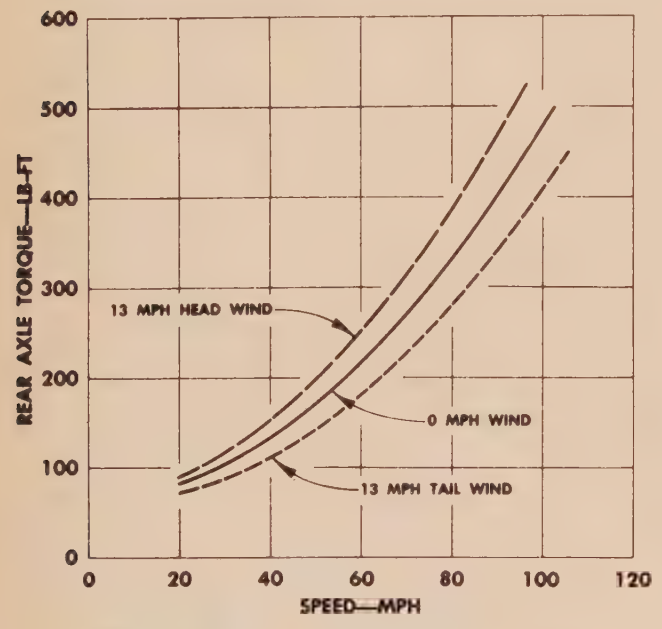


Fig. 33—Effect of wind on road load torque

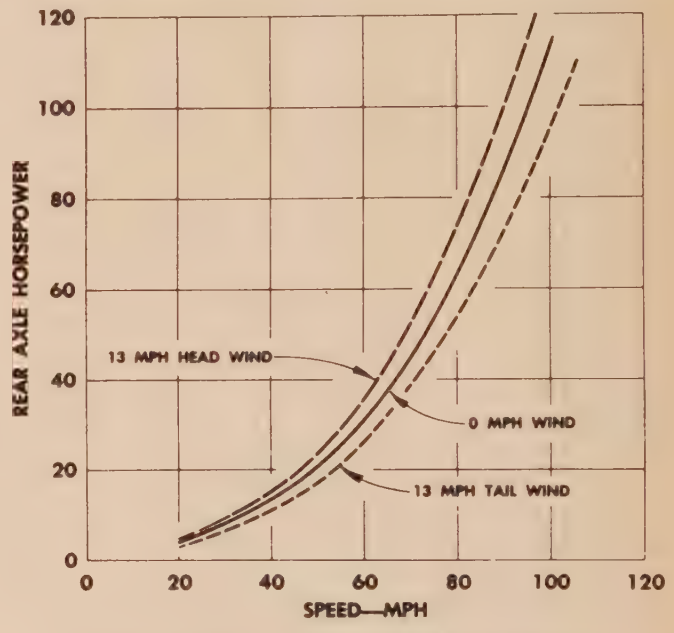


Fig. 34—Effect of wind on road load power

tures usually were not recorded, and insufficient data were taken to obtain a temperature correction factor. Considerable experimentation still will be required before final air temperature correction factors can be added to the road load equation. We can suggest only that temperature data be recorded on all road power tests and corrections be made to a 70 F baseline, if at all possible.

The influence of temperature likewise can be shown on rate of warmup tests. If the theoretical car starts a warmup evaluation from 70 F, the road load axle torque and power will vary with distance traveled, as illustrated in Figs. 31 and 32, respectively, owing to change in rolling resistance with tire temperature. Over 20 miles of driving at 30 mph is required to stabilize the rolling resistance factors. As indicated previously, the engineer must test at the stabilized condition, but many cars are driven for short distances only and never reach the stabilized minimum values of rolling resistance.

In keeping with the title of this part of the symposium, the words "wind resistance" have been used in place of the more exact terminology, "air resistance." Regardless of wording, it is still remarkable that the modern car can be driven at speeds that produce velocities equivalent to winds of hurricane force. For simplification of engineering studies, the wind (or air) and rolling resistances usually are computed for still, ambient air conditions. However, natural wind currents either may increase or decrease the air resistance factors, depending upon direction of the wind. This is illustrated on Figs. 33 and 34, by plotting respectively the torque and power required to overcome the "air" and rolling resistances plus (or minus) the additional power to overcome a wind of 13 mph. This wind velocity is quite normal

for most of the country, and at 60 mph, car speed can increase the wind resistance factor by approximately 5 hp. If the car is operating with a tail wind, a corresponding 5-hp reduction in rolling resistance is obtained.

A discussion of rolling resistance factors would not be complete without some mention of the effect of road surfaces. Figs. 35 and 36 show the rear axle torque and power to propel the theoretical car on various road surfaces. As only the rigid pavements retain consistency, the values for damp gravel and soft gravel and sand are plotted for general interest only. It is interesting to note that the effect of moisture on the pavement can be evaluated. For the data plotted, the moisture increased the rolling resistance by approximately 5%.

Although the symposium deals with power, the corresponding torque values also have been presented. The theoretical symposium car could not be tested, but the values discussed were based on extrapolated data from cars of approximately the same specifications. For these computations, the tire rolling radius (for the 6.70 x 15 tires) was assumed to be 1.11 ft. Using this figure, the torque values also can be reduced to the equivalent forces to overcome wind and rolling resistance.

A review of the data just presented will indicate the difficulty in predicting accurate wind and rolling resistance values ahead of actual road test. Variations in tires and the effects of temperature appear to be the largest uncontrollable items. In addition, the air resistance coefficient will change as the degree of streamlining of the body changes.

Fig. 37 is included simply to illustrate the range of power that is required to overcome wind and rolling resistances. At the normal driving speed of 60 mph, the power ranges from 29 to 49 hp. This difference is attributed simply to a change of tires, a 40 F drop in temperature, wet pavement, and a 13-mph head wind. By going to extreme conditions, the range of values could be doubled over that shown.

While the modern automobile has sufficient power to overcome even the highest wind and rolling resistance values, the engineer should, and does, recog-

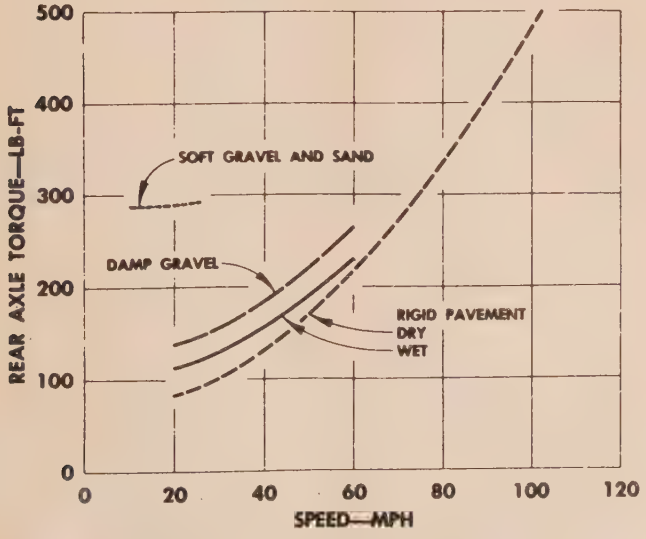


Fig. 35—Effect of road surfaces on road load torque

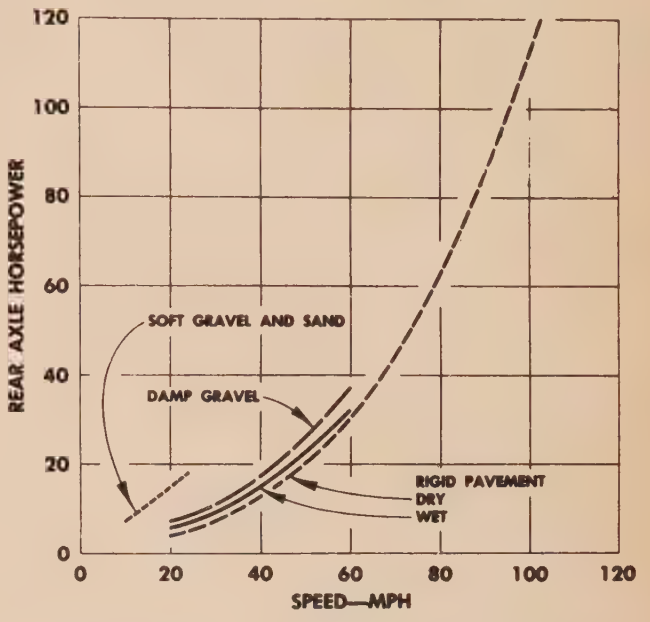


Fig. 36—Effect of road surfaces on road load power

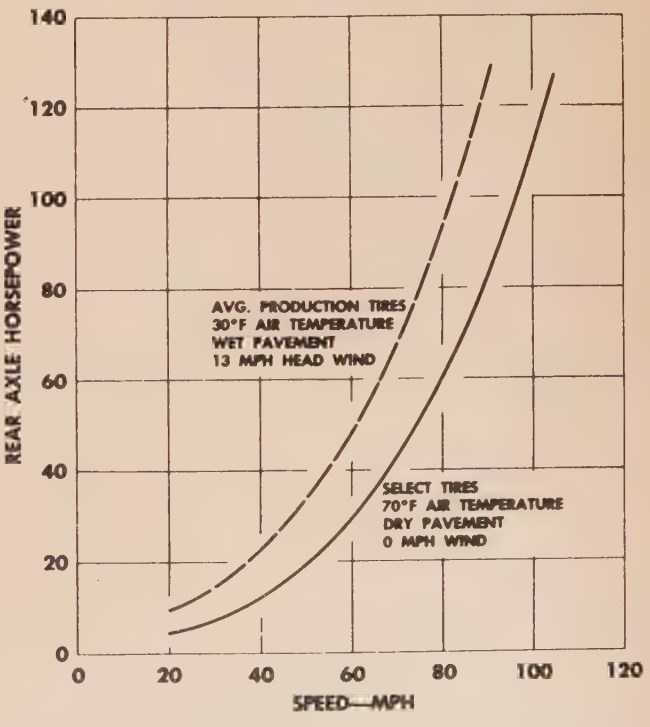


Fig. 37—Effect of test conditions on road load rear axle power

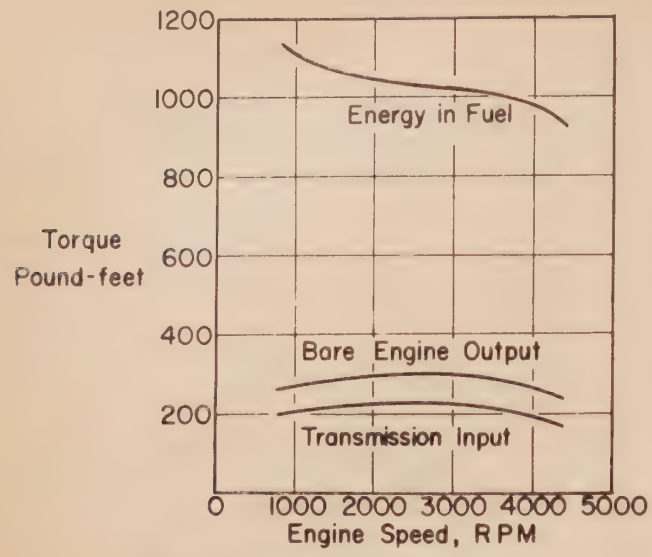


Fig. 38—Reduction in fuel energy

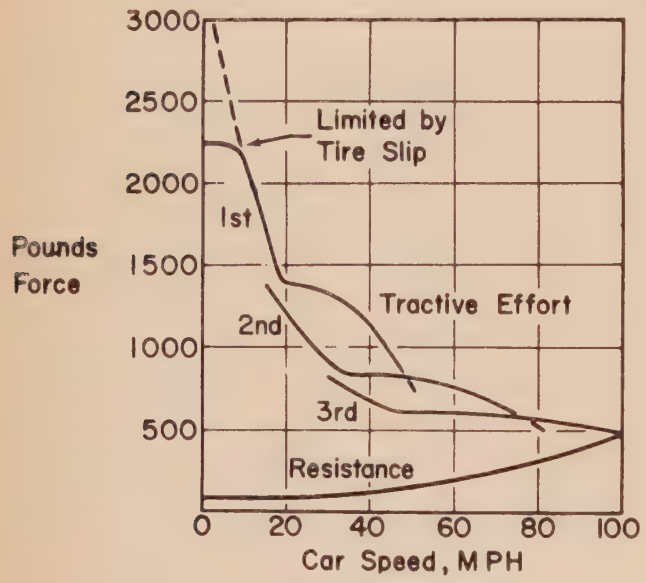


Fig. 39—Tractive effort and resistance

nize that the higher resistance values demand higher fuel consumption. In fact, the variations in fuel are even more striking than the variations in road load axle power.

What the Customer Gets

by

T. D. Kosier and W. A. McConnell

WE have had built up for us by the preceding parts of this symposium a hypothetical car, although a very typical one in line with cars on the road today.

Figs. 38 and 39 show what has happened so far. Fuel energy has become bare engine output, which has then been further reduced by environment and accessories to deliver an input to the transmission. In Fig. 39, we see what the transmission has done—plotted now against vehicle, rather than engine speed. The variable ratios have multiplied torque at low speeds, making it possible to use more of our maximum power output at whatever speed we choose to run. Transmission output torques have been further multiplied by the axle ratio and efficiency to get a wheel torque, which has been converted into an equivalent force, or thrust on the road. The total bearing loss, tire rolling resistance, and wind resistance which must be overcome at each speed, is also shown. By subtracting resistances from the tractive effort, we find the surplus pulling force, after al losses have been overcome, which our customer can use to climb grades, pull trailers, or accelerate. Fig. 40 shows what is left of our power: at 100 mph, nothing; at 50 mph, some 650-lb of surplus force, or about 88 hp—above that needed to maintain steady speed on a level road—available merely by opening the throttle wide. Notice that the effect of our transmission and the losses encountered is to give the customer the most surplus power in the 40- to 60-mph range, where he most often needs it.

So far, every time some of our power is used, we have called it a loss. To be consistent, we might say also that when the customer opens his throttle, he gets a "performance loss." We have then accounted for where all our power has gone. It's been lost—used up—every bit of it. Horsepower is just a measure of the rate at which energy is expended, and energy—at least until a few years ago—could neither be created nor lost. Our fuel energy isn't really lost either; it has just been spent for a variety of purposes, some of which we may not be aware. Theoretically, at least, and neglecting losses, it takes something less than no energy at all to move a person from Detroit to Atlantic City, since it is all downhill; whether we use 30 or 40 gal of gasoline in the process, our efficiency is still zero. So we aren't able to use the terms "efficiency" or "loss" in describing what the customer gets. Efficiency is meaningless if we can't express comfort, convenience, and travel distance in energy units. Spending energy is like spending money; counting your change is not a very good way to tell whether or not you got what you wanted. Our losses, then, should be considered as costs—what we pay to get something we want.

Grade Capability

One of the first things the customer wants for his money is to be able to climb any hill he might encounter and at any speed he may want—or be permitted—to travel. The surplus force available at full throttle shown in Fig. 40 is a good measure of our car's ability to do this.

The power used in the accessories, transmission, axle, bearings, tires, and to overcome wind resistance

was determined under conditions of full throttle, and will be the same at steady speed on a grade as at steady speed on level ground. The weight of a vehicle is always a force acting straight down, so in climbing a grade, this weight will have a component parallel to the pavement, and a slightly diminished component perpendicular to the road which has no significant effect on our performace. Thus, on a 10% grade, about 10% of our vehicle weight, or 400 lb, is the component in the direction of travel which must be equalled or balanced by the surplus thrust—or drawbar pull as it is sometimes called by test engineers, even though automobiles don't have drawbars like trains and tractors to which this pull can be hooked. By converting our surplus force scale of Fig. 40 into the slope which will produce a downgrade weight component equal to that force, we get the grade capability envelope in Fig 41. This tells us what grades we can climb—and how fast.

Many people think that high engine output isn't really essential to obtain a high force. With a train of cogwheels long enough, you could spin the earth. Some of you may remember that this experiment was tried by an engineer of the Earthworm Tractor Works some years ago. To get a whee of a force, he calculated, would take one heck of a ratio. So he rounded up 25 transmissions, 1/1 ratio in high, 5/1 ratio in low, and coupled them together one after the other in a specially lengthened chassis. With all his transmissions in high, he drove out to the testing grounds, rolled up to the starting line, shifted everything into low, got his timers ready, and kicked open the throttle. He retired recently after 40 years with the company. His engine had been running steadily all those years three shifts a day, seven days a week. He retired before the experiment was completed because he discovered his mechanics had left 1 deg of backlash in that last set of gears, and it would be another 525,000 years give or take a few millenniums, before all that slop would be taken up.

This serves to illustrate, in the extreme, that high ratio produces a high force but only at a sacrifice in speed. Our customers would not be satisfied with a car that would take a lifetime to climb a mountain.

Acceleration Performance

There are only three things our customer can do to put his car where he wants it on the highway. He can start it, he can point it, and he can stop it. In order to keep his car in a spot on the road which is temporarily unoccupied by another vehicle, it is desirable that he be able to do these things rapidly. He wants his car to accelerate rapidly on demand. We have already seen how the Glide Fliteomatic transmission helps him use as much of the available power of his Thunderstorm Firebug V-8 as possible at whatever speed he happens to be going at the time.

Actually, there are also limits to the maximum ratio usable to increase acceleration—limits which are a lot lower than the 300 quadrillion of the experimental Earthworm and which have already been reached by many present-day cars. With this in mind, let's calculate the acceleration performance of our hypothetical car, and see what our customer gets in the way of increased mobility.

Since acceleration, according to Newton, is equal to the force acting on a body divided by its mass, we should be able to take our surplus force or drawbar curve of Fig. 40, divide the vertical scale by the 4000-lb vehicle weight, and come up with an acceleration curve, in g units. Unfortunately, the actual acceleration we measure on automobiles falls short of what we calculate in this way by an amount varying from 0% at maximum speed to 100% short at the start. In situations like this, it sometimes helps, no matter how absurd it may seem, to assume that nobody is wrong. The engine and advertising de-

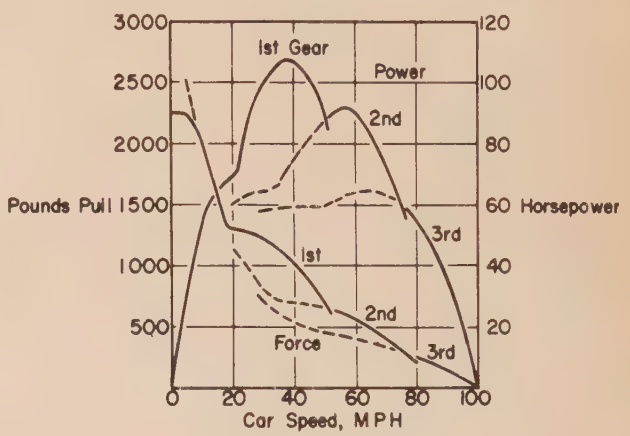


Fig. 40—Surplus pulling force and power

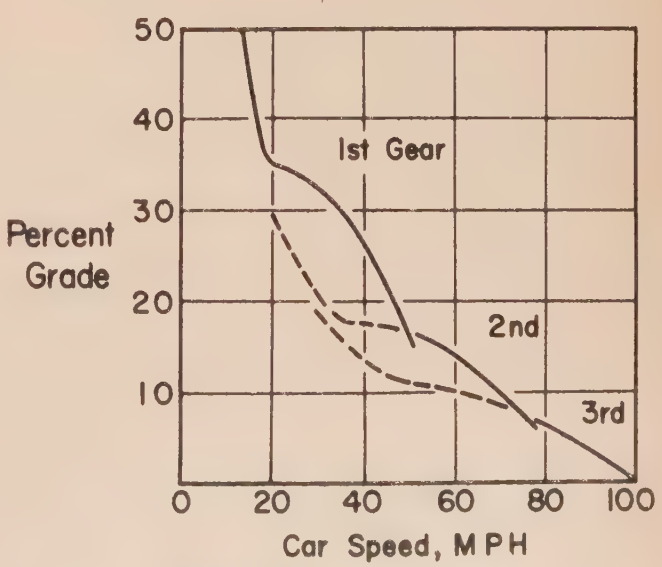


Fig. 41—Grade-climbing ability

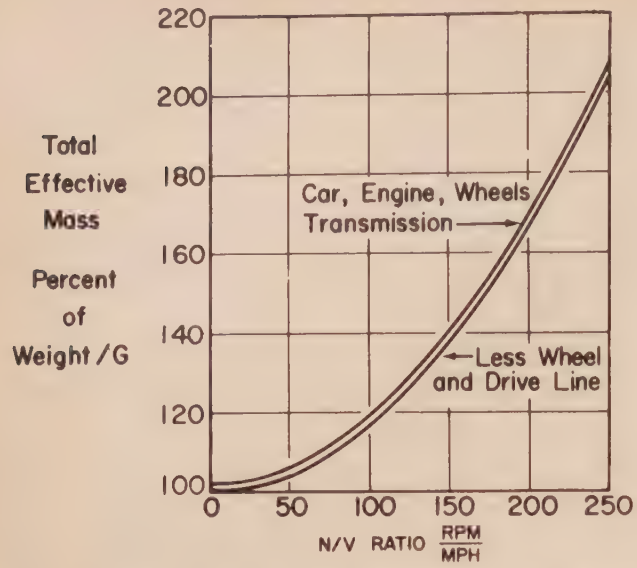


Fig. 42—Relation between overall speed ratio and effective mass

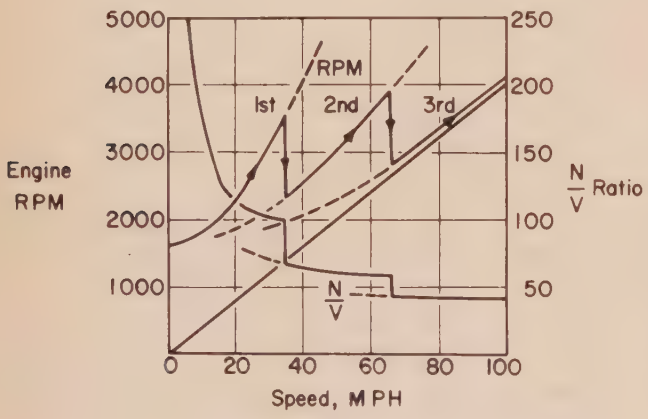


Fig. 43—Engine speeds and overall speed ratios

partments are talking about the same size horses. The transmission engineers did not inadvertently use axle grease for converter fluid. The car does not have dragging brakes, and the test engineer read his stopwatch correctly. Newton may be right, too. If all this is so, it is apparent that the weight, or rather the mass of our car, must be changing. That is the basis of the curve we have drawn in Fig. 42, plotted against N/V ratio, and showing what the mass of our vehicle would have to be for everybody to be right. You will recognize this as the familiar "answer factor" which we learned in school, and it was derived in exactly the same way. It is the number we have to multiply the actual observed acceleration by in order to make it agree with what we calculated for an answer.

This particular answer factor, however, exhibits some interesting qualities. To begin with, it is a "square" curve. If we double our N/V ratio, the amount we must add to our vehicle mass is increased 4-fold. But more interesting is the fact that this answer factor curve seems to be the same, or very nearly the same, for all the automobiles we have checked. In test work, a phenomenon like this is strange enough to require investigation.

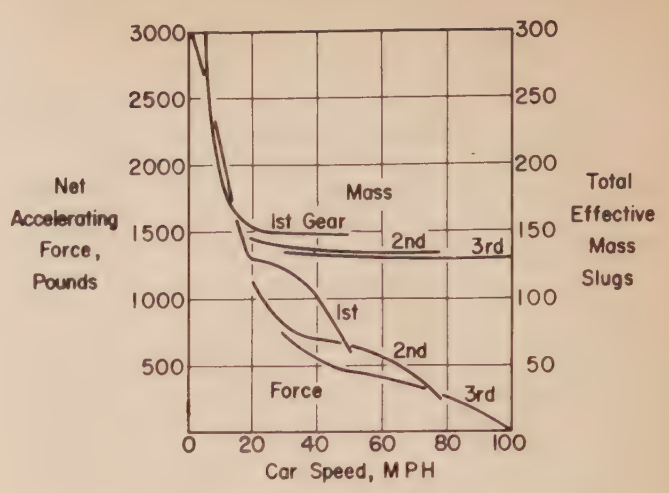


Fig. 44—Available force and effective mass

A measurement of the polar moments of inertia of engine-transmission combinations can be found by observing the change in full-throttle driveshaft torque of a vehicle as it is caused to accelerate or decelerate at various rates by varying a towed load. All inertias we have checked so far fall within the range of 0.40 to 0.50 lb-ft sec,^{2} with the higher values associated with the larger engines on the heavier cars. It is therefore possible and convenient to express this engine rotational inertia as a per cent of the vehicle mass, or, as we have done with our "answer factor," we can plot a factor which can be used, with vehicle weight, to establish the total effective mass of our vehicle as a function of N/V ratio.

This mass factor has been increased by a constant amount to accommodate rotational inertia of the wheels. Wheel angular acceleration bears a fixed relationship to vehicle acceleration, and car wheels have inertias ranging between 0.90 and 1.10 lb-ft sec^2 each, again with the higher values found on the larger wheels and tires of the heavier vehicles. Expressed in terms of a linearly-accelerated weight, wheel inertia is equivalent to very close to 2% of the vehicle mass on all cars.

Thus, if we know or can calculate tractive effort under steady speed conditions, and have N/V ratios and vehicle weight, we can get a very good approximation of the total mass to be accelerated and can then determine the vehicle acceleration performance.

Fig. 43 shows the engine speeds of our hypothetical car, and the resulting N/V ratios at various vehicle speeds. Using this curve to find the overall N/V ratio at each speed, we refer then to Fig. 42 to find the total effective mass corresponding to each N/V ratio. Now that we know the total mass to be accelerated at each speed and the total available surplus

force (Fig. 44), we can divide one by the other and get our vehicle acceleration. That is how the acceleration curve in Fig. 45 was obtained.

There is, of course, another way of making this calculation. We can show that under accelerating conditions, the engine uses some of its power just to wind itself up, and fewer horsepower get into the transmission. However, the transmission engineers assure us that the torque-ratio versus speed-ratio characteristics of their boxes are not much changed—that is, efficiencies are relatively constant—for a fairly wide range of input powers. We find, therefore, that calculations are much simpler, and the engine engineers are happier if we assume, instead, that full power is being developed and the total mass is varying. Engine, transmission and chassis are all responsible for making that mass increase at the higher N/V ratios, and they can share the blame.

Since our customer may be more acquainted with speed or distance than with feet per second^2, the acceleration versus speed curve can be used to find out what the speed will be after various elapsed times and also to find the distance which will have been traveled. Rather than do double integrations in our heads, we'll skip over the calculations. These were done by a committee; if they aren't right, nobody's responsible. Time-distance relationships such as this, shown in Fig. 46, can be used to find passing times, and passing distances required—something of special interest to our customers.

If a car travels at a constant speed, the time-distance relationship is a straight line, with different slopes for different speeds. By comparing the straight line representing, say, 50 mph, uniform speed with enlarged portion of the time-distance curve of our hypothetical car which starts at 50 mph, we can find the elapsed time and the distance required for our Firebug V-8 to gain any desired distance over some other car going 50 mph. The example shown in Fig. 47 depicts this situation. Our car is going 50 mph behind another car going 50 mph. When the throttle is opened, we begin to accelerate, and our time-distance curve begins to rise above the uniform time-distance line of the other car. After 6 sec have elapsed, and we have traveled 520 ft, the other car will have traveled 440 ft, and we have gained 80 ft—enough to come from behind, pass, and pull back in line. This is where our 200 hp and automatic transmission are especially appreciated. If there is another car coming from the opposite direction at about the same speed, we still need over a fifth of a mile clear road ahead before we dare pull out.

The curve (Fig. 42) which relates total effective mass to N/V ratio reveals also a limit on gear reduction which can be used to gain acceleration. The equivalent mass of the engine and transmission rotating parts increases as the square of the overall ratio. The force available for acceleration only increases in

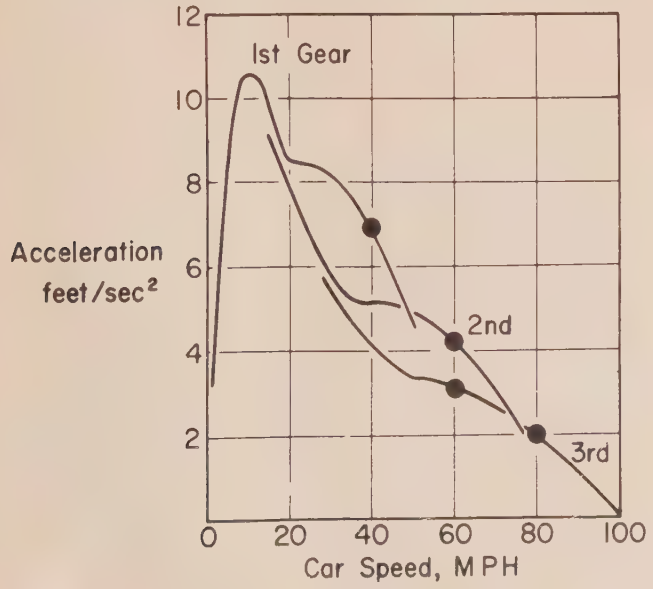


Fig. 45—Acceleration performance

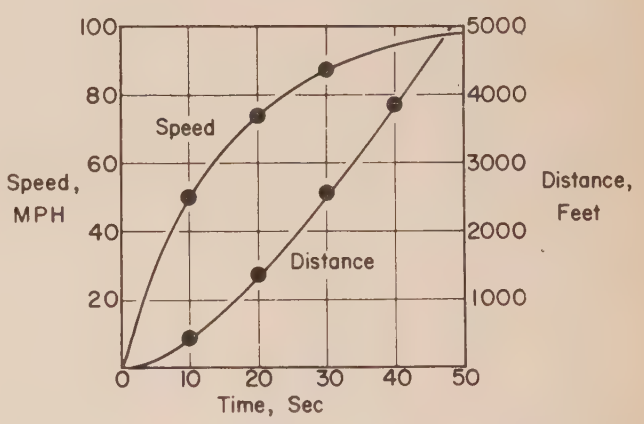


Fig. 46—Speed and distance versus time for full-throttle acceleration

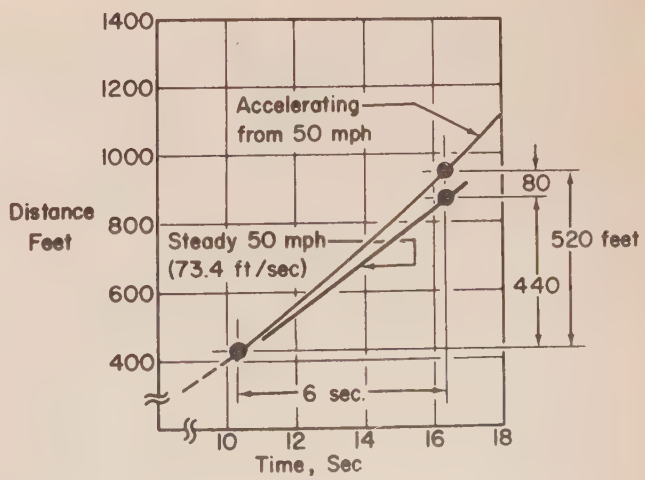


Fig. 47—Passing time and distance when both cars are initially traveling 50 mph

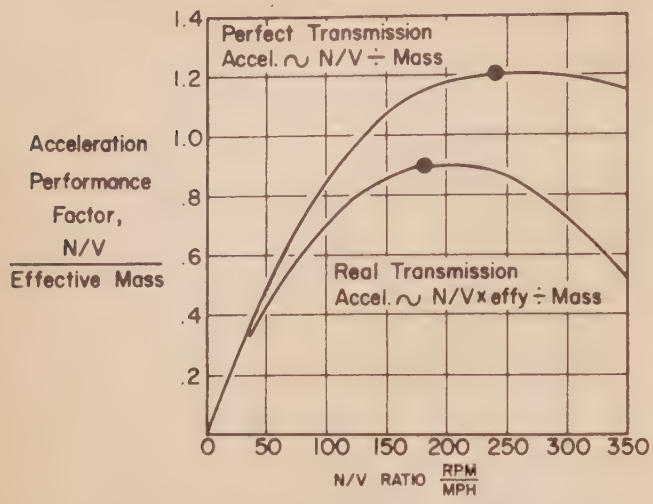


Fig. 48—Relation between acceleration and overall ratio

Fig. 49—Warm foggy day in Denver

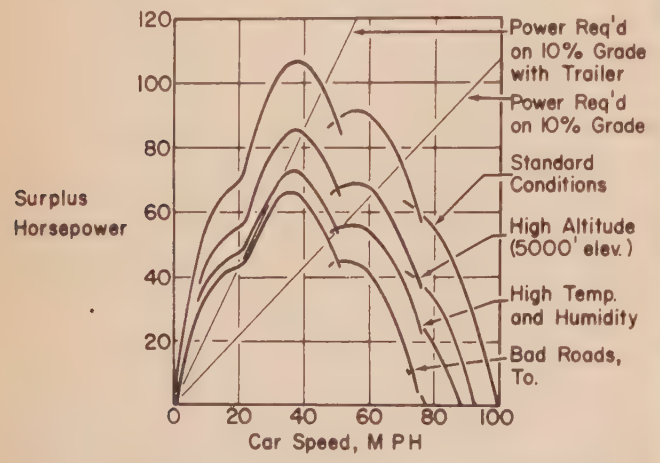


Fig. 50—Effect of operating conditions on available surplus power

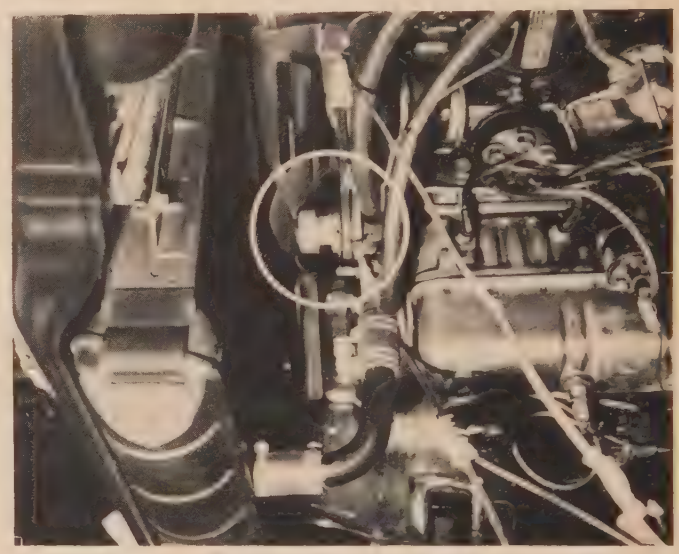


Fig. 51—Fan torquemeter

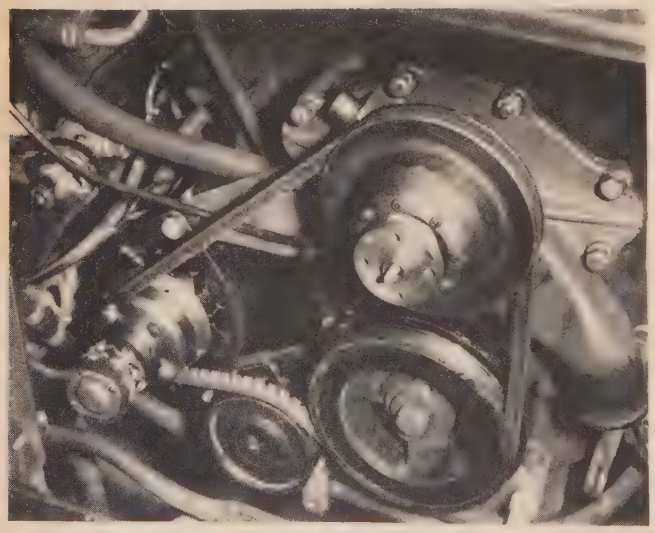


Fig. 52—Generator, pump torquemeter

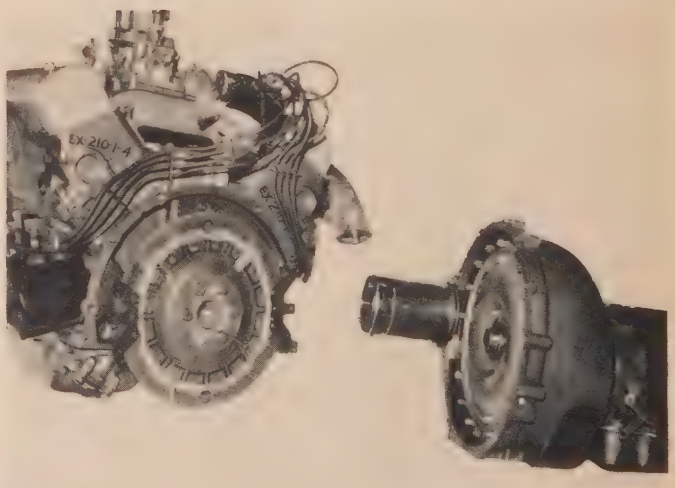


Fig. 53—Flywheel torquemeter

direct proportion to the speed ratio—or something less, if our transmission is not 100% efficient. If we start increasing ratios, we soon reach a point where a further increase adds more to mass than it does to mechanical advantage, and we lose rather than gain. We saw this in Fig. 44, where, below 10 mph, mass was rising faster than available force, and acceleration dropped. Acceleration, at least with a perfect transmission, should increase with mechanical advantage, or N/V ratio. Likewise, it should be inversely proportional to the total effective mass. If, therefore, we divide mechanical advantage, or N/V ratio, by total mass, we will get a quotient which we can call an acceleration performance factor. Using this quotient, we can explore how acceleration performance will vary with our choice of the overall ratio, which affects both mechanical advantage and total mass. Fig. 48 shows that with a perfect transmission, an overall ratio of 240 to 1 is as high as we should go. At this ratio, the rotating inertias in the average automobile account for half the total mass, and a further increase would hurt rather than help acceleration performance. The lower line in Fig. 48 shows the behavior of our real transmission, in which torques are not doubled when output speeds are halved. The net result of the transmission efficiency is to reduce the N/V ratio for peak performance to 180 to 1 in this case. Since our rear axle and wheels gave us an N/V of 40, any reduction above 4.5 to 1 in our transmission and converter will not help us on acceleration. We have evidently used a little more ratio than is desirable in low gear on our hypothetical car. In this respect, we are no different from many automobile makers today who have reached or passed the ratio in the lower gears of their cars which can be used to gain acceleration. Improved performance must be sought by increasing engine torque and using less overall reduction in axle and transmission. This combination not only gives better performance, but quieter and more economical operation.

So far, we have only considered performance under standard, average conditions. This is reasonable, not only because conditions are, on the whole, average, but also because many of the variations cause compensating, or partially compensating, effects. Thus, high temperatures and low pressures which reduce an engine's breathing capacity, also reduce running friction and wind resistance. However, occasionally factors can stack up against us.

Here's a fellow in real trouble (Fig. 49). It was a warm foggy day in Denver when he set out with other members of his football team for a short vacation. As has been shown, engine output is down as much as 18% at an altitude of 5000 ft. When our driver gets to the top of the pass, it will be down twice that much, and his engine will need an oxygen mask.

Fig. 50 shows what the drop in power looks like. Altitude takes the biggest chunk. When he turned his air conditioner on, his underhood temperature rose to nearly 200 F, and the output fell some more. The wet gravel road used up some more. By running full throttle in first gear, he can barely haul his trailer up that 10% grade—at 10 to 20 mph—if he stops every few feet to cool off. Now we know why they put that extra ratio in the transmission.

Checking Losses on the Road

The next few figures are included to illustrate some of the methods the test engineer uses to check the performance of actual vehicles against the projections of the design engineers. To check power costs of the various accessories, we have fan and generator torquemeters (Figs. 51 and 52). We use flywheel and driveshaft torquemeters to check on the costs of an automatic transmission under steady and transient conditions on the road (Figs. 53 and 54). Driveshaft and wheel torquemeters measure what happens in the rear axle, and wheel torquemeters and towing dynamometers show what it takes to transmit power through tires (Figs. 55 and 56), as well as

Fig. 54—Driveshaft torquemeter

Fig. 55—Wheel torquemeter

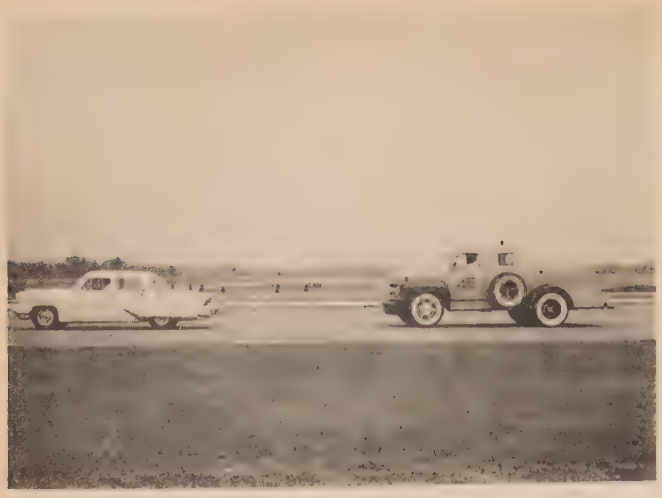


Fig. 56—Drawbar dynamometer

Fig. 58—Fuel burette

Fig. 57—Acceleration chronograph

measuring what it takes to move the car down the road. The methods of separating tire, wind, and bearing friction from each other are somewhat involved and beyond the scope of this paper.

But what the customer gets is measured by our drawbar scale readings—which can be interpreted in terms of grade capability, by our acceleration timers (Fig. 57), which show getaway and passing performance and with the fuel burettes which measure the economy, or total cost (Fig. 58).

Lest someone think that our hypothetical car has no relation to real automobiles, we've searched through our test data, and, as luck would have it, were able to find four vehicles, which, when averaged, gave us within 1% of the advertised power, within 2% of the displacement, and within 3% of the test weight. In case you've wondered, that's what the little dots are on the various performance curves which were developed during this panel meeting. They are test points which show how closely a real hybrid car of nearly identical specifications fits the hypothetical vehicle put together here this afternoon.

If any of you here have a Ford car with a Nash engine, Chrysler transmission, Chevrolet chassis, body by Fisher, and it doesn't perform like this—see your dealer.

What has all our power gotten us? Well, a short time ago we had occasion to run a 30-year-old car against one of our latest models—and were accused of cheating. Not only did the new model out-accelerate and out-stop the early model—but it was more economical as well. Of course, we did cheat a little bit. We ran our economy test at 40 mph, which was full throttle, and as fast as our antique car could go using all its 23 hp. At this speed, the 1956 car was loafing along—silently—using only 15 hp. Those of you who think the old Model T had more head room in its 6 ft 10½ in. of height would be fooled; if you think its 35-sq ft frontal area allowed roomier seats, then you've forgotten they were only 36 in. wide. If you feel the old car was easier to get into, you don't remember that the floor was as high as the top of the wheels, and access through the driver's door wasn't easy, because there wasn't any driver's door.

In the last 30 years progress has been made. The average mileage accumulated by an automobile during its lifetime has increased almost five times. The miles traveled annually in this country have quadrupled. The number of vehicles on our roads has tripled. The average life, in years, of these vehicles has doubled. The average speeds on our main rural roads have risen—not quite doubled. The total mileage of roads and streets has changed hardly at all—although these roads and highways have been vastly improved. The accident rate has been more than halved—although the total number of accidents has remained fairly constant over the last 20 years.

Where there are trends, they are steady, and we

expect they will continue. We believe, with the exception of the accidents and the miles of roads, that these trends indicate the customer got what he wanted. The evidence suggests that certainly in the matter of increasing the capacity and utility of our roads and even in the matter of accidents, the increased power and performance of cars has helped reduce the price we pay for travel.

APPENDIX

Performance Projection Calculations

Table 1—Acceleration Calculations

Speed, mph	Engine rpm	Driveshaft rpm	Driveshaft Torque, lb-ft	Wheel Thrust, lb	Roll Resistance, lb	Surplus Tractive Effort, lb	Surplus Tractive Power, hp	$\frac{N}{V}$	Mass Factor	Total Effective Mass, slugs	Acceleration, ft/sec^2
					First Gear						
0	1635	0	1175	3150	90	(3070)[a]	0	∞	∞	∞	0
5	1650	200	1010	2710	75	(2635)[a]	30.0	330	2.80	348	7.6
10	1800	400	800	2140	75	2035	54.2	180	1.56	194	10.5
15	2000	600	635	1700	80	1620	64.8	133	1.32	164	9.9
20	2275	800	515	1380	85	1295	69.0	114	1.24	154	8.4
25	2650	1000	510	1370	95	1275	84.7	106	1.21	150	8.5
30	3075	1200	495	1330	105	1225	98.0	103	1.20	149	8.2
35	3550	1400	465	1250	115	1135	106.0	101	1.19	148	7.7
40	4050	1600	420	1130	130	1000	106.5	101	1.19	148	6.8
					Second Gear						
20	1850	800	450	1210	85	1125	60.0	92	1.16	144	7.8
25	2000	1000	390	1050	95	955	63.6	80	1.13	140	6.8
30	2150	1200	340	910	105	805	64.4	72	1.11	138	5.8
35	2350	1400	310	830	115	715	66.7	67	1.10	137	5.2
40	2550	1600	310	830	130	700	74.6	64	1.09	135	5.2
45	2800	1800	310	830	145	685	82.0	62	1.08	134	5.1
50	3050	2000	305	815	160	655	87.2	61	1.08	134	4.9
55	3325	2200	300	805	180	625	91.6	60	1.08	134	4.7
60	3600	2400	285	765	200	565	90.4	60	1.08	134	4.2
65	3900	2600	265	710	225	485	84.0	60	1.08	134	3.6
70	4200	2800	245	655	250	405	75.5	60	1.08	134	3.0
75	4500	3000	220	590	285	305	61.0	60	1.08	134	2.3
					Third Gear						
30	1875	1200	310	830	105	725	58.0	62	1.08	134	5.4
40	2075	1600	255	685	130	555	59.1	52	1.07	133	4.2
50	2300	2000	225	605	160	445	59.3	46	1.06	132	3.4
60	2600	2400	225	605	200	405	64.8	43	1.05	130	3.1
70	2925	2800	220	590	250	340	63.5	42	1.05	130	2.6
80	3300	3200	215	575	315	260	55.5	41	1.05	130	2.0
90	3700	3600	200	535	390	145	34.8	41	1.05	130	1.1
100	4100	4000	180	480	480	0	0	41	1.05	130	0

$$\text{Thrust} = \text{Driveshaft Torque} \times 3.2 \text{ Axle} \times 0.94 \text{ efficiency} \times \frac{12 \text{ in.}}{13.5 \text{ Tire Radius}} = \text{Torque} \times 2.68$$

[a] Limited by tires to 2250 lb on steady pulls (0.94 friction, 24-in. c. g. height, 115-in. W. B., 48% rear weight distribution)

Table 2—Full-Throttle Speed—Time and Time Distance

Speed, mph	ΔV, fps	Average Acceleration ft/sec^2	ΔT, sec	Σ, time	Speed at Time	ΔT, sec	Average mph	Average fps	ΔS, ft	Σ, ft	Time, sec
0–10	14.7	7.6	1.93	1.93	10	0–5	15	22.0	110	110	5
10–20	14.7	9.9	1.49	3.42	20	5–10	40	58.7	293	403	10
20–30	14.7	8.5	1.73	5.15	30	10–15	57	83.6	419	822	15
30–40	14.7	6.4	2.29	7.44	40	15–20	69	101	505	1327	20
40–50	14.7	5.1	2.88	10.32	50	20–25	77.5	114	570	1897	25
50–60	14.7	4.7	3.12	13.44	60	25–30	84	123	615	2512	30
60–70	14.7	3.3	4.45	17.89	70	30–35	89	131	655	3167	35
70–80	14.7	2.3	6.38	24.27	80	35–40	92.5	136	680	3847	40
80–90	14.7	1.6	9.17	33.44	90	40–45	95	139	695	4542	45
90–100	14.7	0.6	24.45	57.89	100	45–50	97	142.5	713	5255	50
						50–55	98.5	144	720	5975	55

DISCUSSION

Summarized Distribution Of Energy in Engine Performance

—W. S. James

William S. James & Associates

THE following is a brief review of the material presented in the symposium in which an attempt is made to summarize the various ways in which the working gas pressure developed in the combustion chambers was used.

As Messrs. Kosier and McConnell said in their section of the symposium, "Spending energy is like spending money . . . we pay to get something we want." Following out this line of thought one of the first questions asked might be "How much energy do

we have to spend?" Although there may be some differences of opinion on the answer to this question, a reasonable estimate can be made from the excellent work of Hershey, Eberhardt, and Hottel on the "Thermodynamic Properties of the Working Fluid in Internal-Combustion Engines" which appeared in SAE Transactions in 1936. When the rather complicated relations between such things as gas temperature, dissociation, and change in specific heats with temperature and pressure are allowed for, they estimate a mean effective pressure of 225 psi would be the theoretical maximum for an engine with 8.5 compression ratio and the cylinders charged with a combustible mixture of about 91% of the theoretical amount of dry air required for the chemical correct combustion, at 14.7 psi and 60 F. This means that the net average (mean) ordinate of the work area of the indicator card would be 225 psi. It assumes 100% volumetric efficiency, zero heat loss to the jackets, and complete combustion of all the air in the cylinders. This could be considered as Aunt Jemima's legacy. We know it could hardly be more than a certain amount, but we also know we had best not plan on spending as though we were going to get it all. This value of maximum theoretical mean effective pressure is shown by the horizontal line at the top of Fig. A (225-psi mep at 4000 rpm in a 300-in. engine will give 340+ hp).

If the engine is put on the dynamometer and run equipped as it will be when installed in the car and supplied with air at the temperature, pressure, and humidity that it will receive when under the hood of the car, it will indicate on the dynamometer a torque which will correspond to a mep as shown on the fifth curve from the top, labeled "mep at flywheel." The mep indicated on this curve represents that portion of the total theoretical energy available in the cylinders that finally leaves the engine as mechanical energy and is passed on into the transmission.

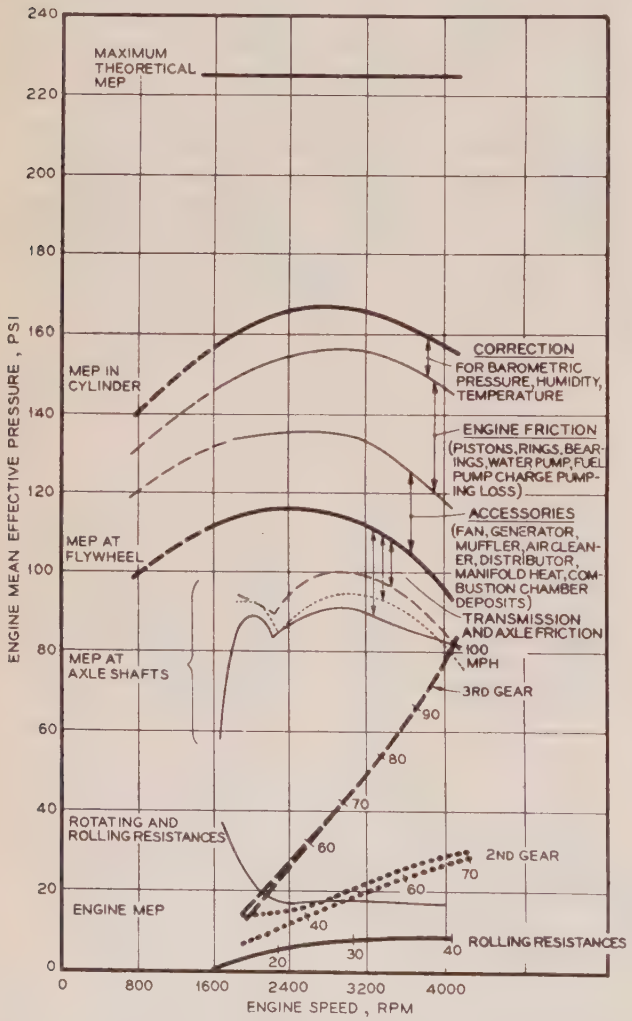


Fig. A—Engine mean effective pressure versus engine speed

What has happened to the 340+ hp? It is down to 145+ hp! Nearly 200 hp lost? No, not lost just spent. But for what?

The "accessories" namely the radiator fan, generator, muffler, air cleaner, manifold heat, combustion-chamber deposits, and distributor advance curve take about 33 hp. The engine friction, including the piston and ring friction, the water pump, the fuel pump, the bearing friction, and the charge pumping loss take about 44 hp. The correction for having air entering the engine at 29.00 instead of 29.92 in. of Hg, its temperature being 100 instead of 60 F and the humidity being the equivalent to 0.40 instead of 0.0 in. of Hg accounts for about 16 hp. These items together with the brake horsepower of 145+ total to 238 hp which can be accounted for. The difference between this 238 hp and the 340 hp theoretically possible, or 102 hp, is the result of heat loss to the cylinder walls, volumetric efficiency lower than 100%, and incomplete combustion of the air in the cylinders. The actual effective mep has shrunk from the theoretical possible of 225 psi to a maximum of 117 mep at 2400 rpm and 96 psi at 4000 rpm.

So far we have spent quite a little of our energy on "accessories," but the engine would not run well without these expenditures. We have also spent energy on overcoming engine friction, certainly a necessary expenditure. Then we have spent some on correcting for differences in barometric pressure, atmospheric humidity, and ambient temperature. This is like saving against a rainy day. The rest, the 102 hp going to such as the cylinder walls and incomplete combustion, is a kind of inheritance tax on Aunt Jemima's legacy. Some day this tax may be reduced.

What is the horsepower of the engine: 340, 238, or 145? Possibly it all depends on how you look at it.

Now let's look at what happens in the transmission and axle. Directly below the curve of mep at the flywheel are three curves, one solid, one dotted, and the other dashed. These are the three curves showing the cylinder pressures equivalent to the torque reaching the axle shafts. The solid line represents the mep reaching the wheels when the transmission is in first or low gear. The two solid lines near the bottom of the plot are the cylinder mep required to overcome the rotating inertia of the engine and driving train and also the rolling resistance of the vehicle (tire losses, air resistance, and the like). The upper of the two solid curves is the sum of the rotating and rolling resistances of the car and its driving train when in low gear. The lower curve shows the rolling resistance alone. The three figures along the lower curve 20, 30, and 40 indicate the road speed of the car when in low or first gear and the corresponding engine rpm. It will be noted that below 20 mph the inertia resistance is much greater than the rolling resistance. It will also be noted that at wide-open throttle the rpm of the engine is never less than 1650. This is the full-throttle "stall speed" of the torque converter.

The distances between the curve of mep at the axle shafts and the rolling and rotating resistance curves represents the cylinder pressure remaining to accelerate the car or push it up hills. The shorter distance represents the cylinder pressures available for car acceleration on a level road. The longer distance is the cylinder pressure available for hill climbing at a constant speed.

The similar sets of curves for second gear and third or high gear are shown in dots for the second gear and dashes for the direct or high gear. The curve for high gear mep at the axle shafts and that for the rolling resistance in high gear intersect at 100 mph, about 4100 rpm and 84 psi, or about 130 hp. We have spent about 15 hp in the transmission and axle.

As Kosier and McConnell show in their paper the power available for acceleration to give agility to the car in traffic, on hills, and for towing is over 100 hp in first gear, 90 hp in second, and over 60 hp in direct. Their work gives a very clear picture of the performance requirements of the present-day passenger car.

Aunt Jemima's legacy is spent. If we have bought what we wanted was it wasted?

Horsepower race? Possibly. It might also be providing a very necessary requirement for a nimble car.

The General Motors Fuel Injection System

John Dolza, E. A. Kehoe, Donald Stoltman, and Zora Arkus-Duntov, General Motors Corp.

This paper was presented at the SAE Annual Meeting, Detroit, Jan. 15, 1957.

THE carburetor expert knows the difficulty of properly metering fuel with a carburetor to satisfy all engine conditions: speed, load, temperature, altitude, operating transients, and fuel variables, but with patience he may achieve a fairly good "flow curve" or "calibration curve." Next, the air and fuel have to be mixed as uniformly as possible either within the carburetor or between carburetor and center portion of manifold, commonly called the "riser." The more or less heterogeneous mixture must then be carried through the intake manifold to the individual cylinder ports. Transporting the mixture to the intake ports with a minimum of fuel separation requires (1) maintaining a high flow velocity in the manifold by using small passages and (2) adding heat for further fuel vaporization. At best, this combination leaves some lean cylinders. The overall mixture must then be richened to assure adequate fuel for the leanest cylinder.

The resulting average carburetor and manifold system may then have the following inherent deficiencies:

1. It may use 10% more fuel at wide-open throttle than it should. This not only represents a waste of fuel but also exhausts to the atmosphere annoying amounts of carbon monoxide.

2. Cold starting and warmup may require substantial amounts of fuel to provide the initial wetting of the manifolds, and still have left adequate liquid and vaporized fuel for the cylinders to insure firing. This adversely affects fuel economy and increases air pollution.

3. The hot-spot manifold required by carburetors results in (1) loss of bowl fuel through evaporation because of the increased carburetor temperature, (2) promotion of vapor lock, and (3) some octane depreciation of the fuel because of increased mixture temperature.

4. Manifolds have to be a compromise for volumetric efficiency and good distribution and may not permit full utilization of the power potentiality of the engine.

5. Decreasing manifold vacuum during throttle opening transients increases the manifold wetting. For a smooth engine feel during the transient, additional fuel is required as "make-up" for the increased wetting. This "make-up" fuel supplied by the accelerator pump may, under certain conditions, represent

THIS paper discusses the fuel injection system which General Motors has been working on. According to the authors it will give greater simplicity, economy, and reliability than the conventional carburetor.

Basic development work was done on single-cylinder units. Air and fuel metering methods are discussed for best performance during such operations as warmup and idle.

Basic components of the fuel injection system such as the filter and pump are described in detail for their design and performance. Comparison with performance of conventional carburetors is also made.

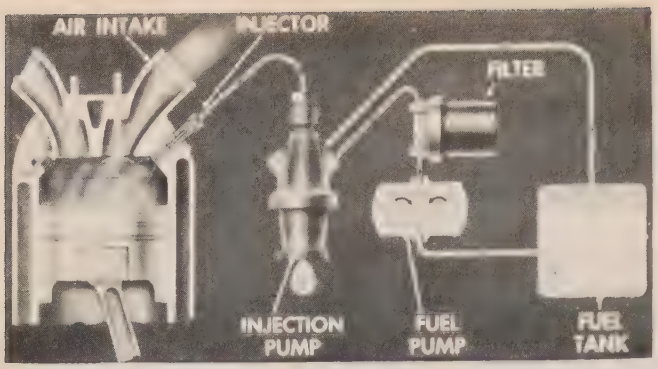


Fig. 1—Aircraft injection system

a loss because this fuel is mainly exhausted as unburned hydrocarbons.

A fuel system capable of correcting even 50% of the average carburetor deficiencies may produce a 10% improvement in fuel economy that would ultimately result in substantial savings to the customer.

The possibility of an exhaust free from hydrocarbons and a fuel system free from evaporation would greatly help in an important phase of reducing air pollution.

It was with these goals and good intentions in mind that a development program was initiated at the General Motors Engineering Staff several years ago toward achieving the best possible fuel system.

Early Fuel Injection Systems—Since the aircraft engine had the most exacting carburetion requirements, this industry was first to recognize the carburetor's limitations and to develop new systems.

One of the earliest successful systems consisted of injecting fuel into the combustion chamber (Fig. 1). A diesel-type injection pump was provided with metering controls to obtain the air-fuel mixture required for spark ignition engines. This system did a good job of fuel distribution in the rather narrow cruising and full power range of aircraft engines. This injection system not only overcame the difficult distribution problem on radial aircooled engines, but also eliminated the fire and explosion hazard created by the combustible mixture in the manifold and supercharger of carbureted radial and V-type aircraft engines.

Subsequent development work indicated that:

1. The design of the nozzle could be simplified and its reliability enhanced by injecting fuel into the intake ports rather than into the combustion chamber.
2. No appreciable power difference existed between port and combustion chamber injection.
3. The use of individual plungers for each cylinder was sufficiently accurate for aircraft engines, which operate at fairly substantial bmep, but was very poor for automotive engines at idle and city driving because of poor distribution characteristics inherent to this system at low fuel rates.
4. The system, even with volume production, was too high priced for general use. At the time of our early work it was possibly eight times as expensive as the carburetion system.

Another interesting system that achieved a great deal of attention by Italian manufacturers was the Fuscaldo method (Fig. 2). This system consisted of a source of fuel under pressure at the engine intake ports. Electromagnets opened precision-made valves at each intake port to deliver fuel in relation to engine requirements. While the system offered some simplification over the diesel-type pump and injector, it did not appeal to us as the ultimate in simplicity and reliability.

After a thorough analysis it became evident to us that none of the existing fuel injection systems could compete with carburetors for simplicity, low cost, and low-speed performance and at the same time correct the mentioned deficiencies. Therefore, we decided to work out our own system.

Fuel Injection System

Single-Cylinder Tests—The basic development work was done on single-cylinder units. From these test results obtained at a compression ratio of 8/1 (Fig. 3), we established that:

1. Direct injection into the combustion chamber had no appreciable advantage over injection into the intake ports.
2. Intermittent (timed) injection in the intake ports gave slightly less power and used slightly more fuel than continuous injection.
3. Injection directed toward the intake valve gave the most power, the best economy, the fastest warm-up, and the best acceleration response.

Based on these facts, we chose continuous-flow injection into the intake ports with open orifice nozzles directing the fuel toward the intake valves. Further development work covered extensively the function of every component and represented successive steps of simplification of design, improving metering means, and increasing reliability. I shall not take time to trace this development work. Instead, I believe a short discussion of metering in general would be most helpful before describing the operation of our fuel injection system in detail.

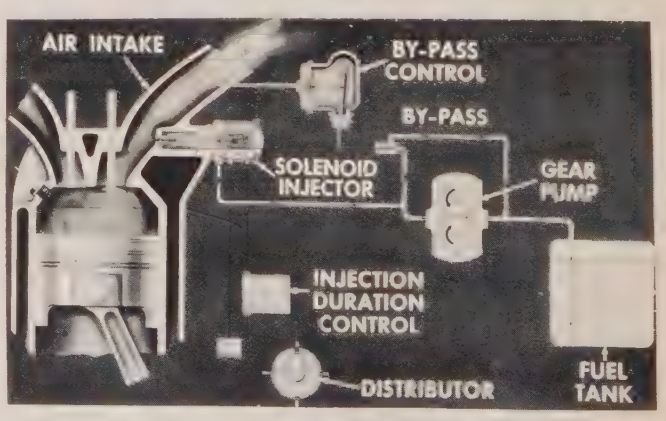


Fig. 2—Fuscaldo injection system

Metering Methods (General)—Two methods of air and fuel metering were considered for our continuous-flow fuel injection system. These methods can be classified as speed-density metering and mass flow metering.

1. Speed-Density Metering. The fuel injection systems used for aircraft were controlled by means of engine-speed air-density responsive metering systems. This choice may have been dictated by the wide air-density range encountered in flight. The sensor of pressure and of temperature must be correct in proportion to absolute temperature and absolute pressure.

An ambient temperature change from −30 to 100 F or 430 to 560 R, requires a metering change of $560/430 = 1.30$ or 30%. This means that air at −30 F requires 30% more fuel than the same volume of air at 100 F. A similar large change would be shown versus atmospheric pressure (as climbing Pike's Peak).

The speed-density metering system requires a device which will: (1) sense absolute temperature and pressure accurately and (2) correlate these absolute values to the metering means. These requirements make the sensing device costly. Furthermore, this type of metering requires means to measure engine volumetric efficiency versus speed which, on "rammed engines," may prove to be rather complicated and costly.

The speed-density metering system usually requires a supply of fuel in equal amount per cycle. The fuel pump must therefore be a uniform-flow pump which may not depreciate in service.

2. Mass Flow Metering. Let us now examine the method of metering in function of mass of air consumed by the engine, in other words, by using a high efficiency venturi. The temperature and pressure effect have to be corrected only in function of the square root. Consequently, a temperature change from 430 to 560 R causes a metering change of only $\sqrt{560/430} = \sqrt{1.30} = 1.14$ or 14% as compared to 30% in speed-density metering. Actually, fuel-density variations with temperature cancel 4% of this change, and proper utilization of fuel vapor bubbles at high temperatures tends to cancel a large part of the remaining 10%.

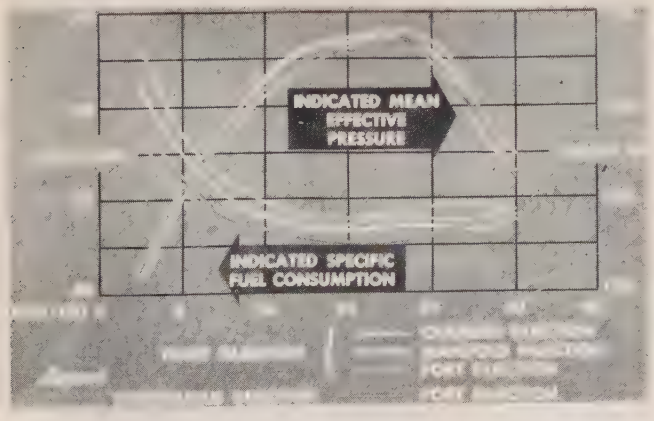


Fig. 3—Comparison of timed and continued injection

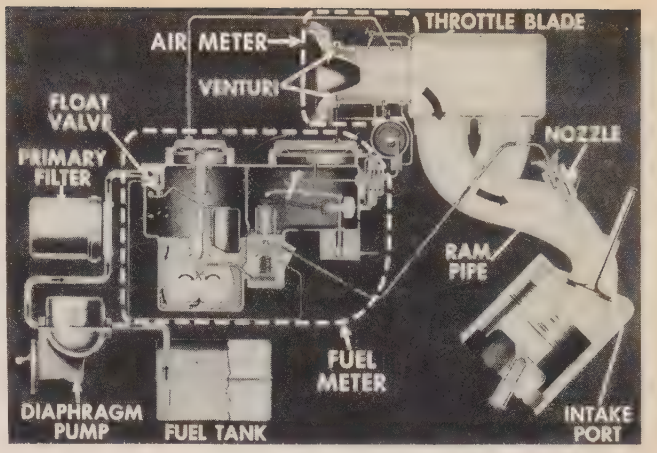


Fig. 4—General Motors injection system

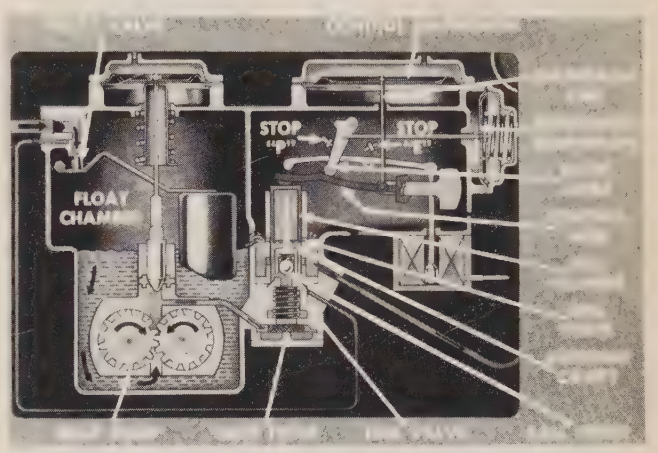


Fig. 5—Fuel meter

The volumetric efficiency of the engine is sensed by the venturi, and no additional correcting signal is required. The pump delivery does not have to be constant (as is the case with the speed-density system) as long as it is in excess of the desired flow at the nozzles.

The control method selected for our injection system is based on mass flow metering in which venturi throat depression is related mechanically to fuel pressure. (See Appendix.)

Metering System—Fuel supplied by a conventional 6-psi diaphragm pump (Fig. 4) passes through a 10-micron primary filter to the fuel meter. The fuel is admitted through a float valve—similar to its counterpart in a carburetor—into the float chamber where any vapors which have formed due to temperature rise are vented. A small gear pump located in the bottom of the float chamber delivers the fuel through a second inlet filter and a fuel valve to the metering cavity (Fig. 5). The fuel valve contains an antipercolation ball check to keep the fuel between the valve and the pump at a sufficiently high pressure to elimi-

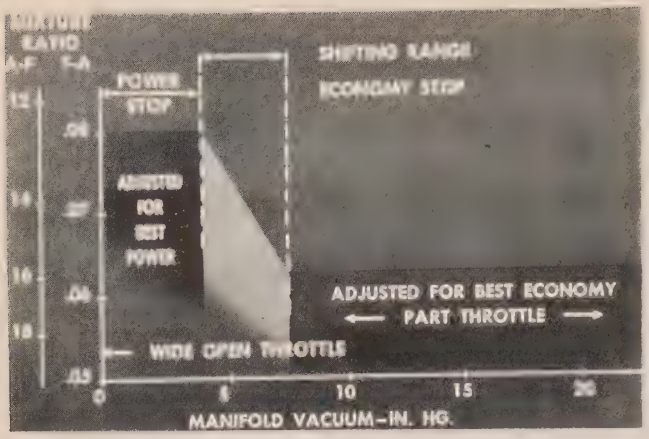


Fig. 6—Operating mixture ratios

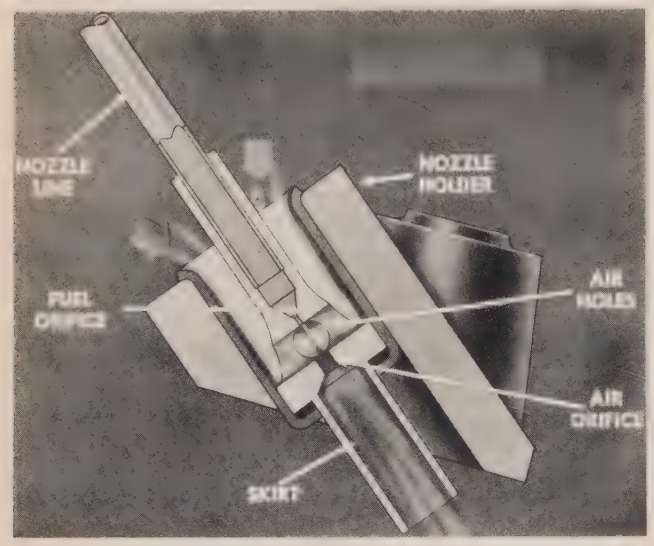


Fig. 7—Injection nozzle

nate vapor pockets. Some of the fuel delivered to the metering cavity flows directly to the eight continuous spray nozzles; the remainder of the fuel flows through the spill ports back to the float chamber. The amount of fuel spill is regulated by the spill plunger.

The air enters the air meter through the annular venturi section and flows past the throttle blade to the manifold header and then to the individual intake ports. The air velocity through the venturi causes a depression signal related to the mass airflow. This venturi signal acts on the control diaphragm to create a force which is transmitted to the spill plunger by means of the diaphragm link and control arm, the latter being pivoted on the ratio lever.

An increase in airflow through the venturi causes a relative increase in venturi signal which, acting on the control diaphragm, results in an increase in the force acting on the top of the spill plunger. The spill plunger then moves to a new balanced position to obtain a fuel pressure increase proportional to the venturi signal increase. Since the increased fuel pressure results in a fuel flow proportional to the increase in airflow indicated by the venturi signal, a

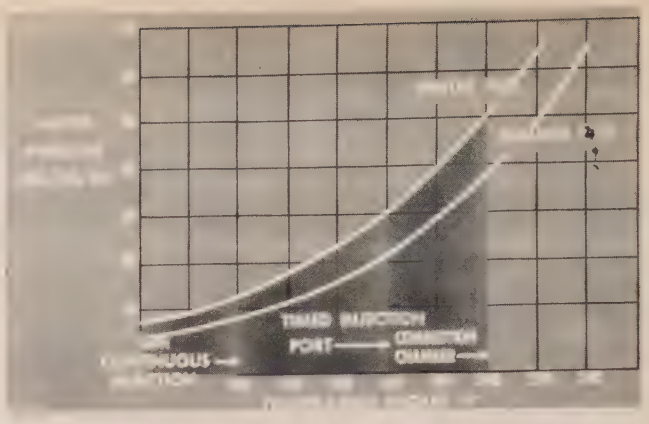


Fig. 8—Average nozzle operating temperatures

constant air/fuel ratio will be maintained as long as the linkage ratio is not changed.

The air/fuel ratio can be varied by changing the linkage leverage between the control diaphragm and the spill plunger. This is accomplished by shifting the pivot point on the control arm. When the "ratio lever" rests against stop "P" (Power), the air/fuel ratio obtained is for wide-open throttle operation; (Fig. 6) when resting against stop "E" (Economy), part-throttle fuel requirements are fulfilled. For automatic operation, the ratio lever is controlled by the spring-loaded enrichment diaphragm which is subjected to manifold vacuum; at light load (high vacuum) the lever is held at position "E," at full load (low vacuum) at position "P."

The stops are set to obtain a best economy air/fuel ratio (about 15.5/1) at part throttle and a maximum power air/fuel ratio (about 12.5/1) at wide-open throttle. The enrichment diaphragm spring is set to hold the ratio lever in the economy position above about 7 in. of Hg vacuum.

The nozzle (Fig. 7) has an accurately calibrated open-fuel orifice somewhat less than 1/64-in. diameter. The nozzles are mounted, at the entrance of each intake port, in plastic nozzle holders. Just below the fuel orifice, there is a small air chamber which receives filtered air through four 0.100-in. diameter air holes from the air cleaner. This air supply assures that the nozzle discharge is at all times near atmospheric pressure regardless of manifold vacuum fluctuations, and consequently the amount of fuel injected depends solely on the metering system pressure. In line with the fuel orifice across the air chamber is the opening to the intake port. This is a 0.040-in. diameter air orifice. It should be noted that under all manifold vacuum conditions the atomization of fuel is enhanced by the mixing of air and fuel through this 0.040 orifice. The air volume passing through the air orifice represents about one-fourth of the air used at closed throttle for idling.

The tubular skirt below the air orifice is cooled by evaporation of the fuel which in turn cools the fuel line by conduction, thus preventing metering disturbances caused by fuel vapor bubbles. In fact, the

fuel at the nozzle and in most of the line is at a lower temperature than when it entered the system from the diaphragm pump.

By using this scheme of refrigerating the fuel by means of its latent heat, this open-orifice type of nozzle becomes suitable for handling high vapor pressure fuel even under summer conditions. The nozzle operating temperature range of this nozzle, due to its location and insulation, is much lower than that of nozzles used in other fuel injection systems (Fig. 8). This lower temperature range greatly reduces vapor problems.

The fuel orifice size was selected to give a minimum pressure at idle and cranking of about 8-in. fuel head and a maximum pressure of the order of 200 psi at engine top speed and wide-open throttle. Under normal driving conditions, the pump pressure seldom exceeds 20 psi (Fig. 9). These moderate pressures can be achieved with a plain gear pump.

The fuel pressure at idle and when starting has been found sufficiently high to prevent adverse distribution effects caused by normal engine attitude changes.

Idle Operation—During idle operation the throttle is closed and about one-fourth of the air requirement is supplied through the individual nozzle air chambers. The rest is provided through a passage around the throttle blade and is regulated by the idle-air bypass screw (Fig. 10) within the throttle body. The correct idle mixture is set by means of the idle mixture screw. This screw controls the amount of venturi signal boost caused by bleeding in manifold vacuum from the idle orifice.

Off-Idle Enrichment—To give best economy for steady-state operation at idle and off-idle, the engine requires a mixture which is richer than for higher speeds (Fig. 11). The basic venturi system results in a leaner mixture at low airflows but, because of the plunger weight and the nozzle placement below the spill level in the fuel meter, there is some inherent enrichment near idle. The remainder of the required idle mixture enrichment is supplied through the idle orifice, and the remainder of the enrichment required in the off-idle range is supplied through the off-idle orifice (Fig. 12).

As the throttle is opened from the idle position, the off-idle orifice is subjected to manifold vacuum, thereby intensifying the venturi signal. The amount

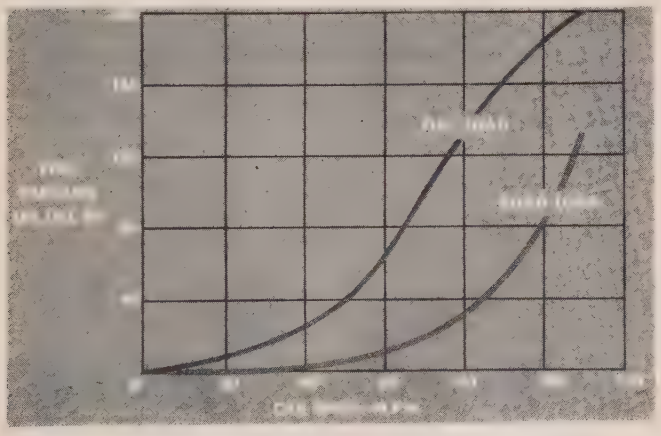


Fig. 9—Fuel pressure versus car speed

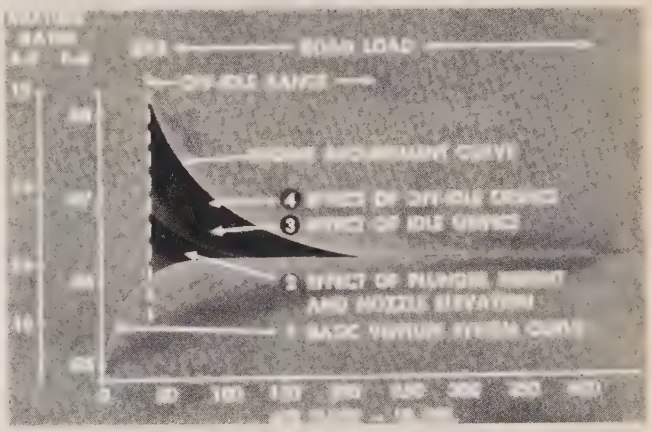


Fig. 11—Idle and off-idle mixture ratios

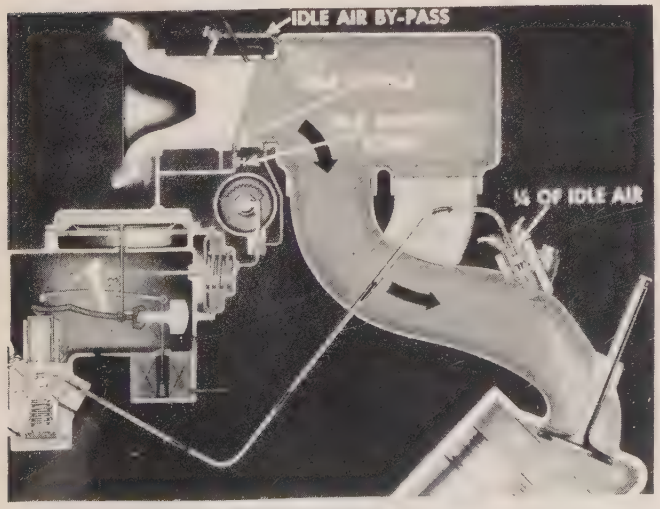


Fig. 10—Idle operation

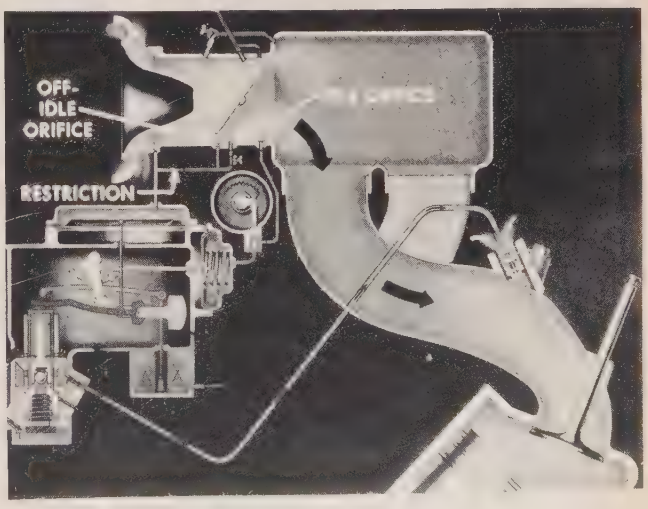


Fig. 12—Off-idle enrichment

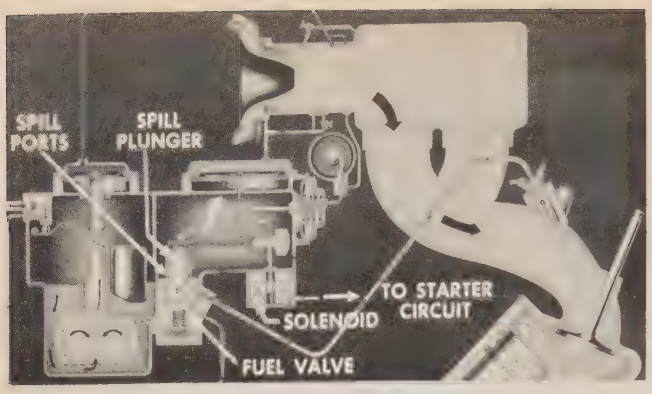


Fig. 13—Cold starting

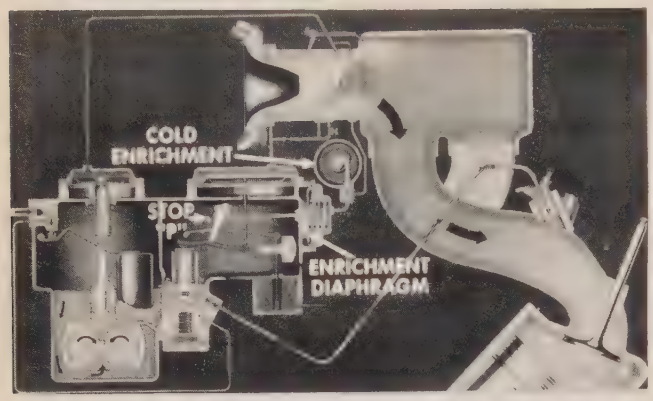


Fig. 14—Warmup enrichment

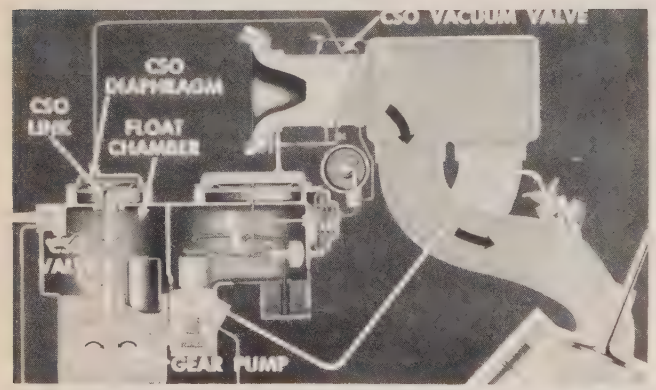


Fig. 15—Coasting shut-off

of off-idle bleed is regulated by the restriction in the bleed passage.

Transients—Our fuel injection system does not require an accelerator pump to supply extra fuel during throttle opening transients. Two factors are responsible for the elimination of the accelerator pump: (1) manifold wetting is kept to a minimum by port injection and (2) instantaneous fuel meter response. Upon throttle opening, the manifold is filled by an immediate inrush of air greater than the stabilized air requirement, causing an equivalently larger venturi signal. Since the fuel meter linkage parts are very light and the travel small, their inertia is negligible so that the inrush of air will instantly cause a rise in fuel

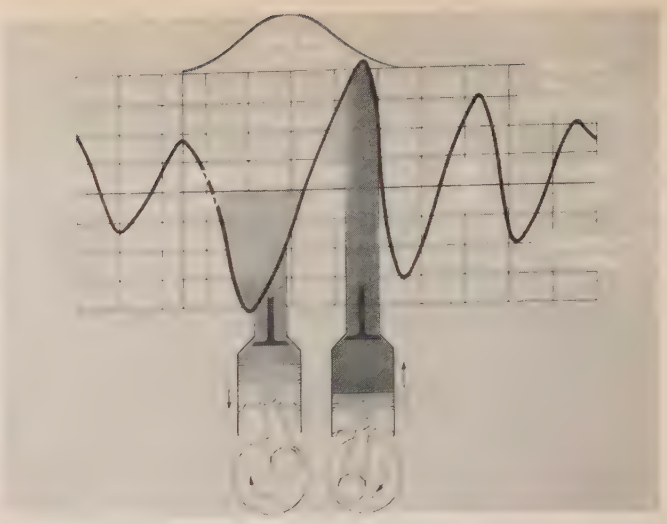


Fig. 16—Dynamic supercharging

pressure. The result is a shot of fuel proportional to the inrush of air.

Cold Start and Warmup—Because of the direct distribution of the fuel to the intake ports, it is not necessary to provide a substantial fuel enrichment to wet and flush the manifold as is the case with a carburetor. The starting enrichment required is achieved by means of a solenoid connected to the starter circuit. This solenoid (Fig. 13) forces the plunger down far enough to unseat the fuel valve. This action results in: (1) completely cutting off the spill by covering the spill ports and (2) uncovering a passage from the float chamber inlet to supply extra fuel directly from the diaphragm pump.

During warmup the cold enrichment mechanism keeps the ratio lever at the power stop position (Fig. 14) by cutting off manifold vacuum to the enrichment diaphragm. This enrichment assures solid steady-state engine operation during warmup. By directing the fuel stream upon the intake valve, the incoming mixture is warmed up considerably faster than by means of the "hot spot" of the intake manifold of the carbureted engine. A fast idle cam keeps the engine idle speed high enough to eliminate stalling by slightly opening the throttle blade during warmup.

Coasting Shut-Off—The fuel injection system operates with dry manifold, and consequently, the amount of fuel vaporization in the manifold during deceleration at closed throttle is negligible. A further step to eleminate exhaust fumes containing unburned fuel during coasting is provided by a "coasting shut-off" (CSO, Fig. 15) located directly above the gear pump outlet, and consisting of a valve, which under normal operating conditions is closed by a spring. A CSO link connects the valve to the CSO diaphragm which is subjected to manifold vacuum at the throttle body. During coasting, high manifold vacuum exerts a force on the diaphragm in excess of the spring load, and the pull rod lifts the valve off its seat, thereby allowing the pump to discharge directly back into the

float chamber, no fuel flowing to the nozzles. The vacuum valve is so arranged that vacuum can be applied to the diaphragm only when the throttle is closed to eliminate CSO action as the engine is accelerated at no load.

Dynamic Supercharging

Fuel injection makes possible the use of any type of supercharging, the most simple one being a dynamic supercharging method by means of ram pipes. This phenomenon is basically similar to the hydraulic ram or "water hammer" occurring when rapidly closing a water faucet at the end of a long pipe. The kinetic energy of the fluid is momentarily spent to produce a pressure rise. A typical pressure record obtained with a pressure pickup located close to the intake valve (Fig. 16), shows the amount of ram that can be realized. The effect upon the engine torque characteristics of this type of supercharging is well known. Fig. 17 shows imep versus rpm curves for a V-8 engine using various lengths of ram pipes. Their effective use is one of the several indirect gains made possible by fuel injection.

The basic components of the General Motors fuel injection systems are shown in a schematic cross-section (Fig. 18). This illustrates the system's basic simplicity and inherent self-compensation for wear and manufacturing variation.

The spill plunger always controls spill rate to accomplish basic fuel metering as it seeks to balance:

1. The force exerted on the control diaphragm by the static pressure drop at the venturi.
2. The force exerted on the spill plunger by the nozzle metering-pressure.

Airflow establishes a diaphragm force which is transmitted through the control arm to the spill plunger. The plunger is depressed to reduce the flow of spill fuel and raise the pump outlet pressure. The plunger seeks a spilling position where the fuel and air forces balance. The spill-plunger fuel pressure is applied to the nozzles.

The "gear" pressure pump is purely an energy source and cannot affect calibration providing it delivers a supply of high-pressure fuel in excess of engine requirement. Normal wear or manufacturing variation of the spill plunger, the sleeve, the gear pump, or the mechanical components will produce changes in operating positions and fuel leakage. However, these changes will not upset the air-fuel force-balance or the metering calibration, hence the system is self-compensating.

A primary objective of fuel injection is improved distribution. With continuous injection the nozzles must be matched (Fig. 19). Present practice is to guarantee that on any engine the richest nozzle will not be more than 5% richer than the leanest. Unfortunately distribution may be acceptable at low pressure but out of limits at a higher pressure. Semi-automatic equipment establishes the flow calibration to eliminate operator error. Each nozzle, at both a high and a low flow, is classified by the time required to pass a fixed volume of fluid. All nozzles on the same unit are then within 5% at either calibration point. Actually the production variation in nozzle flow is about 15% at high and low calibration. If

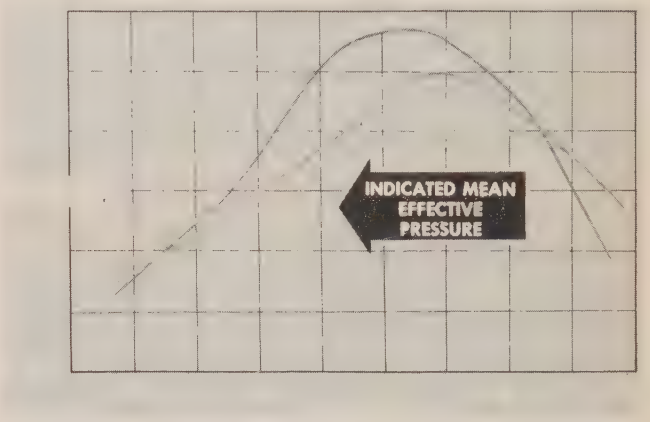


Fig. 17—Comparison of ram pipe lengths

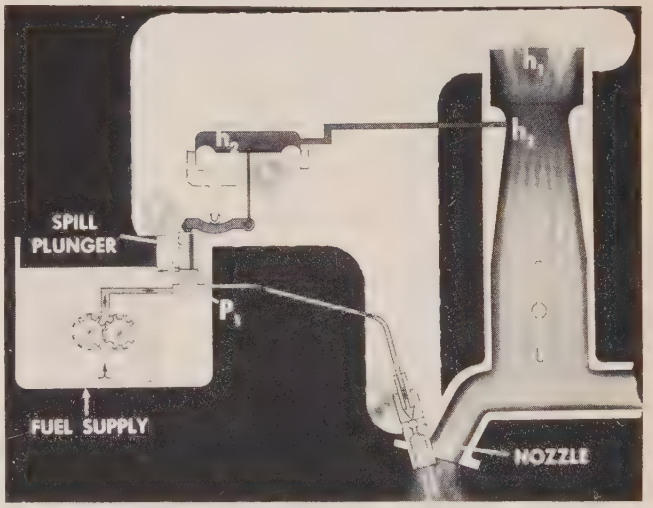


Fig. 18—Fuel injection system of General Motors

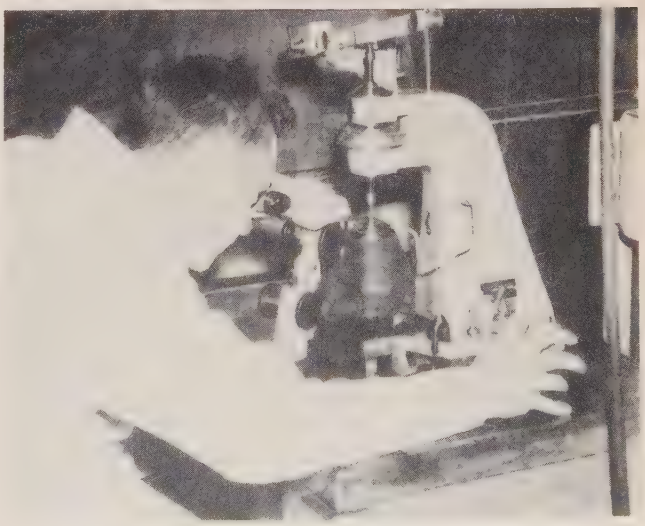


Fig. 19—Nozzle drilling operation

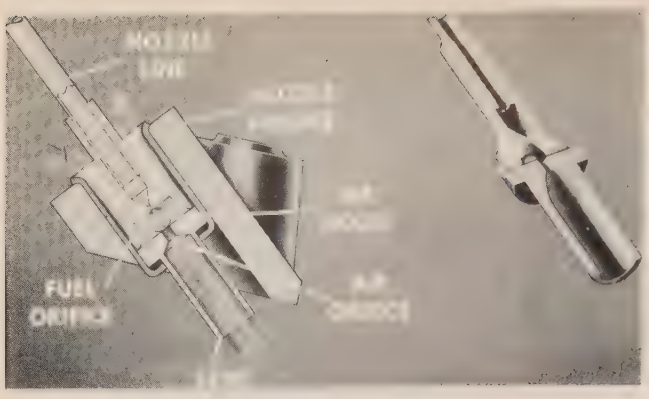


Fig. 20—Injection nozzle

Fig. 21—High-pressure pump

all nozzles are to be used, overlapping bands with varying slopes are required and this results in about 20 different nozzle calibrations. Selection of calibration assists in satisfying a specific engine model's fuel requirement.

The nozzles (Fig. 20) are fed by individual tubes which are either soldered or attached by a tube fitting. If threads are added to the nozzle they are external so that tightening chases chips away from the tube joint.

Fuel flows between the two lands of the spill sleeve to either one or two distributors and then to each nozzle. It is important to make all connections to eliminate the entry of foreign matter and also to protect all lines from absorbing heat by radiation or conduction. The inside diameter of the nozzle line is 0.049 in. All fuel passages are controlled so that the nozzle orifice is the only significant restriction. This permits minor line-length variation and sharp bends without affecting fuel distribution. Obviously nozzle plugging is a possibility; theoretically however, one nozzle may plug completely or partially without interfering in the slightest with metering of the remaining nozzles. This problem, (like getting three square meals a day), is something that can be handled with experience and proper attention to details as proven by actual practice.

Some logical steps toward the solution of this problem are:

1. Cleanliness of parts and of the assembly and test areas.
2. Provision of reliable filters in the fuel system.
3. Protection in storage or shipment from entry of foreign matter.
4. Prevention of part deterioration in production or service which may result from corrosion, wear, heat-treat scale, loose burrs, or inadequate plating.

High-Pressure Pump

For reasons of economy and reliability, it is desirable to use a pump of basic simplicity. The gear pump offers many advantages in this system (Fig. 21).

1. There are no valves in the pump.
2. No inlet piping is required.
3. Wear on gear-tooth form does not affect efficiency.
4. A pressure relief valve is not required—the spill plunger performs this function.
5. Only one shaft seal is required, and this is subjected to low pressure. Other shaft leakage can easily be dumped to the float bowl.
6. The pump is immersed in fuel which keeps it at bowl fuel temperature.

The pump is flange-mounted to the side of the bowl with a $\frac{1}{64}$-in. Buna N rubber-on-nylon gasket. Delivery to the fuel meter is routed through this flange. A second delivery path is opened to prevent fuel delivery to the nozzles when the car is coasting. The pump runs at one-half the engine speed. At any engine operating speed, the fuel delivery averages twice the wide-open throttle requirement. This allows compensation for normal wear and is a decided benefit to instantaneous acceleration. Power is taken from the ignition distributor driveshaft through a shielded flexible shaft.

The poor lubrication of gasoline and the 200-psi maximum pressure necessitate special manufacturing

Fig. 22—Filter, sleeve, and plunger

and production considerations to prevent wear and leakage.

1. Gears, covers, and housing require high precision in squareness, roundness, flatness, and concentricity. Surface finishes are of high quality and critical dimensions are held to less than 0.0005-in. total tolerance. The simplicity in shape, where these requirements apply, is of great benefit.

2. Gears and shafts are hardened steel; shaft bearings are a carbon-graphite compound; covers and housing are of cast iron with a rust-preventive coating.

3. Rather than use a highly precise set of dowels, the parts are spaced at assembly by a special oil film applied to the gears. This assures running clearances equal to the oil film between the gears, covers, and gear housing.

Filters

A supply pump is required to transfer fuel from the gasoline tank to the float valve. This fuel is supplied at about 5 psi and passes through a 10-micron (0.0004-in. particle size) filter. This filter is intended primarily to protect the gear pump and to accumulate foreign matter from the fuel.

A second filter (Fig. 22) is placed between the gear pump and the spill plunger to trap any rare particles which might plug a nozzle. Present design calls for a $^3/_{16} \times 1$-in. diameter capsule of compressed Monel wire, which has the following ideal qualities:

1. It passes no particle above 0.004 in. (100 micron).

2. It passes all smaller particles so as to prevent unnecessary pressure loss with dirt accumulation.

3. It has an infinite number of fuel passages to minimize pressure loss.

4. It will discourage the flow of thread-like materials.

5. It is made of an inert material and is strucurally strong to eliminate deterioration or disintegration of any type.

6. It is sealed in assembly with a diametral and end-compressive load.

Spill Plunger and Sleeve

These parts perform the dual purpose of controlling the spill-fuel volume and transmitting a force to the control arm proportional to the metering pressure. The parts are made of stainless steel with a Rockwell C hardness in excess of 45. The two parts are lapped to about 0.0006-in. clearance to produce unbinding motion and minimum leakage. The lapped clearance is checked by the time required to leak a fixed air volume under controlled pressure. A sharp-edged spiral groove is cut in the plunger to facilitate the removal of foreign particles trapped between the plunger and the sleeve.

The spill ports in the 0.281-in. ID sleeve consist of 10 radial 0.060-in. diameter holes. With a gear pump of 200% of WOT engine consumption, the spill

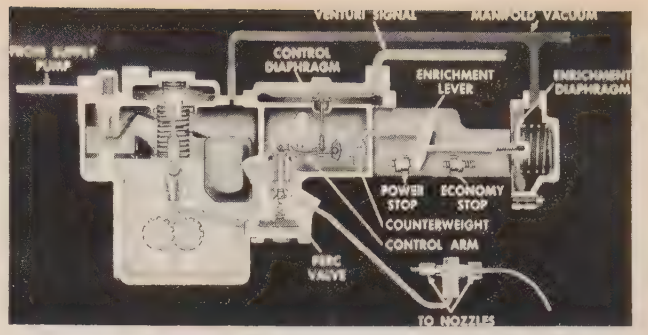


Fig. 23—Fuel meter operating at wide-open throttle

flow and area at WOT are nominally equal to the nozzle flow and area. This implies that the plunger would uncover enough of 10 (0.060-in. diameter) holes to equal the area of eight nozzle orifices. This is equivalent to less than a 0.005-in. opening, hence 0.005-in. closing travel will double fuel flow. A low travel for the spill plunger and control diaphragm is an obvious advantage because of the minimized inertia and delay in metering response to accelerations.

The spill plunger is hollow to reduce its mass. Actually, a pressure head of 4-in. fuel on the plunger will support its weight. The lowest operating pressure is about 8-in. fuel at idle.

Control Diaphragm

The control diaphragm (Fig. 23) must have an accurate response to air pressure differentials of 0.01-in. H_2O while having structural strength to withstand safely a 13-lb load at the highest conceivable signal of 55-in. H_2O. Both the spill plunger and the diaphragm work over a range where the highest pressure differential with normal operation is in excess of 3000 times the lowest. For acceptable accuracy no fewer than three precision manometers are required merely to measure this range of vacuum.

The same low travel requirement which applies to the spill plunger is essential for the diaphragm, in order to minimize both friction and changes in the diaphragm's 3-in. diameter effective area. This diaphragm is a nearly flat, 0.008-in. thick, Buna-N-coated-nylon material bonded to a single, ribbed, magnesium plate to reduce mass and to increase section modulus.

Internal Linkage

The design and fabrication of these parts stress low friction, low mass, strength, and static balance. The control arm is an aluminum die casting with a stainless steel roller to minimize fretting on the plunger. It is counterbalanced about its fulcrum with a matched lead ballast so that its mass will not disturb the delicate balance between air and fuel metering forces. For the same reason a second pivot and counterweight supports the control arm, the diaphragm

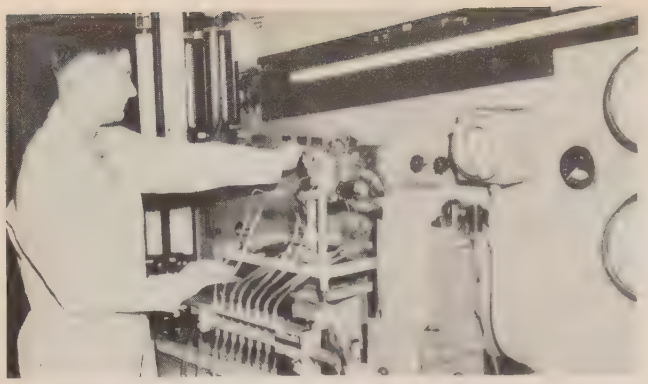


Fig. 24—Fuel meter test setup

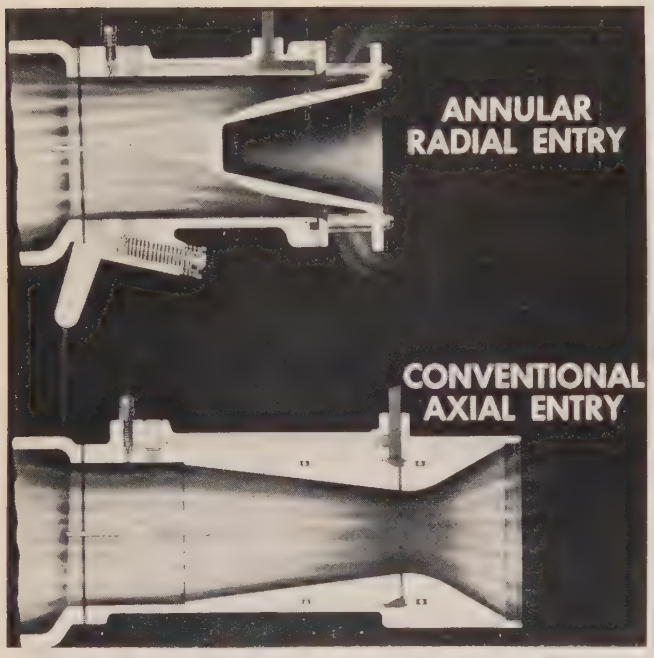


Fig. 25—Air meter design comparison

link, and the unsupported weight of the diaphragm.

A second reason for counterbalancing is to prevent any motion or vibration from producing an unbalanced inertia force which may disturb the calibrated forces.

Power and Economy

The mixture ratio is proportional to the mechanical advantage between the air and fuel metering forces. To change from power to economy, the ratio lever and roller move, thus changing the fulcrum point between the two forces. Contrary to the schematic, the enrichment and ratio levers are on a common shaft and are positioned by the spring-biased enrichment diaphragm.

As shown, the manifold vaccum on the enrichment diaphragm is below 5-in. of Hg. indicating a heavy load condition, and the ratio lever is positioned to give power mixtures. If manifold vacuum rises above 7 in. of Hg., the diaphragm spring is overcome to move the enrichment lever to the opposite stop. The

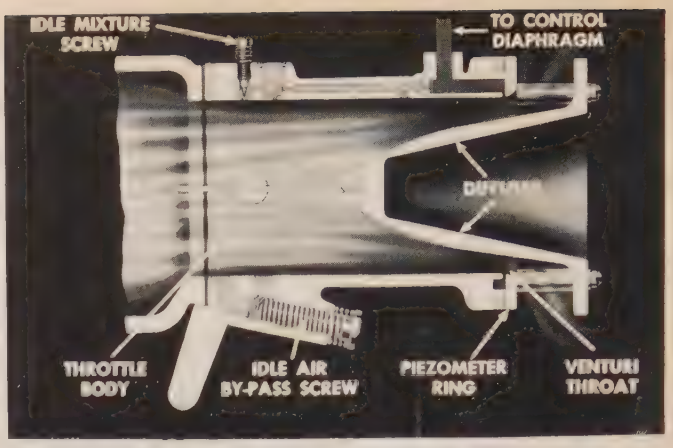


Fig. 26—Air meter at wide-open throttle

ratio lever position now gives the control diaphragm less mechanical advantage, and a leaner economy mixture results from the same venturi signal.

Spill-Fuel Control

This problem is deceiving although the eventual solution in retrospect seems embarrassingly obvious. The spill must be to bowl pressure, and care must be exercised to baffle the spill to remove velocity energy prior to discharge to the bowl; otherwise, there is "fuel injection" in the float bowl, and loss of splashed fuel to the vents becomes a serious problem. The stamped cap is pressed into the casting and seals at the top of the spill sleeve thus forcing fuel through the schematically-oversimplified baffling-passage. Its cross-section must be large enough to avoid any significant back pressure on the 10 spill holes.

Vapor Handling

Below the spill sleeve is a spring-load ball valve which acts to keep pump discharge pressure 4 psi higher than the plunger fuel pressure. At idle, vapor formation is prevented upstream of this "perc (anti-percolation) valve." Downstream, vapor may be released and is free to rise and go out as spill fuel. Liquid gasoline moves radially through eight holes to the annulus around the spill sleeve and then to the nozzles.

Directly below the float valve is an open heat-insulated plastic box which catches hot fuel at hot-idle condition thereby permitting vapor escape prior to its overflow to the main bowl where temperatures are lower. The overall design handles vapor quite effectively. Due to the low control diaphragm idle signal (0.01-in. H_2O), atmospherically venting the float bowl is not adequate to keep vapor pressure from affecting the metering forces. A restricted manifold vacuum vent connection to the bowl is used to prevent metering disturbance. This also simplifies vapor disposition during normal or hot operation.

An effort has been made to reduce the mass of all

components of the system. The heat retention of any part is proportional to its weight; therefore, thin sections are a benefit. This requirement necessitates some compromises with other considerations, but at present all body and cover castings on the air and fuel meters have been made of sand-cast aluminum. Intake manifolds are of fabricated sheet steel or cast aluminum. With more experience and production, changes may be made to die castings of zinc or aluminum.

Fuel Meter Calibration

After checking each subassembly as thoroughly as possible, the nozzles and fuel meter are calibrated as a unit. As pointed out, the ratio lever position controls the mixture ratio. At an artificial control diaphragm signal, and an arbitrary pump speed, the enrichment lever stop is adjusted to produce a required fuel flow for maximum power. The enrichment lever is then moved to the economy stop where a second screw is adjusted to give maximum economy. By adjusting the two stops as required, compensation is made for:

1. The matched set of nozzles used (a set of nozzles with a different flow calibration would require readjustment).
2. Variations in effective area of spill plunger and diaphragm.
3. The production variation in unintended fuel passage restrictions and the relative positions of mechanical forces.

Fig. 24 shows the production test stand. After proper adjustment of pump speed and the artificial diaphragm signal, both a high and low fuel flow are checked for total flow and nozzle distribution. Delivery from each nozzle is collected in a burette to verify that the allowable 5% tolerance is met and to prove that flushing out the complete fuel system has not washed dirt into any orifice. No nozzle or meter should thereafter be disassembled unless a subsequent distribution check is made. Routine testing is also performed to test or adjust:

1. All valve seals and opening calibration.
2. Spring calibration or adjustment.
3. Fuel or diaphragm leakage.
4. Fuel metering—hysteresis and repetition.
5. Pump drive torque, and so on.

Venturi Principles

The venturi differs from a carburetor in that the maximum suction at the throat relative to the inlet pressure is lower (30-in. H_2O at 24-lb per min airflow). Air expansion with this relatively low pressure drop is negligible, and the incompressible fluid flow equation can be used for both air and fuel flows to make the metering system theoretically sound. The resulting larger venturi also helps volumetric efficiency.

Compactness, low frictional loss, and high venturi metering signal were stressed in the air meter design (Fig. 25). A high signal over loss ratio is a direct function of the venturi L/D ratio (exit taper length/throat diameter). A long exit taper forms an efficient diffuser to reconvert air velocity to static pressure without frictional loss. Originally, a conventional axial flow venturi with 1¾-in. diameter throat was used. It had an L/D of only 2.7 in., while the overall length was 7 in.

The unique radial-entry annular venturi presently used proved better in many respects. D is equal to the annular throat width or 0.300 in., hence the L/D ratio increased to 8.3/1. The length required for just the original venturi was as great as now required by a complete air meter. At 24 lb per min, the loss across the venturi is only 2-in. H_2O, while the signal or velocity pressure is 30-in. H_2O.

The inlet to the venturi throat is so arranged that no component of air velocity can act upon the piezometer slot (Fig. 26). Vertical location and concentricity of the diffuser fortunately are not too critical; therefore, controlling dimensions have from 0.002–0.004 in. total tolerance. An unexpected critical part proved to be the blending of the inlet radius on the piezometer ring. It appears that streamline variation in the vicinity of the slot cannot be tolerated if the signal is to be held constant.

Throttle Valve and Shaft

As with a carburetor, the driver controls the engine through throttle valve opening. A single $^{3}/_{32}$-in. brass throttle valve is used in a 3-in. diameter bore. Because of a 75-lb vacuum load, the shaft is ½-in. diameter stainless steel and runs in bronze bearings. The single bore gives the maximum projected valve opening with minimum circumferential leak area. The 3-in. diameter bore was selected to reduce total air meter frictional loss to about 6-in. H_2O at maximum airflow (assumed at 24 lb per min). For the area of bore, the air meter presents much less restriction and better volumetric efficiency than a conventional carburetor.

Idle Control

About 25% of the idle air passes through the nozzle or leaks around the closed throttle valve. The remainder bypasses the valve through an adjustment so as to give warm-engine idle-speed control.

Because idle mixture needs to be richer than road-load economy and because a high-capacity venturi is partially ineffective at low airflow, the venturi signal is artificially strengthened by the application of above-critical vacuum on idle drillings as a function of throttle-valve position. The same critical vacuum across the throttle valve results in airflow varying directly with valve opening and independently of vacuum variation. The resulting fuel flow is directly

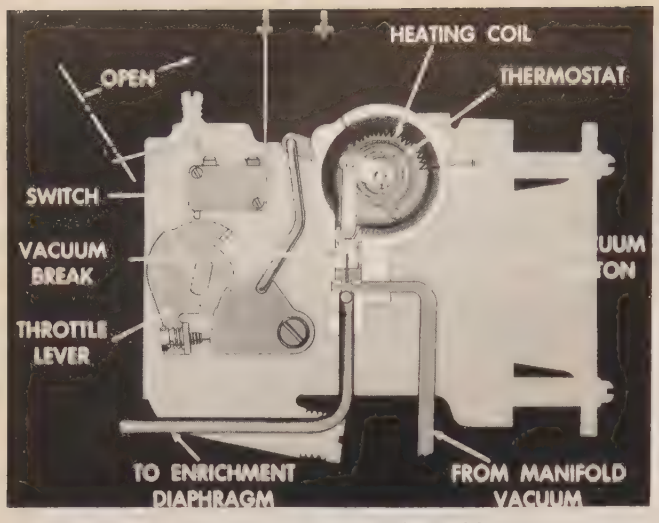


Fig. 27—Air meter calibration for cold operation

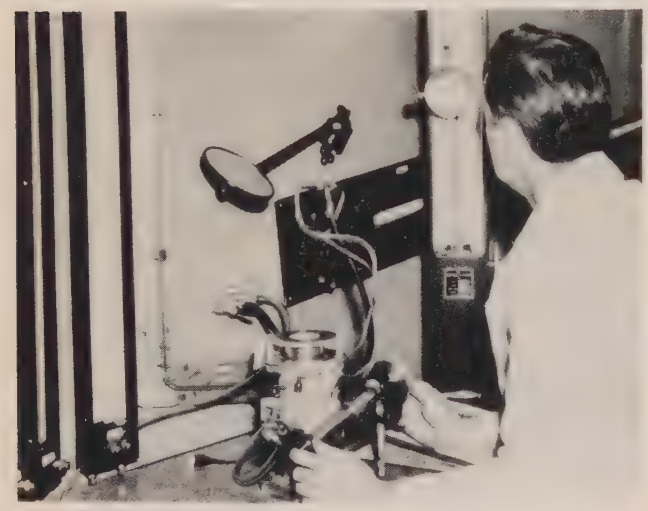


Fig. 28—Air meter test setup

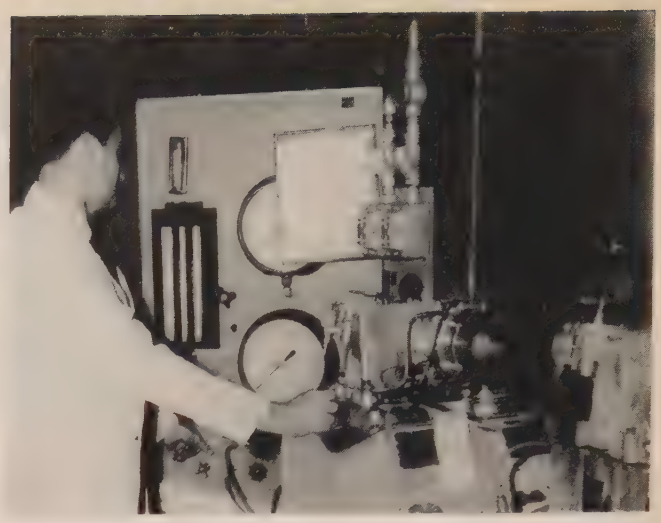


Fig. 29—Final test setup

proportional to airflow until the pressure drop is below critical when the venturi is effective.

The lowest idle-speed mixture may be tailored to a specific engine's requirement with an idle mixture adjustment screw. This adjustment has a diminishing effect as the valve is opened. The entire idle mixture tailoring becomes ineffective at about 3 to 4 lb of air per minute.

Fast-Idle Control

As in a carburetor, (Fig. 27) a thermostat opposed by a vacuum piston senses ambient engine temperature so as to raise a fast-idle cam when the engine is cold. The high step of the cam gives optimum throttle opening for cranking. After the engine first fires, manifold vacuum acts on the piston and overcomes the thermostat. The piston moves to the vacuum break-point thus lowering the cam to the second step. Here, piston vacuum is bled by a port in the housing, and the throttle now may close to the optimum throttle opening for warmup.

To stimulate warmup, an electric heating coil adjacent to the thermostat is connected to the generator armature terminal and is energized whenever the engine is running. The thermostat progressively relaxes to drop the cam to reduce the idle speed.

Cold Enrichment

The enrichment required for engine warmup has been defined. The manifold vacuum applied to the power enrichment diaphragm is taken through the cold enrichment housing. The vacuum, normally applied to the thermostat piston, also acts to seat the ball valve below, which blocks diaphragm passage. This produces a richer mixture for cold light-load operation. As ambient plus thermostat-heater temperatures rise, the piston is permitted to move, and the protruding pin unseats the ball. Finally, as cold enrichment becomes unnecessary, full manifold vacuum acts on the enrichment diaphragm to regulate the ratio lever position, and the piston seals its own vacuum supply passage.

Hot Starting

If the throttle is always partially opened on a hot engine, the solenoid control of cranking mixture produces quick starts. A hot engine cranked at closed throttle may become so overrich that subsequent starting with the throttle open will be delayed. To be positive, a normally-closed throttle switch on the air meter is opened whenever the throttle is more than three-fourths open. This eliminates the solenoid as a fuel enrichment force and leans the overrich engine so it may be started with the same maneuver used on a carbureted engine.

Air Meter Test

Testing is done basically by passing various measured airflows through the air meter and testing the

various output signals. (See Fig. 28.) The characteristics of the venturi are very consistent so that signal test at one airflow is indicative of the overall performance above 5-lb air per min. In the idle range, a more thorough test is needed to measure the air meter signal since it is created by manifold vacuum acting on calibrated idle orifices. As previously stated, readings taken are proportional to the square of ultimate fuel flow so the test is proportionally more accurate than if fuel flows are measured. The mean limits of the air meter are used as adjusted diaphragm signals in the fuel meter calibration. Other adjustments and tests are made to verify the calibration of idle and cold enrichment devices.

Final Test

After having verified the output signals of the air meter, and fuel-flow output of the fuel meter with adjusted signal inputs, the complete manifold, fuel and air meters, nozzles, and so forth, are built up as they will be shipped to the customer. At this point, a final test as shown in Fig. 29 is run with adjusted airflows through the air system to permit measuring total fuel flows. Nozzle targeting is checked, and all functions are spot-checked to verify the final assembly work and initial calibration. No readjustment should be required at final test.

It is desirable that the manifold become a component part of the fuel injection system chiefly to permit the final test of the fuel meter and nozzles after all assembly work is complete. This may lead to an integration of all the components and to eventual design simplification.

The need for fuel injection has been established. The customer is primarily interested in the advantages of today's fuel injection system but to the engineer, the door is open to a new potential of advancements not possible with the conventional carburetor. In the present and future fuel injection designs, the keynote should be simplicity. This will demand a greater measure of ingenuity to achieve the function but it cannot help but attain the two major goals of every engineer, economy and reliability.

Chevrolet became actively interested in the GM fuel injection system primarily for the Corvette sports car, in order to realize the full potential of the engine for high performance. We had been aware of the theoretical advantages of fuel injection, but were also aware of the relatively high cost. The GM system promised maximum advantage with a minimum of disadvantage, such as cost. As we progressed in our application development of this system to the Chevrolet engine, it became obvious that we should extend its use to our regular passenger cars as well.

Our immediate task was to determine how much of the advantages are attainable, how much of the drawbacks we have to contend with, separate inherent advantages from incidentals, and to evaluate the possibility of attaining comparable results with improvements in conventional carburetion. The areas of evaluation were to cover dynamometer and vehicle operation and embrace all aspects on which carburetion is judged.

The first step was to equip a conventional 4-barrel manifold with an air meter and nozzles, as shown in Fig. 30. Care was taken to avoid a pressure drop

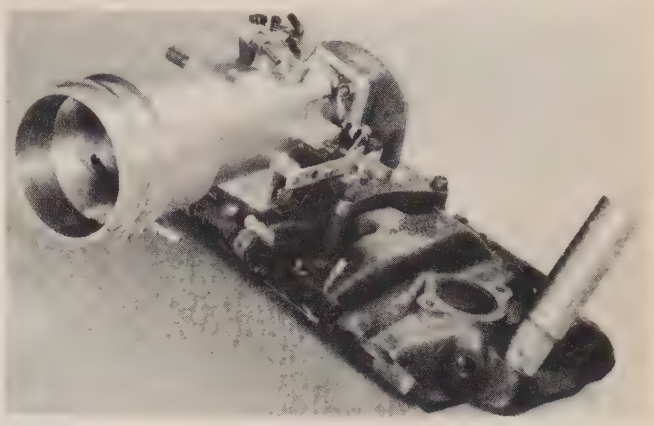


Fig. 30—Conventional 4-barrel manifold with air meter and nozzles

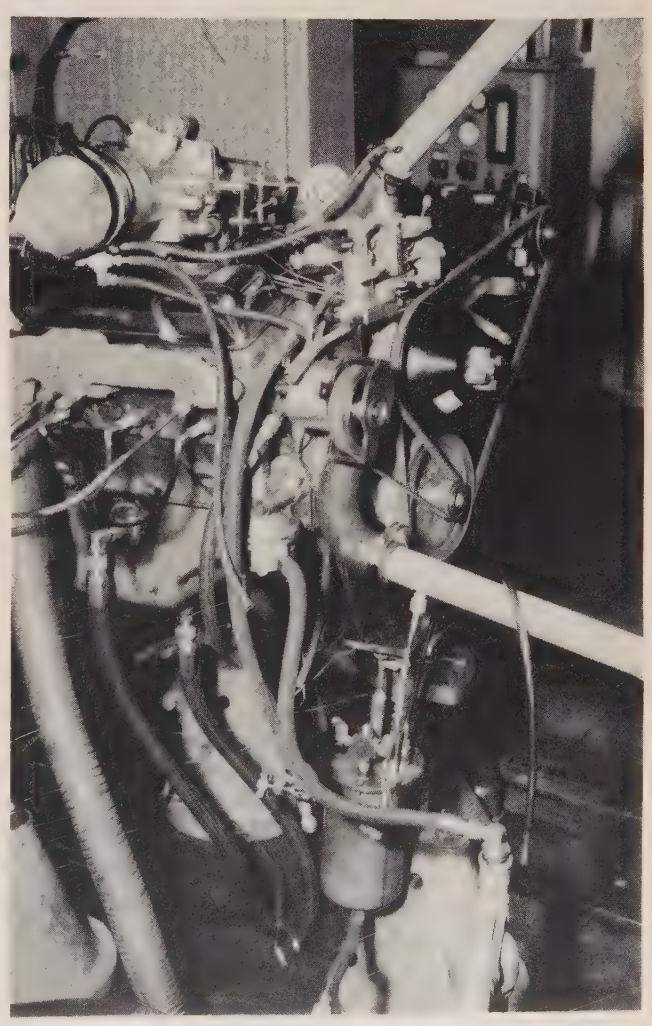


Fig. 31—Complete installation of fuel injection system

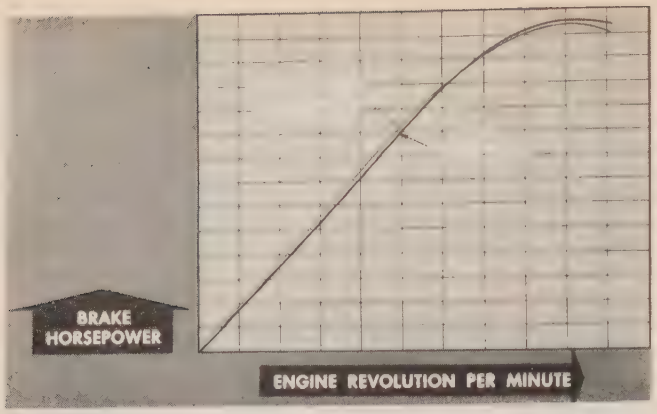


Fig. 32—Power comparison of fuel injection versus 4-barrel carburetor using 4-barrel carburetor manifold

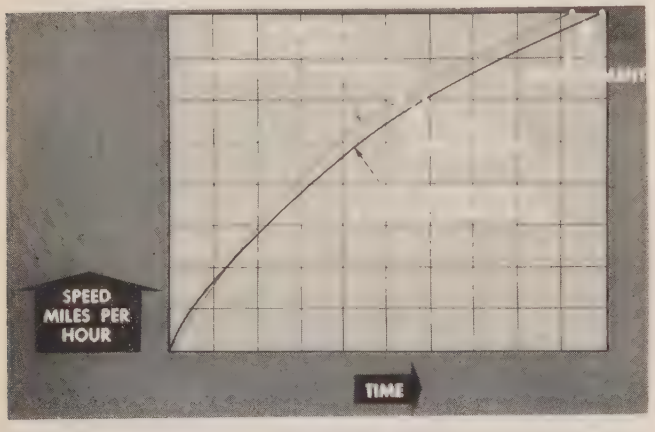


Fig. 33—Acceleration of car equipped with engines of identical power

in the transition from the air meter to manifold, and the nozzle location was carefully selected.

Fig. 31 shows the complete installation with a cobbled up and, at that time, a very experimental injection system.

A power comparison between the 4-barrel carburetor and fuel injection on the 4-barrel manifold was run on the same engine, and the results of comparative tests are shown in Fig. 32. For all intents and purposes they can be termed identical.

The results were not surprising since, within the manufacturing precision at that time, flow variations of the order of 5% were accepted on the nozzles, and the mixture distribution of the well-designed carburetor manifold was well within that range. Also, it should be noted that in this test the carburetor runs were made with heat blocked off, which, in this comparison, nullifies the advantage of the cold manifold inherent to fuel injection. True, the compression ratio increase with fuel injection was not explored in that test, but the spark requirement curves suggested very little possible gain on that score.

The conclusion of the test was that in a steady-state condition, that is, on a dynamometer with air and fuel flow established, fuel injection, as such, did

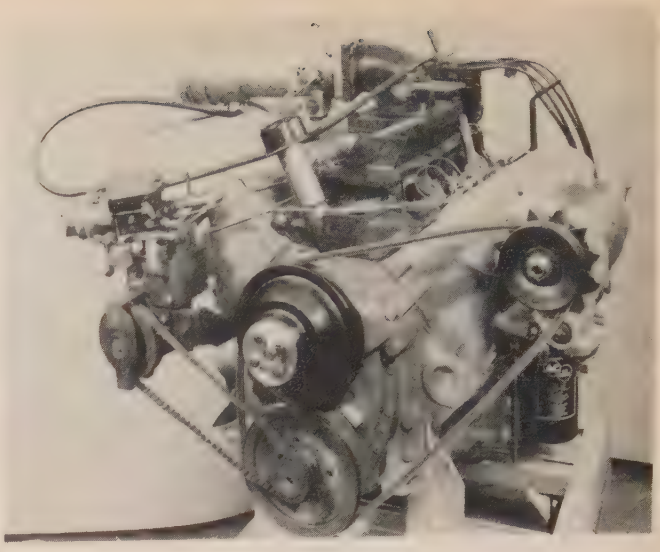


Fig. 34—Two 4 barrel manifolds equipped for fuel injection

not show appreciable superiority over the conventional carburetor.

Having injected and carbureted engines of identical power enabled us to isolate the behavior under transient conditions. The acceleration of a 1956 car was tested alternately with fuel injection and carburetor. The results are shown in Fig. 33. It can be seen that the dynamic condition tells an entirely different story.

The spectacular increase in vehicle performance with otherwise identical dynamometer power can be attributed to the ability of fuel injection to maintain the best power mixture throughout the transition. The mixture composition which the carburetor provides during the transition, relying largely on the acceleration pump, at best could be termed erratic, and the results of the acceleration test brought out the inherent advantage of fuel injection which a carburetor operating with a wet manifold cannot attain. The difference in performance indicated that fuel injection brought out the work potential of an engine which otherwise existed on the test graphs only. We valued this quality since it meant more work per engine pound and dollar.

Our base line high performance engine was equipped with two 4-barrel carburetors. It is a well known fact that engines of that character do anything but justice to their name in the low and medium rpm range and come into their own toward the upper end of the operating speed. By necessity a high-performance engine has a low-velocity manifold, and the problem of mixture formation during the transition is thus aggravated further.

In a manner similar to that used on the 4-barrel manifold, a two 4-barrel manifold was converted for fuel injection as illustrated in Fig. 34. It can also be seen that the fuel injection unit, without reaching a pinnacle of compactness, progressed in that direction, and the maze of injection lines was replaced

by two main lines and eight short pipes, as shown in Fig. 35.

The dynamometer tests yielded roughly the same results—that is, similar output was obtained from both carbureted and injected engines.

The injection equipment was transferred to an automobile and the behavior on acceleration was startlingly superior. Not only was acceleration superior throughout the speed range, but the responsiveness even at lower engine speeds was instantaneous and solid.

The most important lesson resulting from these tests was the establishment of the fact that fuel injection offered freedom from compromise in one aspect of induction system design. The eternal compromise between "high end" and "low end" in terms which affect mixture formation does not exist with fuel injection. This is a very valuable quality, not attainable with single or multiple carburetors, meaning, again, more work per engine pound and dollar.

With the inherent advantage of fuel injection as a work producer established, the incidental advantages, such as the possibility of dynamic supercharging by means of appropriate manifold design, could be explored. Length, cross-section, and changes in cross-section of the intake pipes have a marked effect on volumetric efficiency of the cylinder.

Furthermore, the end result is very closely related to the effective flow areas of the intake valves and their timing. Therefore, basic information had to be obtained on the engine itself using telescoping pipes of different cross-sections. Fig. 36 shows the installation with telescoping pipes on cylinders two and seven. The photograph was taken at the end of the run so the pipes look somewhat stubby.

Some of the characteristic curves obtained from these tests are represented in Fig. 37. It is evident that while fuel injection did away with compromises peculiar to the carburetor, it introduced the necessity for compromises all its own.

We can state broadly that long pipes of small cross-section produce high and narrow peaks, and short pipes of large cross-section produce low and broad peaks, and that large cross-sections show lower flow losses at high engine speed. The choice of pipe, apart from space consideration, is largely governed by the desired engine characteristics. Broadly, it can be stated that engines with a narrow range of operation will benefit by a high but narrow ram peak and engines with a wide operating range will avoid such pipes since they impose a compromise on the compression ratio and violently restrict the upper end performance.

It is evident from the compression pressure curve of the long and narrow pipe that, if the compression ratio is chosen to match the peak, a loss of thermal efficiency must be accepted at other operating speeds. If the compression ratio is matched to some median value, then the ignition advance curve must execute an about face near the ram peak only to leap forward again as the density of the charge rapidly thins out after the peak is past.

Fig. 38 shows the comparison between two pipes of convenient length with small and large cross-section which came under consideration for the manifold design.

The engine configuration at that time, that is, the

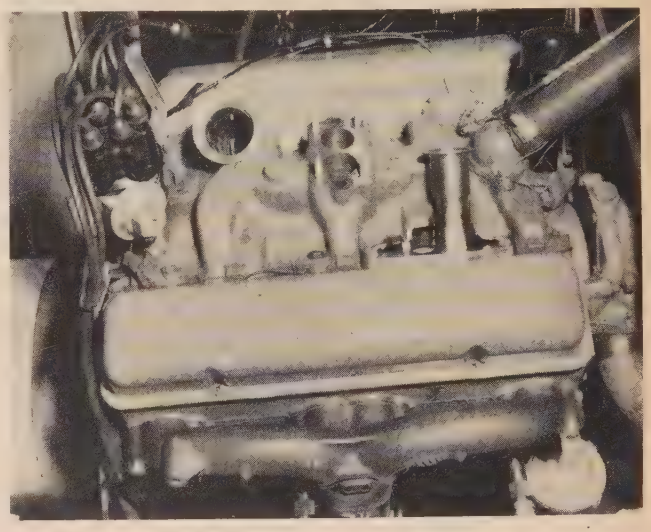


Fig. 36—Telescoping pipe installation on cylinders

Fig. 35—Compact fuel injection unit

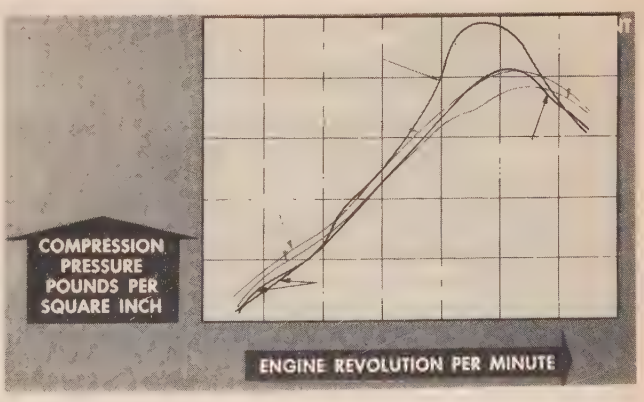


Fig. 37—Motoring compression test on 1955 Corvette engine

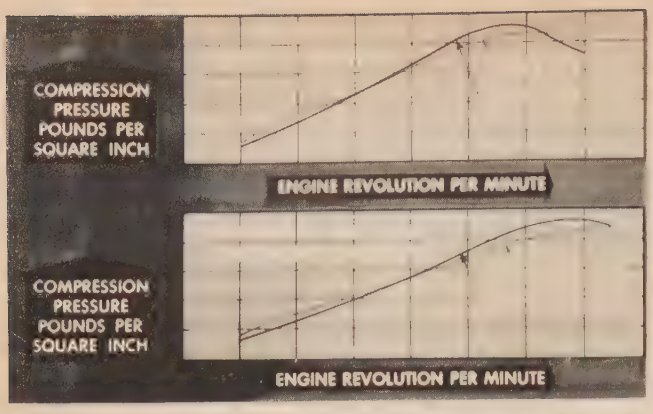


Fig. 38—Gains and losses of small-diameter pipe versus large-diameter pipe where both are of convenient length

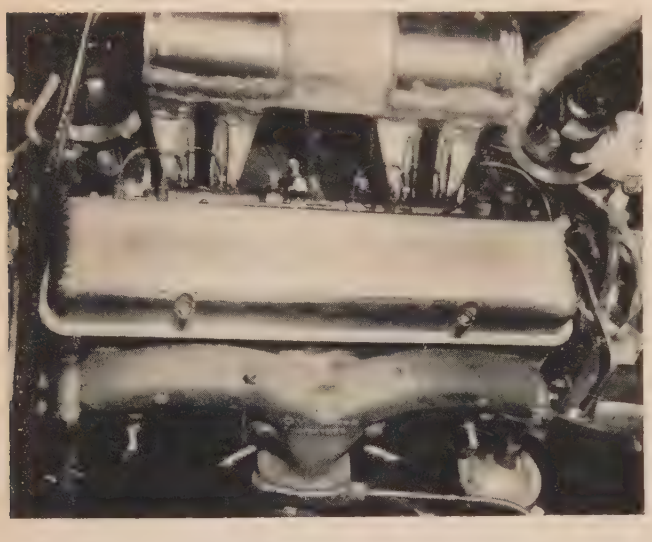


Fig. 39—Welded manifold

cylinder head and camshaft and our concern with the upper end of the curve indicated the pipe of large cross-section as the basis of the most desirable manifold design.

Other primary considerations of the injection manifold design are air distribution and air supply. In respect to air distribution, the necessity for a 2-branch manifold, common to 90-deg V-8 engines, does not exist.

All cylinders can aspirate from a common plenum, provided care is taken that the volume from which overlapping cylinders aspirate is adequately supplied. Furthermore, manifolds with a pronounced ram effect demand either a large venturi cross-section so as not to throttle the flow during ram peaks or a considerable plenum volume to provide an air accumulation for individual pipe pulsations.

Fig. 39 illustrates the welded manifold which incorporated the above thinking, and its tapered "sturdy legs" show evidence of the desire to reduce flow losses at some sacrifice of ram. The results were very satisfactory, and a number of cast manifolds of the type in Fig. 40 were produced to satisfy the need for dynamometer and vehicle development units. The early experimental injection system is very well shown in this illustration including an appendix (acceleration pump) which was later amputated as superfluous.

While experiments with hot and cold starts and the behavior of the vehicles went on with the above described manifolds, further experimentation with the ram pipes was carried out.

An improved head design and a new camshaft imposed a different ram pipe requirement, and Fig. 38 shows that, with new configuration, the straight pipe of low cross-section was markedly superior in the intended operating range. Therefore this pipe became the basis of our production manifold design.

Fig. 41 shows the evolution of the manifold from early experimental designs to the present production parts. As can be seen, the first and last designs are similar, being of 2-piece construction. However, while the reason for the two piece construction of the first manifold was convenience, in the case of the production manifold it was a recognition of desirability. The intermediate design taught us the necessity of insulating the nozzles against heat to achieve desired idle characteristics when the engine was hot. The water crossover passage in our 1-piece manifold, and exposure of the bottom of the manifold to hot oil raised the nozzle temperature to the extent that hot idle was limited to a few minutes. The 2-piece design of our production manifold allows us to insulate the upper part from the lower part with its hot water passages and other hot surfaces.

Furthermore, Fig. 42 shows that the metal-to-metal contact of the nozzles with engine parts is prevented by the insulating nozzle holder and by the air space surrounding the nozzle skirt. The cumulative effect of this change is the ability to maintain idle operation even after the engine coolant may have reached the boiling point.

This functional consideration which led to a clean

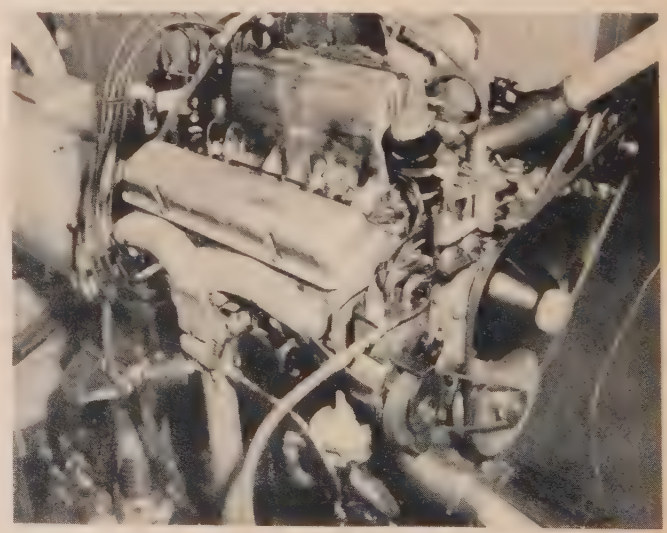


Fig. 40—Cast manifold

separation between the engine and fuel injection system has the advantage of assembly and handling convenience. The engine is "sewn up" with the lower part in the same manner as it would be with our conventional manifold, and the upper part, consisting of the induction and injection system, is bolted on as a package—very much like a conventional carburetor.

The nozzle location, in terms of its distance from the cylinder-head intake port, direction of the spray, and its relation to the passing airflow, has significant bearing on dynamic performance, and on the ease of cold starting and warmup. It is evident that the farther the nozzle is removed from the port the more it will resemble the carburetor in dynamic operation.

It appeared, therefore, that nearness to the intake valve is desirable, and the nearest convenient location for the nozzle outlet is at the juncture between the cylinder head and manifold.

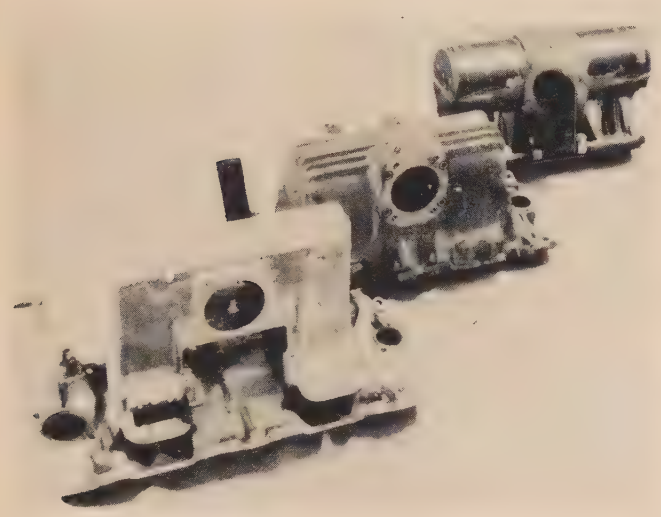


Fig. 41—Evolution of manifold

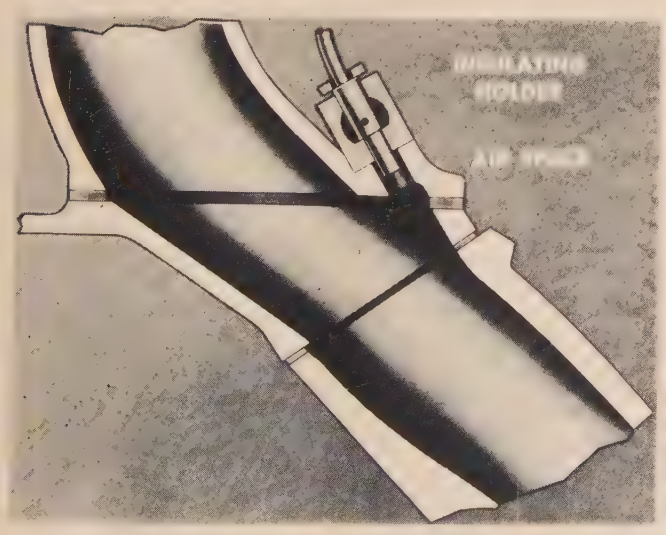


Fig. 42—Chevrolet nozzle installation: nozzle holder and heat insulator

In regard to the aim of the nozzle, the above considerations suggest an aim which avoids the walls and is directed at the intake valve itself. Furthermore it seems desirable to direct the fuel toward the periphery of the valve rather than on the valve stem, particularly in the case of the Chevrolet port layout which imparts moderate rotary motion to the incoming air. The aim and location of the nozzle in respect to the airstream is shown on Fig. 43. The nozzle is placed in a local indentation to combine the necessary intrusion with a minimum protrusion and expose the nozzle skirt to tangential airstreams.

The views expressed in regard to air distribution are fully confirmed by our manifold design. Fig. 44

Fig. 43—Nozzle aiming

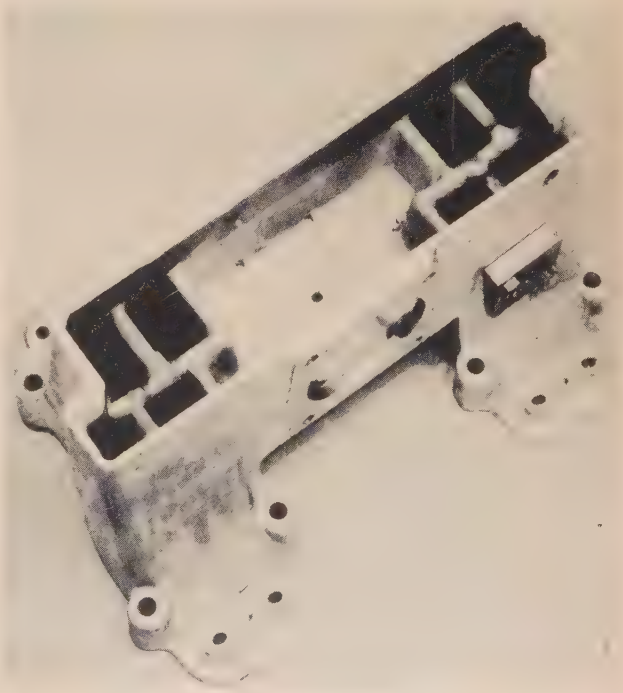


Fig. 44—Cross-section of manifold

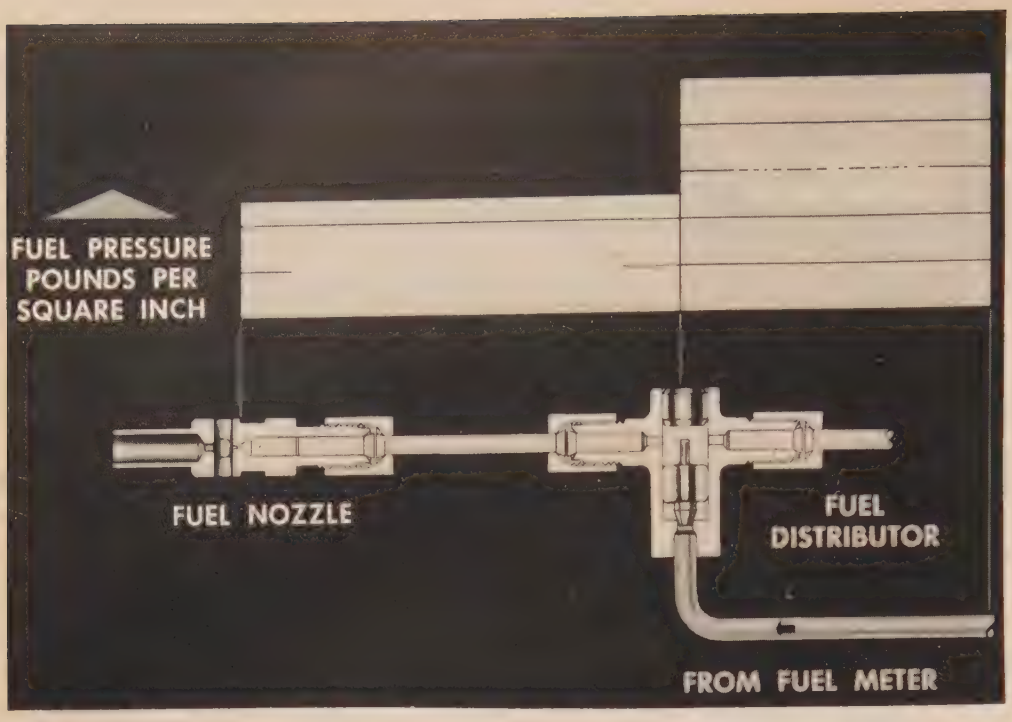


Fig. 45—Reduction of idle fuel rate

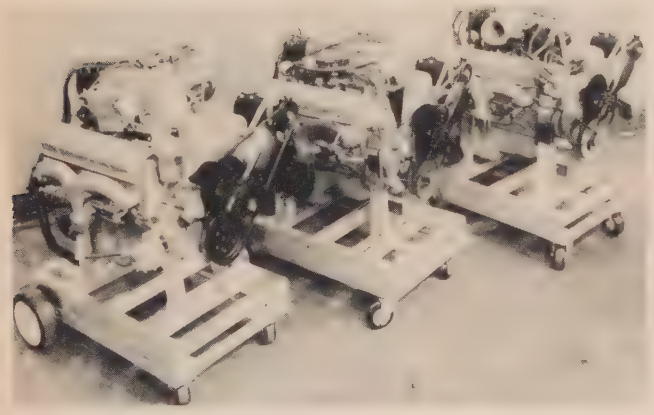


Fig. 46—Evolution of engine with fuel injection

shows the manifold with its top sawed off, and, as can be seen, the individual rampipes aspirate from a common plenum and are grouped without regard to the firing order but according to their geographical location. The showing of an air distribution graph is dispensed with since the identicalness of the values plotted in different colors resulted in an undefinable single blur.

Contemplating the entrails of the manifold, it may occur to the onlooker that the sizes of cross-sectional areas of the air meter and individual pipes are somewhat disproportionate. There are two reasons for this apparent discrepancy.

During the ram peak the pressure waves of the air pulsations in the individual pipe travel at the velocity of sound, whereas the steady flow through the air meter is far from being sonic. Furthermore, the only way to design a Chevrolet is like buying children's clothing—one has to allow for growth.

To increase the range of idle adjustment for all atmospheric conditions and all altitudes, and particularly at the height of Pike's Peak which is so dear to Chevrolet's heart, a fuel rate reduction device was incorporated. Fig. 45 illustrates the method by which a pressure drop is induced at the low fuel pressures. The reduction of rate is a function of balance between the check needle weight in the fuel distributor and its exposed area on the pressure side of the line. At pressure higher than idle fuel pressure the check valve is lifted off its seat and does not interfere with normal metering operations.

Fig. 46 shows some major steps in our development. It can be seen that the metering system hesitantly, but unerringly, moved from the front of the engine to the manifold to form a compact, unified package.

Installed in the vehicle, either Corvette or conventional passenger car, the fuel injection engine provides superior performance, reduced fuel consumption, and instantaneous response to the throttle at all speeds and loads and in all attitudes and states of motion. It enables us to provide a high-performance vehicle with a low speed and low load docility heretofore unattainable. Above all, fuel injection makes usable the work capacity of the engine previously hidden, and with this provides a further step in progress toward efficiency.

This section of the paper has discussed the essen-

tial phases of the application development. A great deal of work went into the development of hot and cold running, and above all into the transient responses for all speeds and loads, which do not lend themselves readily to factual and graphic representations.

Acknowledgment

The fuel injection system described in this paper represents the coordinated efforts of many people. Mr. Dolza would like to acknowledge the outstanding contribution of members of this group, especially Stephen Kalmar, his assistant, George Ransom, William Kolbe, Stanley Mick, Max Homfeld and John Zimmerman of his staff, and Raymond Heafner now of Rochester Products Division.

Mr. Arkus-Duntov would like to acknowledge the contributions to this project by J. Burnell, D. Bedford, Jr. and F. Frincke, all of Chevrolet, and would like to state that all of this work was carried out in close cooperation and with the unreserved help of the Power Development Group of Mr. Dolza. I would also like to acknowledge the contributions made by E. N. Cole and H. F. Barr, who were instrumental in making fuel injection possible for Chevrolet.

APPENDIX

Airflow

The pressure drop dp at the venturi throat is expressed by

$$dp = K_1 \rho V^2 \tag{1}$$

where:

K_1 = Venturi coefficient

ρ = Air density

V = Air velocity

The weight flow rate of air a is given by

$$a = \rho A V \tag{2}$$

where:

A = Throat area

Combining Equations (1) and (2)

$$a = \rho A \sqrt{\frac{dp}{K_1 \rho}} = A \sqrt{\frac{\rho}{K_1}} \sqrt{dp} \tag{3}$$

and if

$$C_1 = A \sqrt{\frac{\rho}{K_1}}$$

$$a = C_1 \sqrt{dp} \tag{4}$$

Fuel Flow

The pressure drop Δp across the nozzle orifice is expressed by

$$\Delta p = K_2 \delta v^2 \tag{5}$$

where:

K_2 = Nozzle orifice coefficient

δ = Fuel density

v = Fuel velocity

The weight flow rate of fuel f is given by

$$f = \delta W v \tag{6}$$

where:

W = Nozzle orifice area

Combining Equations (5) and (6)

$$f = \delta W \sqrt{\frac{\Delta p}{K_2 \delta}} = W \sqrt{\frac{\delta}{K_2}} \sqrt{\Delta p} \tag{7}$$

and if

$$C_2 = W \sqrt{\frac{\delta}{K_2}}$$

$$f = C_2 \sqrt{\Delta p} \tag{8}$$

The air/fuel ratio is then

$$\frac{a}{f} = \frac{C_1}{C_2} \sqrt{\frac{dp}{\Delta p}} \tag{9}$$

that is, the air/fuel ratio (by weight) is proportional to the square root of the ratio of air pressure drop at the venturi throat (venturi pressure) to fuel pressure drop at the nozzle orifice (fuel pressure).

The metering force F_a on the main control diaphragm is

$$F_a = M\,(dp)$$

where:

M = Diaphragm area

The opposing force F_f on the plunger can be expressed as

$$F_f = N\,(\Delta p)$$

where:

N = Plunger area

For any given air/fuel ratio $F_a = F_f$ and hence for a lever ratio of 1,

$$\frac{M}{N} = \frac{\Delta p}{dp} \tag{10}$$

that is, the ratio of control diaphragm diameter to spill-plunger diameter is equal to the ratio of fuel pressure to venturi pressure.

For the nozzle orifice size selected, the value of the fuel pressure Δp at full throttle and top speed may be of the order of 200 psi, while the signal pressure dp under the same conditions may reach about 2 psi.

According to equation (10) the ratio of the area is

$$\frac{M}{N} = \frac{200}{2} = 100$$

Discussion...

of this paper as well as the following Winkler-Sutton paper is found on pp. 765–768.

ELECTROJECTOR—

BENDIX ELECTRONIC

THIS paper discusses the new Electrojector which is Bendix's answer to the fuel injection problem for passenger-car installation. Their approach includes an electronically controlled and electrically actuated system.

Details of this fuel injection system, such as the solenoid valves, are given here.

Fuel supply to the engine during performance, especially in cold weather, is evaluated.

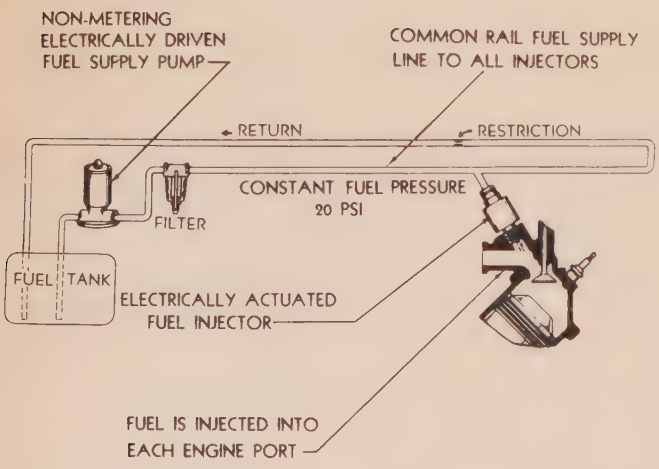


Fig. 1—Fuel supply system

WE of the Bendix Aviation Corp. have for some time been trying to devise a fuel injection system that would be adaptable to passenger-car installations. To promote ready acceptance, the system should be adaptable to existing engines with a minimum amount of modifications; its silhouette should contribute to the lower hood-line trends; and it should be priced so that eventually it could be used in the high volume field.

Approximately four years ago, we initiated work on a system using a solenoid-controlled valve in conjunction with an engine-driven commutator with segments shaped to vary the solenoid-energizing time in relation to engine characteristics. A sample was fabricated and tested, but the preliminary results indicated the impracticability of mechanically modulating the operation of the solenoid valves. However, from this first development, a second system was devised which utilized an electronic control or brain box to modulate the operation of the solenoid injection valves. The test results with this electronically controlled fuel injection system were very encouraging. In conformance to our policy of combining the specialized engineering talent of various divisions, and to develop the system's approach to the problem, the Research, Radio, Friez, and Scintilla Divisions were contacted to assist and counsel us in this program.

The Bendix Electrojector is a new and novel approach to the fuel injection problem for passenger-car installations. As indicated in its name, it is electronically controlled and electrically actuated. It has timed intake port injection and a low pressure, 20-psi common rail fuel system and employs controls that are responsive to intake manifold pressure, engine speed, air pressure, and temperature.

FUEL INJECTION SYSTEM

A. H. Winkler and R. W. Sutton,
Bendix Aviation Corp.

This paper was presented at the SAE Annual Meeting, Detroit, Jan. 15, 1957.

As in any fuel system, whether it uses a carburetor or fuel injection, the metering system must meet certain engine requirements: (1) optimum air/fuel ratio under varying loads and speeds for maximum economy; (2) starting enrichment which tapers off during warmup for good cold weather operation; (3) idling enrichment to compensate for exhaust gas dilution; (4) full throttle or load enrichment for maximum power; (5) acceleration enrichment to avoid momentary lean-out during that transient period; and (6) a fast idle during the warmup period to prevent engine stalling because of the added friction load during this period. In addition to these basic requirements, it is also highly desirable to cut off the fuel flow to the engine during deceleration to reduce the smog problem and to include automatic altitude compensation to expand the optimum operating range of the vehicle.

The following outline and schematics show how the Electrojector system accomplishes these objectives.

Fig. 1 indicates the basic fuel supply system in which fuel is drawn from the fuel supply tank by an electrically driven, nonmetering, fuel supply pump that maintains the line pressure to the fuel injector valve, of which there is one for each cylinder of the engine, at 20 ± ½ psi. Between the fuel pump and the injector is a fuel filter. In the Electrojector system, it is not necessary to filter the fuel finer than 20-micron particle size as the system does not have close-fitting mechanically operating units. Experience to date has indicated that the best results are obtained when the fuel is directed at the head of the intake valve so that a minimum amount of wall wetness results. A fuel return line is incorporated in the system to continually purge any air or fuel vapor from the fuel supply system. To date, we have not experienced any engine malfunctioning due to air or fuel vapor in the lines even though the system has been subjected to high underhood temperatures and empty fuel lines during a shutdown period.

A more detailed picture of the solenoid fuel injector valve is shown in Fig. 2. Fuel enters at the top of the unit and passes through the center core of the valve, discharging through the nozzle at the lower end when the injector valve is off its seat or open. Considerable experimental work and testing was done

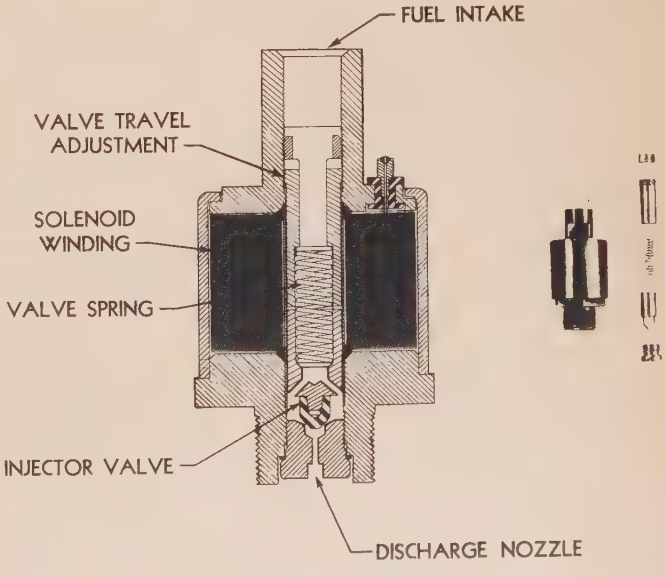


Fig. 2—Fuel injector

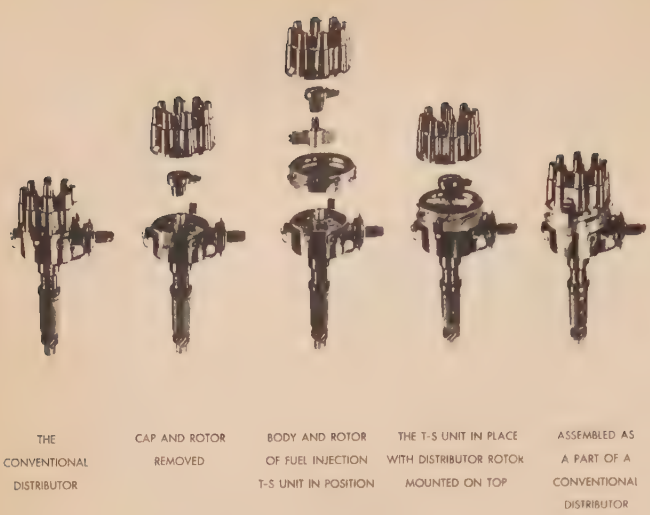


Fig. 3—Triggering selector unit

on this valve and solenoid assembly in order to realize a unit that could operate at high speed, maintain calibration, require low power for operation, and be manufactured at a reasonable price. This, like other units in the system, is new, when compared to carburetor experience, and is continually undergoing changes for improvement in operation and ease of manufacturing.

The Electrojector is a timed fuel injection system, and Fig. 3 indicates the method by which this is accomplished. To a standard engine distributor, a fuel injection triggering selector unit and rotor are added. They are inserted as a sandwich between the base of the distributor and the standard ignition distributor cap. In the triggering selector unit, there is a set of breaker points and a distributing commutator with a section of the commutator for each solenoid injection valve.

The breaker points are actuated by the same cam that operates the regular engine ignition points. Thus, for each two revolutions of the engine, the fuel injector breaker points have made and broken contact the same number of times as there are cylinders in the engine. Each time the fuel injector breaker points make contact, a triggering impulse is transmitted to the electronic modulator box. After which, the modified signal is returned to the selector portion of the assembly and the impulse is distributed to the correct fuel injector valve. Fig. 4 illustrates how the electronic modulator control and triggering selector units are arranged in the system. Through the triggering selector unit, engine speed and fuel injection timing are sensed and the electrical impulse is correctly distributed to the individual fuel injectors. Because the commutator segments neither make nor break the electrical circuit in the system and the electrical energy transmitted by the breaker points is low, very acceptable operating life is expected from this unit. This simple distributor modification is the only change necessary in the engine drive system to adapt it to the Electrojector system.

A separate triggering selector unit can be installed and driven by a flexible shaft from the standard distributor drive in installations where distributor height, configuration, or function prevent the use of the above sandwich-type unit.

The first function of the electronic modulator control or brain box is to transform the spike signals received from the triggering selector unit into an electrical pulse of a given standard width. Simultaneously, signals indicating engine operating conditions are also being received by the electronic modulator control from sensing units located on various parts of the engine. By integrating these external sensing signals into the standard pulse-width circuit of the electronic control system, the standard pulse width is modified to reflect engine conditions. This modified electrical pulse is then transmitted to the selector portion of the triggering selector unit which in turn distributes or directs the pulse to the correct fuel injector.

Originally, vacuum tubes were used in the electronic modulator control, which necessitated a short warmup time for the vacuum tubes before the system could be put into operation and required a relatively high current draw from the standard car battery. To improve the operating characteristics of the unit and to increase service life, the control was transistorized. Fig. 5 shows how the size of the modulator box was reduced as the development program progressed; Fig. 6 shows one of our more recent units. In this unit, the waiting period has been eliminated and the maximum average current draw reduced to 3½ amp.

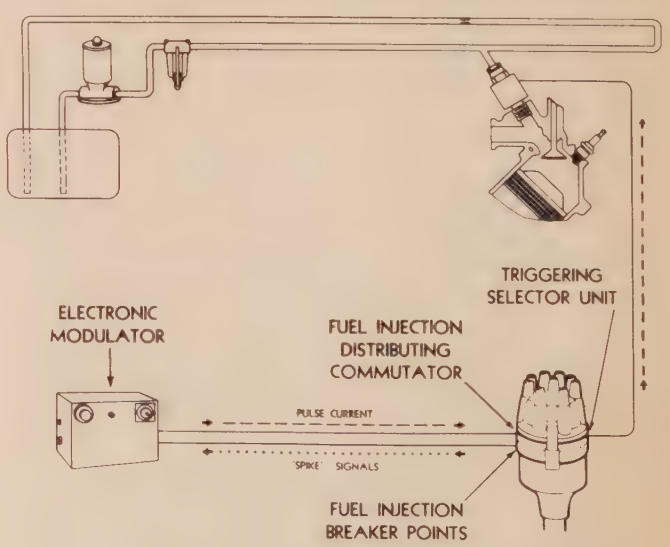


Fig. 4—Injection timing and injector actuating elements

Standard type, but larger, throttle valves are used to control the airflow to the engine. Attached to the throttle body is an intake manifold pressure sensor that transmits a signal to the electronic modulator control box indicating the relative density of the air charge entering the engine.

Fig. 7 schematically indicates the operation of the intake manifold pressure sensor and shows a photograph of a throttle body. The actual resistance characteristic will vary for different makes and models of engines.

Basically, in this and all of the other external sensing units, added resistances are introduced into the fundamental circuit to modulate the pulse width of the electrical impulse that is transmitted to the solenoid injector valve. As the sensing-circuits' resistances increase, the pulse widths increase, and the length of the time that the injector valve is held off the seat likewise increases, which results in additional fuel flow.

Experience has indicated that, when an engine is accelerated, it is necessary to supply a small extra quantity of fuel to the engine in order to obtain smooth engine operation during this transient period. Fig. 8 illustrates one method of accomplishing this temporary enrichment through a manifold-vacuum

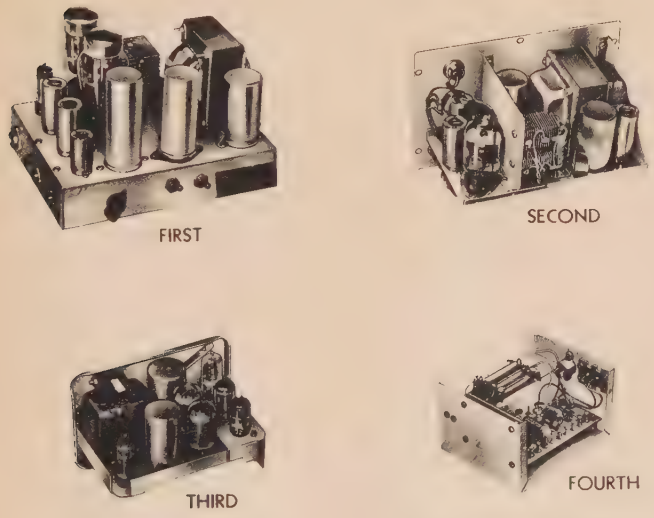


Fig. 5—Development phases of electronic modulator

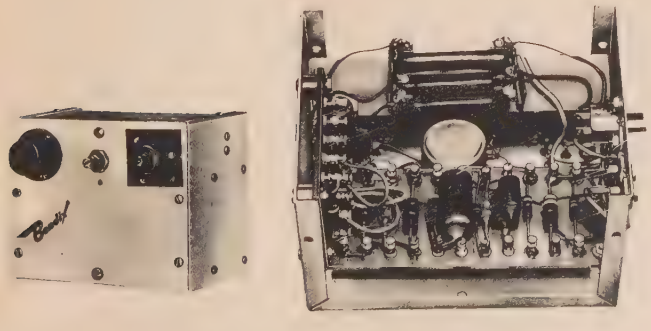


Fig. 6—Electronic modulator

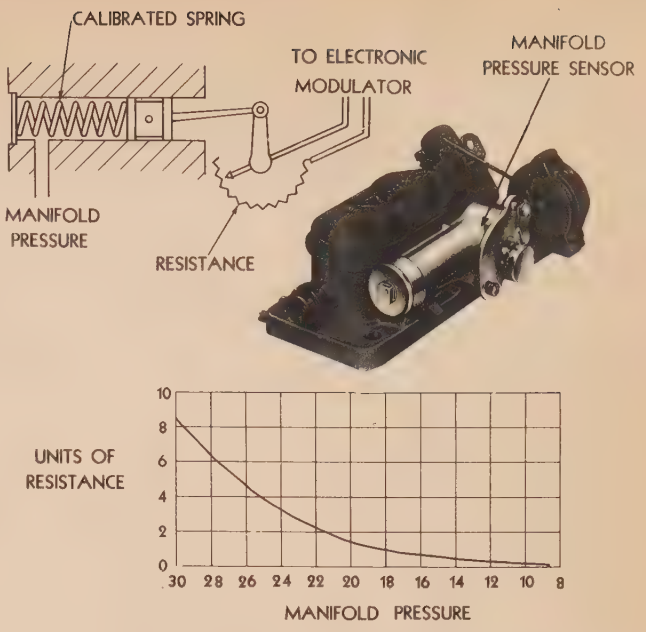


Fig. 7—Intake manifold pressure sensor

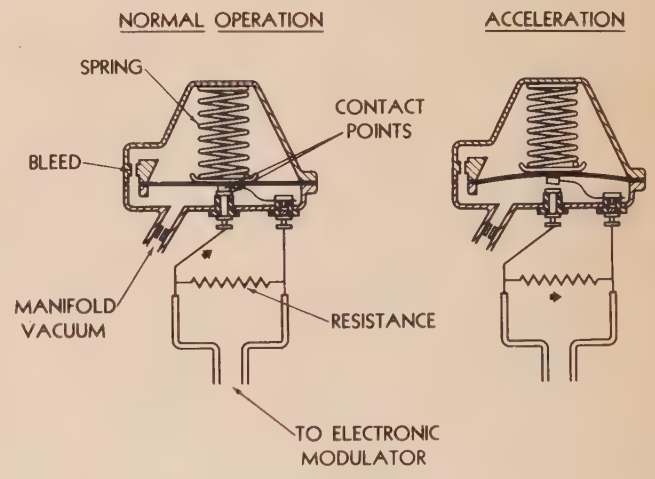


Fig. 8—Acceleration enrichment

change-sensing device. In operation, a rapid change in manifold vacuum will cause the points to separate and introduce an additional resistance in the control system, thus increasing the operating pulse width of the electrical signal for the length of time necessary to equalize the pressure in the chambers on both sides of the diaphragm in the unit. The added pulse width will increase the fuel flow from each injector inasmuch as the injectors will be energized for a longer period of time during this transient period. We are also considering alternate methods such as mechanical operation through the accelerator or throttle linkage.

Smooth idling is important in any car operation and in order to meet the slight variations in fuel requirements between engines of the same make and

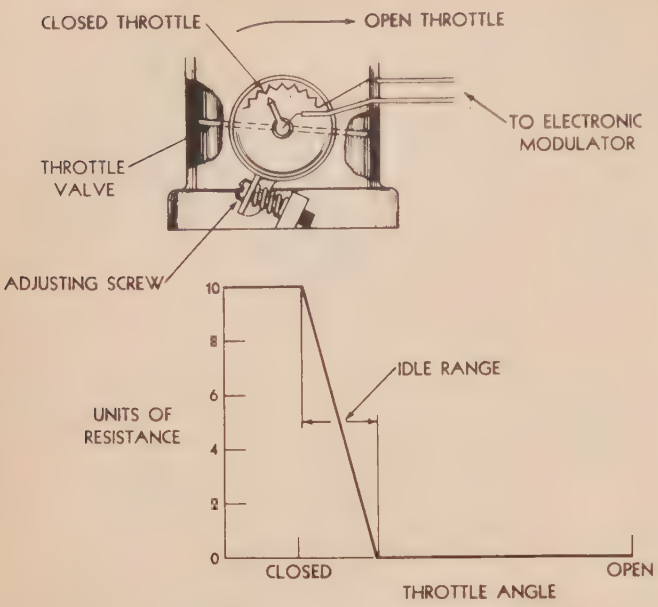


Fig. 9—Idle enrichment

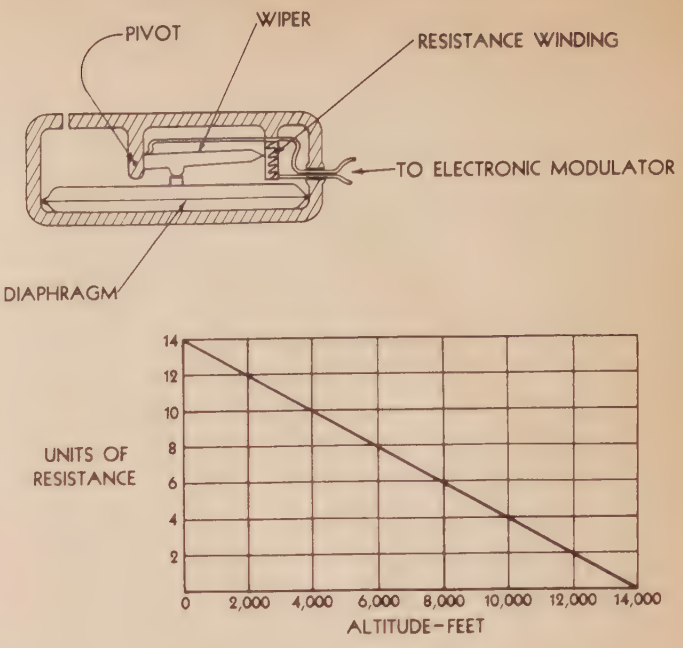


Fig. 11—Altitude compensator

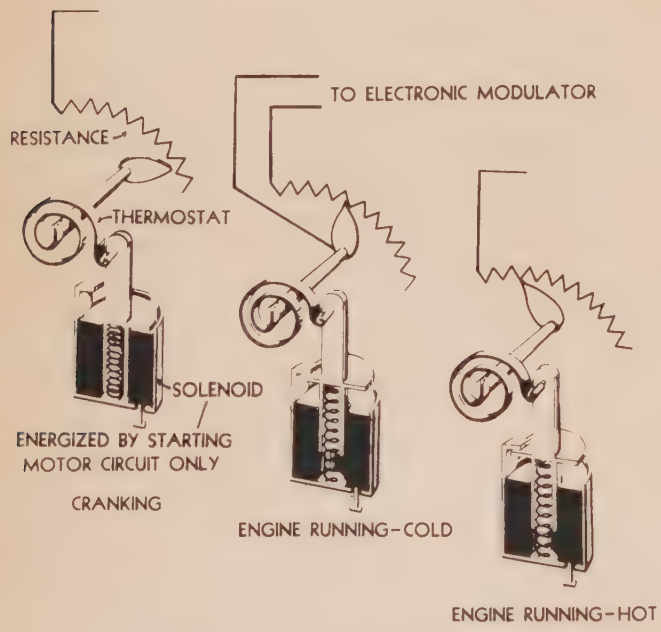


Fig. 10—Starting enrichment

model, a separate idle enrichment control, shown in Fig. 9, is incorporated in the Electrojector system. A rheostat is connected to the throttle shaft to interject a variable resistance in the control circuit when the throttle is in the idle position. The idle adjustment is obtained by adjusting this resistance instead of the conventional idle mixture adjustment needle.

We have now conveyed signals to the electronic modulator depicting the speed of the engine and intake manifold pressure as well as its accelerating and idle requirements. The electronic modulator integrates these various signals and transmits them through the distributing commutator system to the correct solenoid injector valve and causes them to deliver fuel to the engine in the proper quantities. This, however, is not sufficient to satisfy all conditions of operation. During cold weather operation, it is necessary to supply additional fuel for starting and to diminish this enrichment as the engine warms up.

Fig. 10 indicates how this can be accomplished in the Electrojector system. Cold tests have indicated that it is necessary to supply more fuel to the engine during the cranking period per engine cycle than is necessary to operate it after the start and that the quantity of fuel required will vary with the engine temperature. To accomplish this end result, a thermostat positions a variable resistance such as a rheostat in the control system which will increase the resistance as the temperature diminishes thus increasing the pulse width with decreasing temperatures. To obtain the added fuel flow during the cranking period, an electric solenoid operating through the thermostat repositions the rheostat to a higher resistance position during the cranking period. Since the thermostat reflects engine temperature, the amount of enrichment decreases as the engine warms up. This signal, like the others, is transmitted to the electronic modulator box for integration with the other signals.

A conventional fast-idle cam and thermostat mechanism provide the necessary fast-idle speeds during the warmup period.

An additional refinement in temperature control can be obtained by inserting a thermistor that senses the actual air temperature being supplied to the engine

during normal temperature operation in the air intake passage. This signal, in the form of resistance, is transmitted back to the electronic modulator control unit which in turn modifies the pulse width, and thus fuel delivery to the engine, in relationship to the actual air temperature being supplied to the engine.

In recent years, smog control has become more and more of a problem. To reduce this annoyance, it has been found advantageous to cut off the fuel flow to the engine during deceleration when the manifold vacuum is abnormally high. Since, as indicated previously, fuel-flow modulation is partially controlled by manifold vacuum, fuel cutoff is attained when the manifold vacuum exceeds a predetermined value. With the fuel injectors located adjacent to the intake valves, there is very little, if any fuel carryover from the manifold, and thus a clean cutoff is effected.

While altitude control and compensation are commonplace in aircraft units, it is commercially nonexistent in present passenger car installations. By the addition of an automatic altitude compensator sensing unit attached to the electronic modulator box, it is possible, with the Electrojector unit, to obtain automatic altitude compensation. Fig. 11 schematically illustrates a configuration of such an altitude compensator which includes a small aneroid with a variable resistance unit, that transmits its signal to the modulating unit. Hence, we now have modulating and sensing means for controlling the fuel flow to an engine under all engine operation conditions.

To summarize this control system, Fig. 12 schematically indicates the various items and sensing units. Now, to review the cycle of operation: as the engine rotates, the injector breaker points send out spike signals in time with the opening of each intake valve to the electronic modulator unit. The multivibrator circuit in the electronic control box is triggered by the spike signals, and this sets up a series of pulse currents tuned to a standard pulse width. The standard pulse segments are then subject to modification by the various modulating or sensing circuits: the inlet manifold pressure sensor, deceleration cutoff sensor, starting enrichment control, idle mixture enrichment control, altitude compensator, and acceleration enrichment control. This modified impulse is then transferred to an amplifier where the pulse current is amplified to the power needed to energize and to actuate the solenoid injector valves. This amplified electrical pulse is then directed to the selector where it is distributed to the proper fuel

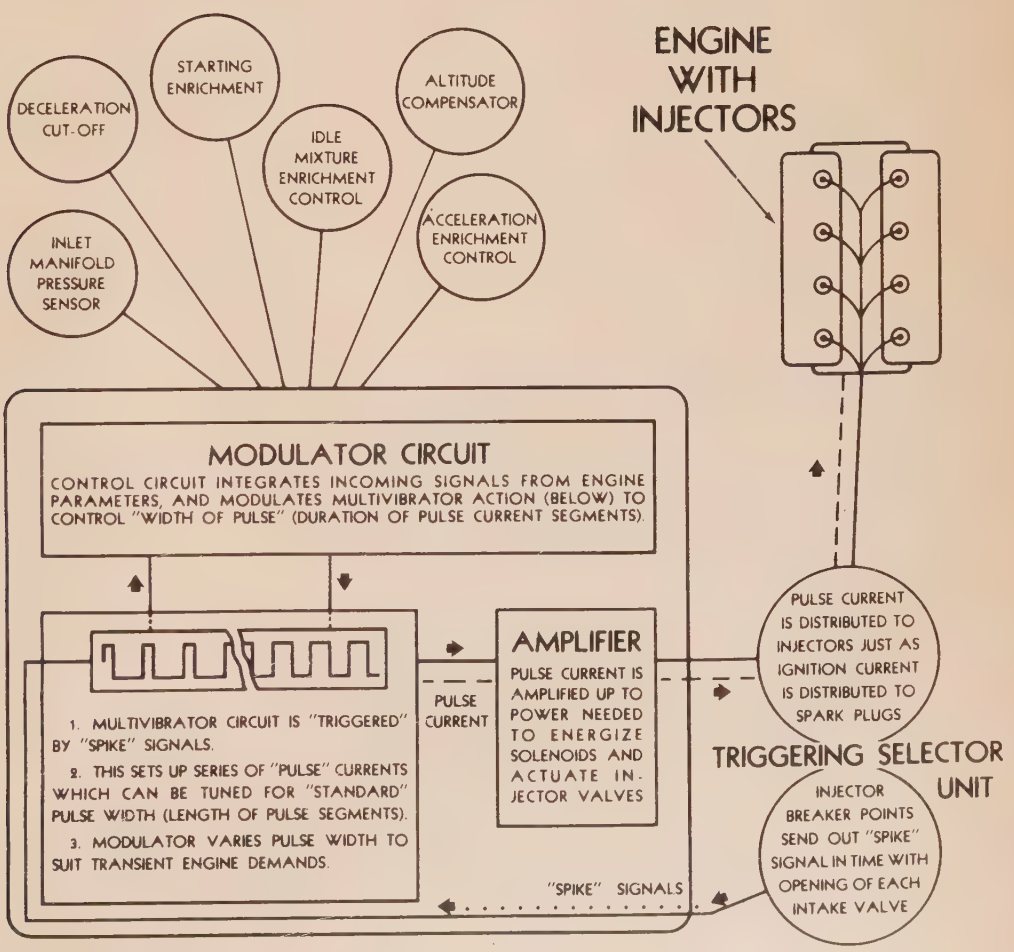


Fig. 12—Engine parameters and demands

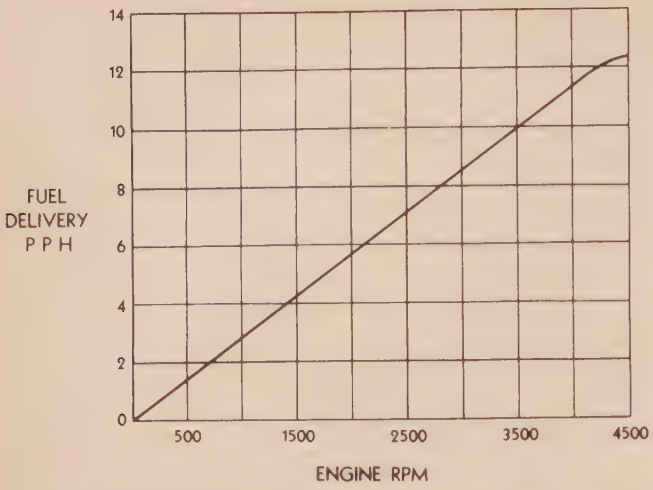


Fig. 13—Fuel delivery per injector at constant pulse width

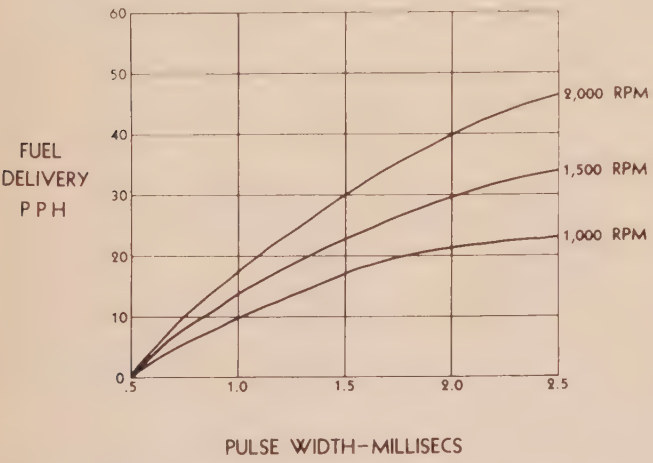


Fig. 14—Fuel delivery increasing with pulse width per injector valve

injectors just as an ignition current is distributed to the various spark plugs. The injectors then react to the power impulse causing the valve to lift from its seat and permit the required fuel to flow to the cylinder.

The Electrojector system maintains a constant pulse width during full-throttle operation, and this results in a fuel delivery that is proportional to engine speed through the major portion of the speed range as indicated in Fig. 13. At engine speeds above approximately 4000 rpm, there is a slight fall-off in the fuel delivery curve. Fortunately, a similar trend is noticed in engine air consumption.

When the pulse width is varied, fuel delivery over the speed range will increase with increasing pulse widths. However, fuel delivery is not directly proportional to the pulse widths, but assumes a characteristic curve as indicated in Fig. 14. The actual shape of the fuel curve with varying pulse widths and at constant engine speed is a summation of the various

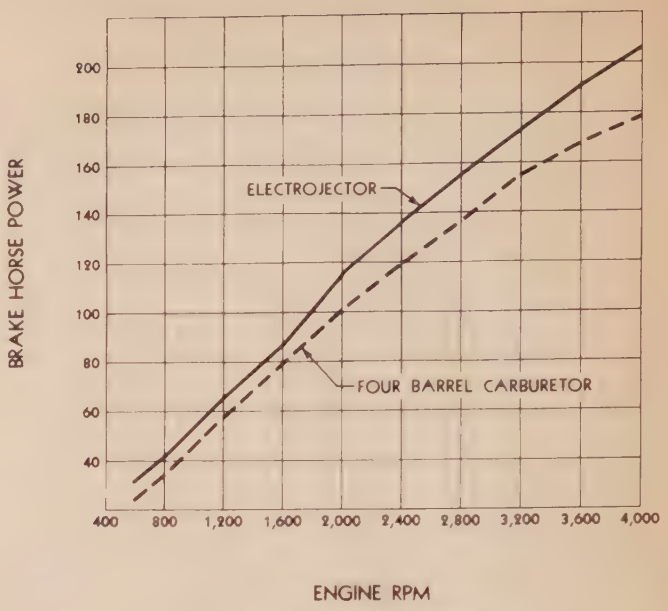


Fig. 15—Typical test results

electrical, flow, and motion characteristics of the units in the system.

With either direct or port fuel injection, increased engine torque and horsepower are obtainable through the use of larger intake manifolds engineered for maximum air delivery and the elimination of intake manifold heat. Fig. 15 is a typical horsepower comparison test between a standard 4-barrel carbureted installation and the same engine equipped with an Electrojector fuel injection system; Fig. 16 indicates the torque results.

In our preliminary experimental road economy test with the Electrojector fuel injection system, we have obtained results indicating better part-throttle road economy as shown in Fig. 17. (We have also had test results on other installations that were not as satisfactory.) Results as shown in Fig. 17 do however indicate a potential improvement in part-throttle economy. For optimum results, the fuel injectors, manifold, and engine must all be compatible with each other. What the ultimate gains and improvements in part-throttle economy will be is partially dependent upon how well the fuel injection concept can be integrated into this overall engine design.

With respect to cold weather operation, we have had some very encouraging cold start and warmup test results. Since the fuel is injected adjacent to the intake valve, the starts have been very good, and the length of time required to supply added fuel during the warmup period was very materially reduced. This characteristic of the fuel injection system should improve the overall tank economy during short-run operations in cold weather. A complete installation is shown in Fig. 18.

When comparing the Electrojector system with

other fuel injection systems, it has these advantages:

1. Elimination of the expense and complication of a high-pressure metering-type pump.
2. Fewer moving parts.
3. No special pump-drive from engine.
4. No critical filtering requirement.
5. No surge or inertia effects as in a pulsating, high-pressure fuel line.
6. No vaporlock.
7. Self-priming.
8. Easier adaptation and assembly line installation.
9. Quieter operation.
10. Low electrical requirements.
11. No ultra-precision machining standards.

When comparing fuel injection against carburetor, the Electrojector system has these advantages:

1. Increased power.
2. Higher torque.
3. Quicker cold starting and warmup.
4. Wider latitude in fuel.
5. More room under hood.
6. Idle cutoff.
7. Ambient air compensation.
8. Altitude compensation.
9. Faster, livelier response to throttle, and better all-around performance.
10. Lower hood line possible.
11. Higher volumetric efficiency—intake manifold can be designed entirely for air.
12. Flow efficiency without compromise for distribution to maintain gas velocity at low speeds.
13. No cold muffler on dual exhaust systems.
14. No need for manifold heat—cooler inlet adds to volumetric efficiency and power.
15. Lower intake temperature allows earlier spark, and higher compression without detonation adding to power.
16. No throttle-valve icing.
17. No cornering or hill-angle effects.

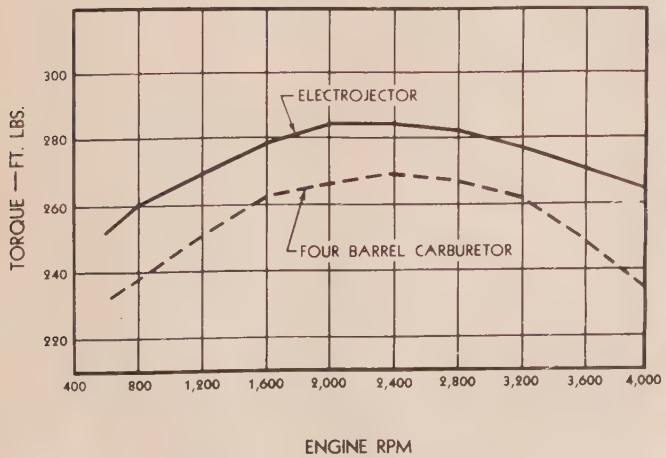


Fig. 16—Typical test results

In conclusion, we are conscious that this is a new concept in fuel metering and poses new and different problems. We and our associates are energetically conducting test programs to add to our knowledge of the system as time and equipment are available.

DISCUSSION

Discusses Pontiac's Fuel Injection System

—J. F. Verkerke
General Motors Corp.

AS you probably know, fuel injection has been introduced by Pontiac for limited 1957 production on our Bonneville Convertible which is being sold to dealers only. This is Pontiac's

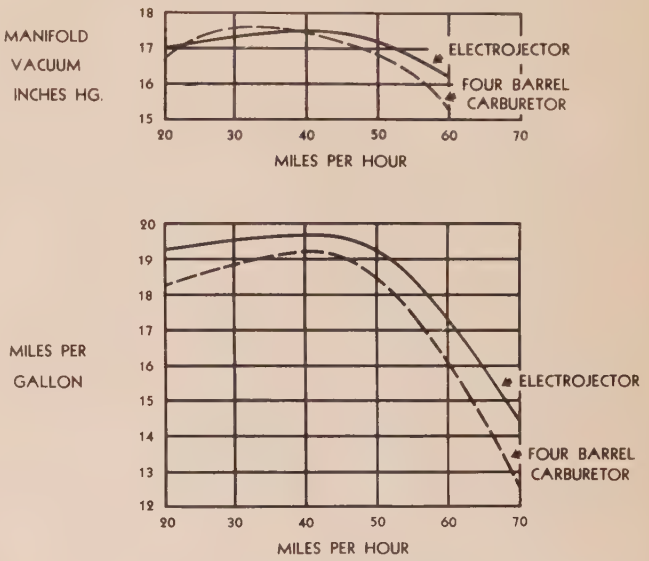


Fig. 17—Road economy tests

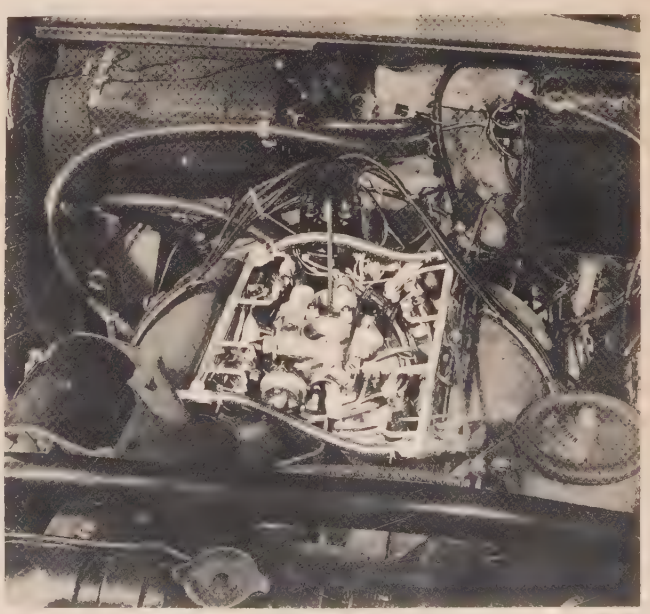


Fig. 18—Electrojector car installation

fuel injection for 1957. Not much more than a name is viewed here so let's look under the hood. Yes, this is the extent of what you'll see of this year's fuel injection for Pontiac. What is actually viewed here is the inlet air duct, the air cleaner, and a shroud which completely encloses the fuel induction assembly. The shroud provides the air inlet box for filtered air to the venturi and nozzles and at the same time provides cooling for the entire system. This feature, in addition to nozzle block and intake manifold insulators, provided the necessary cooling to enable us to still be idling after 30 min at 100 F. A minimum of 30 min at idle at steady state will result in satisfactory city driving at high ambient temperatures. The shroud has also proven itself to be an excellent shield against dust and dirt from collecting on any part of the injection system.

Inside the shroud, Pontiac's basic system is the same as those described in the foregoing papers; however, our development efforts have been directed toward adapting the GM fuel injection system for conventional passenger-car use. Designing and testing intake manifolds of many configurations having various sizes and length ram tubes, led to our released model which provides substantial torque gains above 2000 engine rpm. This design combined with the GM system has resulted in substantial increases in car performance. A gain in acceleration time from 0–60 mph of more than 10% over our conventional 4-barrel carburetor car was achieved.

While our main goal for this year has been to produce a power-plant having all the advantages that fuel injection can offer for superior performance and smoothness under all car driving conditions, fuel economy has not suffered. We have shown level road fuel economy to be about equal to that obtained on our carburetor cars, and during a cross-country trip, the fuel injection car averaged more than 5% better overall economy than a comparison car equipped with a conventional 4-barrel carburetor.

Our performance development has been carried on quite extensively at both the hot and cold extremes of weather conditions. Running in the hot climates has proven our fuel injection system to be resistant to vapor locking tendencies even with high vapor pressure fuels. High altitude running also presented no particular problems. Cold weather development, however, particularly in the range of 0 F has entailed considerably more development.

It can be said that fuel injection offers the engineer a tool with great possibilities for refinement. Our future goal for fuel injection will be to maintain our high standard of performance while simplifying the system with a resultant reduction in cost.

Discusses General Motors and Bendix Fuel Injection Systems

C. G. Nystrom
American Bosch Arma Corp.

THE excellent papers presented by the authors on the subject of gasoline injection is of interest to a great number of people today and of particular interest to the American Bosch Arma Corp. because we, too, have a gasoline injection system of which we are quite proud.

Basic Development

The basic development of the ramjet constant-flow system as presented by Mr. Dolza of GMC, has some very interesting features of note, and both he and his associates should be complimented for their success in this development.

During the development of the American Bosch metered gasoline injection system, we studied various carburetion systems, including conventional carburetors, pressure carburetors, and multinozzle constant-flow arrangements. Among the latter was one basically similar to the system described by Mr. Dolza which was both dynamometer and road tested. In our analysis of all the carburetor and injection systems considered, we concluded that the timed-metered injection system offered the greatest potenial and the most advtantages, not only for the application on existing engines, but in the application to engines of both two and four cycles designed specifically to take full advantage of the gasoline injection.

Messrs. Kehoe and Stoltman, along with their associates should be congratulated on this presentation of the production development of ramjet constant-flow system. It is interesting to note the great detail used in describing the various component parts and the accuracy needed during manufacture, as well as the numerous tests of the component parts of the system, and the final calibration test of the entire system including the intake manifold. The only conclusion I could arrive at was that this type of fuel system has inherent features indicating a high manufacture cost.

Application Development

I will not comment on this phase of the development because we are accessory manufacturers and have to rely on the various engine manufacturers to design the intake manifolds and cylinder heads to be able to take the full advantage of our injection system.

Electrojector

Messrs. Winkler and Sutton, Fuel Systems Engineering Department of the Eclipse Machine Division, Bendix Corp., have made an excellent presentation of this paper on the Electrojector fuel injection system. This system, incidentally, is a timed system as you know, and basically we are in agreement with this idea. The system, which has been described in the authors' paper, has a number of very interesting features. The electronic modulator or "brain box" for the control is a new and attractive approach which is said to have solved the many complex control problems such as optimum air/fuel ratio over the entire speed and load range as well as starting, idling, load, and acceleration enrichment required during warmup and the smooth transition to the requirements of the warm engine. In addition to these requirements, there is the automatic altitude and temperature compensation and the all-important fuel cutoff during deceleration.

The electronic spray valve for metering the fuel is a rather old idea and was developed by Mr. Kennedy of the Atlas Imperial Diesel Engine Co. in 1932. An installation was made on a 6-cyl low-compression spark-ignition oil engine for marine service. This engine was exhibited at the New York Motor Boat Show in January, 1933. In 1934, a smaller oil engine was installed in a truck and was driven from Los Angeles to New York and returned and proved very successful. The description of the Atlas system is published in *Diesel Power*, February, 1933, and *Automotive Industries*, March 4, 1933. The Atlas system lacked proper automatic control. Of course, the knowledge of electronics was very limited 24 years ago, and I don't believe that the transistor was even conceived. It is in this field that Bendix has made the greatest strides, and we should give credit to the author and his associates for the unique adaption of a basically sound injection principle.

Comments on Both Fuel Injection Systems

—M. J. Kittler
Holley Carburetor Co.

BOTH the General Motors paper and the Bendix Aviation paper on their respective fuel injection systems were of great interest. General Motors is to be congratulated on being the first American manufacturer to offer a fuel injection system to the general public on their cars. Bendix Aviation is to be commended for introducing an electronic system as a part of the control mechanism for an automotive fuel injection system. The conception of the electronic control of metering introduces a whole new dimension to the handling of the basic problem of fuel metering control. The future development of electronic controls for fuel metering systems should prove to be very interesting.

As a general comment on both papers, I was rather pleased to note that the claimed improvements with respect to power and economy were held to fairly reasonable limits. It is my opinion that a great deal of early publicity in both the technical press and the general press was highly tinged with optimism, which cannot help but result in disappointment when the actual results are appraised. Gains of the order of 5 to 10% in power and economy appear attainable with a good fuel injection system. Gains in excess of this would arouse some suspicion in my mind at least. It may be well to mention here that it is only fair that fuel injection power data be compared to carburetor data attained with the use of a dual 4-barrel carburetor installation.

I was also pleased to note that fuel/air ratio indicated in the papers, and also distribution variations indicated were in a region not unfamiliar to those acquainted with good carburetion practice. Distribution accuracy between cylinders of 5% total spread, or better, is not easy to obtain, regardless of what kind of a fuel-feed system is used. In the case of fuel/air ratios there is no readily apparent reason why fuel/air ratios required with fuel injection should be appreciably different than those required with carburetors, assuming that a similar range of mixture distribution variations is attained in both installations.

There was a comment relative to speed-density metering versus mass airflow metering. Carburetors operate on the principle of mass airflow metering and the General Motors system, following this same principle, is subject to similar limitations. If the basic venturi size is adequate for maximum airflow at minmum pressure drop, then the metering forces available at low speeds will be exceedingly minute. For this reason we feel that speed-density methods offer better control of metering force than mass airflow methods.

The construction indicated for the fuel nozzles in the General Motors system raises the suspicion that these nozzles may be subject to icing under adverse conditions. These nozzles are carefully insulated to prevent heat absorption, and yet they are, in fact, expansion nozzles and will therefore produce refrigeration. I would like to ask if this condition has been investigated.

The emphasis on the use of rampipes at the intake ports carries overtones of racing engine thinking. As has been so ably pointed out, rampipes are tuned to certain speeds and are only of benefit under wide-open throttle conditions. Whether or not rampipes are used is of no consequence throughout the broad range of ordinary traffic and highway driving and ordinary passenger-car operation. Another obvious disadvantage of rampipes is their very length itself, which adds directly to the overall height of the engine. To be effective, the rampipes must have length, and when they have length they add unwanted dimensions to the engine envelope.

Referring to the Bendix Electrojector system, there are one or two theoretical questions that come to mind.

The first question has to do with the matter of reliability. It has been generally accepted that a hydromechanical mechanism possesses a greater degree of reliability than a parallel electronic mechanism. Undoubtedly great strides have been made in improving the reliability of electronic mechanisms, but I would like to ask what Bendix's experience has been in this area.

The second question has to do with the inertia—mechanical, electrical, and hydraulic—of the fuel and the solenoid valves at high engine speeds. It would appear that inertia effects could become quite pronounced at engine speeds upwards of 4000 rpm.

Since the concept of using an electric brain box for control of fuel metering is so new, it is difficult to envision how easy it would be to change the calibration curves to suit the requirements of different engines. Engine requirements vary widely, and I would be interested in hearing a discussion of the method used to calibrate the "brain box" to meet the engine needs.

It may well be that these two very interesting papers will mark the start of an extended period of evolutionary development of fuel injection systems for motor vehicle engines.

Questions Aspects of Fuel Injection Systems

—E. R. Mason
Chrysler Corp.

THESE comments refer to General Motors fuel injection system. We agree with Mr. Dolza's conclusions that continuous-flow port injection is the most logical and economical system for volume passenger-car production. We also agree that by utilizing a system of this type engine performance is in no way compromised when compared with timed injection of either the port or direct cylinder type.

Before discussing the mass flow system I would like to call attention to the statement that speed-density metering places exacting requirements upon the fuel supply pump. It is stated that the speed-density system requires a pump with constant delivery characteristics. With this we must disagree. The end result of speed-density metering is a controlled fuel delivery nozzle pressure. The speed-density fuel control may be regarded as a pump pressure regulator. So long as the pump is capable of delivering quantities in excess of engine requirements and at pressures in excess of nozzle requirements no other specification need be imposed.

Mass Flow Control

The General Motors arrangement is interesting and obviously is a carefully planned endeavor to develop a practical automotive system. However, the schematic of this system leads to a comment regarding the fundamental approach which was employed. The basic attraction of a mass flow system for automotive use has been that it required no engine-driven components. This feature has been weighed by many against the obvious drawback of mass flow metering. That drawback is lack of adequate control signal at idle and low engine outputs. This performance region is of extreme importance in passenger-car service. In the case of the General Motors arrangement it appears that benefit has not been taken of the most attractive feature of the mass flow system, namely, by incorporation of an engine driven pump.

Fig. 11 shows mixture ratios and their means of attainment in the low output region. It is interesting to note that control of starting, idle, and road load power requirements is not covered by the mass flow system through 50 mph. Throttle ports and differential vacuum signal manipulations provide the basis for fuel metering up to about 220-lb of airflow per hour. A third comment regarding the mass flow system is its well-known tendency to be influenced by pulsating airflow. Can eight individual ram tubes be gathered into a single volume without a resulting pulsation which interferes with desired metering characteristics of the venturi?

Vapor Handling

Throughout, much attention has been given to vapor handling characteristics. This attention we believe is well placed. Vapor handling and control of vapor formation is one of the most difficult and important features of an automotive fuel injection system. I believe many people will agree with me that the most difficult obstacle to development of a satisfactory passenger-car injection system is proper control of vapor. In Mr. Dolza's discussion of air density compensation, he states that proper utilization of fuel vapor bubbles at high temperature is used to cancel adverse air density effects. If this be true, we have met the master; for Mr. Dolza has trained his bubbles better than we have. When studying the vapor handling characteristics of the schematics, I looked in vain for a fuel bowl vent. Fig. 23, however, shows a manifold vacuum vent. The discussion indicates that this vent is restricted to avoid metering disturbances. Inasmuch as it has been indicated that a variation of 0.01 in. of water will affect metering, I do not understand how a controlled vent can accurately maintain bowl pressure anywhere near such limits. If it can, is it possible to handle the large vapor quantities which may result from coasting bypass or normal recirculation at high ambient temperature in so confined a volume as a fuel float bowl? It would seem that during periods of high vapor generation, this vapor augments the normal idle flow and under such conditions can an acceptable idle be maintained.

Vapor lock is mainly a problem of getting fuel from the tank to the fuel bowl. By utilizing an engine-driven diaphragm supply pump it would appear that this system has not solved the generally recognized vapor lock problem.

Nozzles

With regard to the injection nozzle construction and installation on the manifold, considerable importance is placed upon means of using evaporative cooling to control temperature of the fuel delivery line and nozzle as a means of controlling vapor formation. Our experience with low-pressure continuous flow systems has never given any indication that fuel-line temperature was of any importance. We have found that fuel temperature and vapor formation must be meticulously controlled until the metered fuel is delivered to the nozzle line. In the case of this system we wonder if an adverse feature has not been incorpo-

rated. It would appear that the refrigeration effect would be highly detrimental to engine operation under atmospheric conditions conducive to ice formation in conventional carburetors.

We wonder if the cold start, warmup, and hot-cranking fuel control has been found to be satisfactory. Our experience indicates that the hot-cranking control and the cold start and warmup device must be interrelated to cover all temperature requirements. Normal hot cranking requires a fuel flow of about 10 lb per hr. Cold cranking may require up to 35 lb per hr. To cover this range of fuel flow with temperature we believe the two devices must act together. In connection with this subject, we wonder if an electrically heated thermal warmup unit can ever be related to any significant engine temperature during either warmup or cooldown. We might also question whether elevation of the cruise fuel/air ratio to power mixture is sufficient to cover warmup requirements.

A last comment on operation of the system would be to question the conclusion that no accelerator pump is necessary. We have found that when the leanest possible road load mixtures have been reached, enrichment is required for solid engine operation during transient accelerating conditions.

Authors' Closure To Discussion

MR. KITTLER has pointed to the fact that gains in power and economy are shown only in the order of 5 to 10%. Perhaps we have been too humble in this respect. Most of the improvements shown are on a steady-state basis with carburetor heat blocked off, while actually we find our largest gains on a transient basis. Gains on the order of 15% are not uncommon for both power and economy. Mr. Kittler further suggests that we make all fuel injection comparisons to the dual, 4-barrel carburetor arrangement. This we would do happily because here is where fuel injection really shines. Our fuel injection system not only shows performance gains in the high-speed range but far surpasses the dual, 4-barrel carburetors in performance at low speeds from the standpoints of power, smoothness, and economy.

Mr. Kittler's remarks concerning mass flow metering limitations are correct as applied to carburetors where it is necessary to use the venturi depression directly as the pressure drop to obtain fuel flow. However, in our fuel injection system the venturi depression is accurately multiplied mechanically to obtain a much greater fuel pressure range. We have found mass flow metering to be the most satisfactory system.

Nozzle icing is not as serious a problem as it first appears to be. The refrigeration from the fuel, fortunately, is very effective at high ambient and fuel temperatures and is very small at low temperatures since it is directly related to the vapor pressure of the fuel. Our nozzle is designed to produce about 10-deg cooling at high temperatures and the small amount of refrigeration at 30 to 40 F ambient causes only a small amount of ice under certain conditions. This ice consists of a thin coating in the large air holes and causes a slight amount of fuel enrichment during warmup but disappears as the underhood temperature increases.

Regarding Mr. Kittler's remarks on rampipes, they are very effective in the 40- to 65-mph range of driving especially with automatic transmissions that shift down and permit engine speeds in the peak ram range.

Mr. Nystrom reports that timed injection offers the most advantages on both 2- and 4-cycle engines. We have found that continuous injection can be made for less cost without sacrifice of power and economy. It is true that timed injection will be best for 2-cycle engines, but we believe that the 4-cycle engines should not be forced to use timed injection and suffer cost-wise because of the 2-cycle engines.

Mr. Nystrom raises the question of nozzle clogging. Our experience during development is that dirt can be removed before assembly and the system will stay clean thereafter.

We are happy to learn that Mr. Mason has come to the same conclusions as we regarding types of injection; that continuous-flow port injection is the most economical system with no sacrifice in engine performance.

Our statement regarding the speed-density metering fuel pump was that it "usually requires a supply of fuel in equal amounts per cycle." We realize that there are some exceptions. For example, the engine speed can be sensed electrically, in which case the pump need not be uniform. Perhaps Mr. Mason is referring to a density metering system with no speed sensor.

It is possible with the GM fuel injection system to have the pump driven separately from the engine. However, we believe the direct engine drive is the lowest in cost and the most reliable.

In our mass flow metering system, we use manifold vacuum bleeds to obtain the enrichment required at idle and off-idle. Fig. 11 is an artist's attempt to illustrate the method rather than give exact data. Since the scales do not start at zero, the effect at first appearance seems larger than actually shown; for example at idle, where the bleeds make the largest signal modifications, the effect shown is less than 20%. Actually during our experimental work, we have had some units which required no idle bleed and some required no off-idle bleed.

We have found no disturbance to metering caused by the rampipe pulsations. In fact, we had one experimental installation with only four cylinders and a much smaller plenum volume between the air meter and rampipes, and metering was satisfactory at all engine speeds.

The control diaphragm in the GM fuel injection system is placed between two chambers: the upper chamber is subjected to the metering depression signal, and lower chamber is vented to the static pressure at the venturi inlet. In this manner, the diaphragm is affected by metering signal only and is not disturbed by vapor pressure or manifold vacuum in the float chamber. The lower diaphragm chamber is formed by using a very small hole around the diaphragm link and a large vent to the venturi inlet.

The fuel vapors are then vented from the float chamber to the manifold. Using "educated bubbles," we can alter the effect of the venturi signal so as to lean the mixture at high temperatures.

We have found that most of the vapor bubbles are formed when the fuel is admitted through the float valve; that is, when the pressure is reduced from about 6 psi to atmospheric pressure. During coasting, when the fuel is shut off and no new fuel admitted, there is very little vapor formation.

On hot-idle tests the GM fuel injection system has continued to idle indefinitely where otherwise identical carbureted cars have boiled and were stopped. In these tests, it was interesting ot note that carbureted cars boiled much more readily than fuel injected cars indicating lower heat transfer to the cooling system. As a result, we have found vapor lock much less severe with fuel injection even though we use the same diaphragm pump.

Engines equipped with GM fuel injection system have been found to be much less sensitive to flooding during cranking than with carburetors. The hot and cold starting methods described have been found to be satisfactory on a great number of such tests.

The warmup thermostat is primarily heated electrically by a heating coil energized through the generator armature. At idle when the engine warmup is slow the generator voltage is low, and there is less heat supplied to the thermostat. Also because of its location, the thermostat is somewhat sensitive to ambient air temperaure and engine temperature. This system has proved satisfactory on hundreds of warmup tests.

For warmup enrichment, we have found the air/fuel ratio obtained at the power mixture stop to be sufficient for baseline temperatures as low as −10 F. At lower temperature, a few seconds engine warmup may be required before complete flexibility of the engine can be attained, and this should not be objectionable for the sake of engine wear.

Concerning Mr. Mason's skepticism about our omission of the accelerator pump, we would like to comment that we do not set the control for "the leanest possible road load mixtures." We set our economy stop for the mixture which requires the least amount of fuel for road load. This best economy mixture is considerably richer than the leanest possible mixture at which the engine will run. At this best economy mixture and with the proper attention to details as explained in the paper, we do not need the accelerator pump.

Mr. Kittler has questioned the reliability of electronics. We have had no operational failures during our tests. We have full confidence in electronic components. Also, Mr. Kittler questioned the inertia effects on fuel lines. We have had some trouble, and this was our basic reason for changing to the common rail system.

THIS PAPER discusses the myths and fallacies regarding automation. According to the author, there is no such thing as an "automatic" factory, but merely degrees of automation.

Advantages of automation are discussed in detail, especially in regard to productivity increases and labor displacement.

MYTHS AND FALLACIES OF AUTOMATION

James R. Bright, Harvard University

This paper was presented at the SAE Annual Meeting, Detroit, Jan. 15, 1957.

AUTOMATIC manufacturing is an exciting prospect to many engineers and managers in the metalworking industry. It has proved its ultimate benefit many times, yet it is a simple truth that we have been so deluged with cliches about the impact of automation that it is very easy to be misled as to what to anticipate. The unqualified assumption that these things will inevitably accompany any automation effort is dangerous and misleading. It clouds the judgment of both engineers and managers, and it misleads and disturbs labor and the general public.

During the past two years I have had the opportunity to study in some detail the experiences of 13 highly automatic production systems in a variety of industries. The trends and results seen there have been checked against the experiences of perhaps another 30 firms in lesser detail. When we examine these industries carefully, instead of hypothesizing, the lessons of automation stand out much more clearly. The cliches with which automation speeches and literature abound are true enough—but only in part and only under certain conditions. It is the purpose of this paper to examine some of the fallacies and phobias of automation with view to encouraging all of us to think through the implications of automation more clearly and to eliminate some pointless fears.

There are industries—the process industries, and particularly oil refining—that have gone much further toward automatic operation than the metalworking industry. By studying things that have happened to these firms we can, perhaps, interpret and anticipate some of the ultimate effects of mechanization in the automotive field. Let us look at some of the more common myths and fallacies of automation.

— I —

Myth No. 1—There are fully automatic factories—It has been stated again and again in popular and serious literature that there are not only "automatic" factories, but there are "fully automatic" factories. For three years I have tried to track down such automatic factories and have been unable to find them. The automatic factory turns out to be, in every case, a remarkable advance in mechanization, but it has not resulted in manless production. It also turns out to be a tremendous hodgepodge of semantic confusion and exaggeration. Let us consider four classic examples:

The first is Oliver Evans, a remarkable mechanical genius of the post-Revolutionary War period who has become the patron saint of automation enthusiasts. He is often cited as having built the first automatic factory about 1784–1785. It is true that this was an extremely ingenious mill for producing flour from grain. It incorporated bucket elevators, screw conveyors, weigh hoppers, and a primitive form of belt conveyor combined to achieve a production line that was noteworthy for its high degree of automaticity. Was this the first automatic manufacturing effort? Definitely not. One has only to examine the work of Leonardo da Vinci to see that automatic production machinery was conceived at least 300 years earlier.

Was it really automatic? It is called such in almost every Sunday supplement article on automation; yet, exaggeration is at work even here. Evans wrote "The Young Mill-wright & Miller's Guide" to describe his theories and techniques for mill construction. Here he defined his contribution:

"These improvements consist of the invention, and various applications, of the following machines, viz. 1. The Elevator. 2. The Conveyor. 3. The Hopper-boy. 4. The Drill. 5. The Descender.

"Which five machines are variously applied, in different mills, according to their construction, so as to perform every necessary movement of the grain and meal, from one part of the mill to another, or from one machine to another, through all the various operations, from the time the grain is emptied from the waggoner's bag, or from the measure on board the

ship, until it is completely manufactured into superfine flour, and other different qualities, and completely separated, ready for packing into barrels, for sale or exportation. All which is performed by the force of the water, without the aid of manual labour, except to set the different machines in motion, &c. Which lessens the labour and expence of attendance of flour mills, fully one half. See the whole applied, plate VIII." [1]

It is interesting to note that Evans made no claim for "manless manufacture." He specifically pointed out that he had reduced the labor requirements "one half."

The second example is the automatic frame plant of A. O. Smith Corp. which was built in 1920–1922 and achieved operating perfection about 1928. It has been called "fully automatic" in Dr. Sigmund Giedion's remarkable study, "Mechanization Takes Command." A. O. Smith's then president, Mr. L. Smith, wrote in the *Magazine of Business,* February, 1929, a description of this plant under the title, "We Built a Plant to Run Without Men."

Over 30 years ago this plant turned out an automobile frame every 8–10 sec. This meant performing some 552 operations on each frame in about 1½ hr.

Now, there is no doubt that this is one of the most remarkable achievements in the history of production; however, it was far from fully automatic. The motion picture film of this factory still exists. By studying it one can see that men performed many production line operations and work-feeding activities in this plant. The article referred to specifically states that a work force of about 200 men was employed to operate the machine. This was about one-fifth the former force. Sensational? Yes, but not "fully automatic."

In 1945–1946 many rumors were heard about the government-owned 155-mm shell plant in Rockford, Ill., built by the W. F. and John Barnes Co. It was reputed to be "fully automatic." In the Jan. 22, 1955, issue of *The Saturday Review* an article appeared entitled "The American Factory and Automation." It was authored by the chairman of the board and included this quote . . . "Our own company, U.S. Industries, Inc., operates perhaps the only fully automatic factory now producing. It is a government-owned, 155-mm shell plant at Rockford, Ill. The plant was designed for very high production, using 20% less manpower, both direct and indirect in conventional plants . . ."

If it uses only 20% less manpower than a conventional plant, is it "fully automatic"?

As a final example let us take Ford's Cleveland Engine Plant. In *Business Week* an article appeared describing Ford's Cleveland Engine Plant. The article was entitled, "Automation: A Factory Runs Itself." During my visit to this plant in mid-1954 I found that over 4500 people were at work in this plant and more than 2700 of these were on the production line. This 2700 did not include quality control personnel, material handlers, or the maintenance force. It is somewhat difficult to understand how a factory with 2700 production line workers can be said to "run itself."

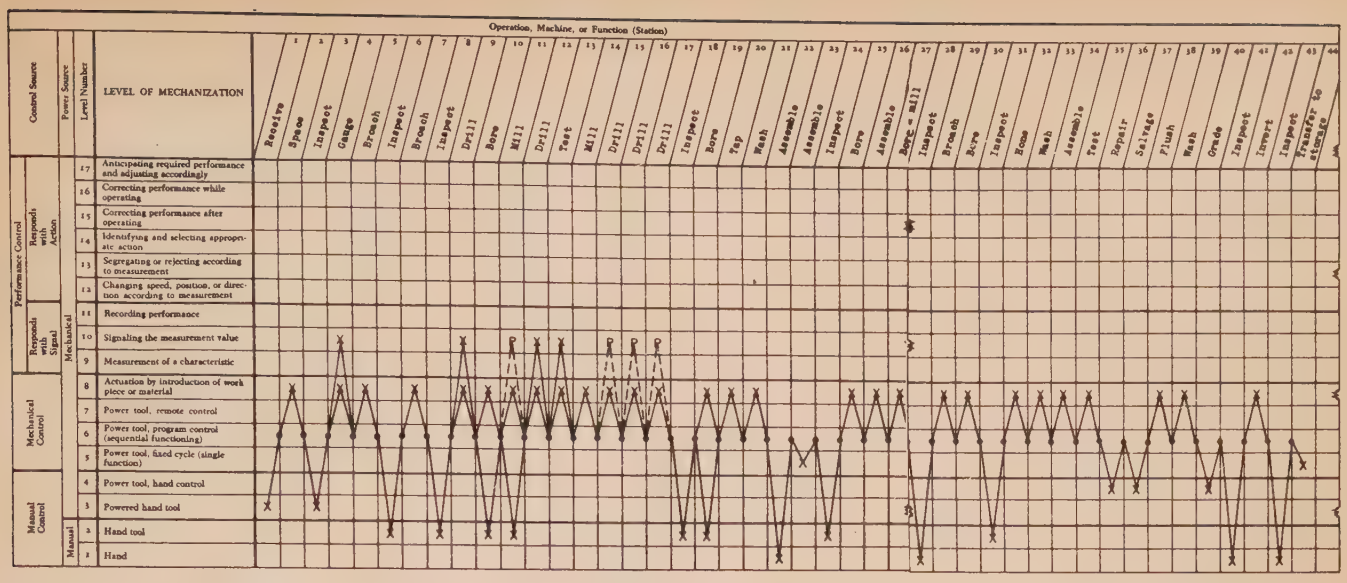
It is very evident that the "automatic factory" is not so automatic after all, and that there has been a tremendous amount of exaggeration and distortion floating around. These plants called "automatic" or "automated" are only "more automatic" than something that went before. There are hundreds, even thousands, of people working in them. Whatever "automation," "automatic," or "completely automatic" should mean, it is clear that these words are not being used to mean absolutely devoid of human assistance in the performance of the production activity.

As a matter of semantics, it is not easy to say what an "automatic plant" should mean. Do we mean that production line workers do no labor at their machines and that they merely "patrol" or "monitor" the system, or that there are no production line workers but only maintenance and set-up workers? Or do we mean that there are very, very few production line workers, or that there are also no workers at either end of the production line? In other words, that the introduction of raw materials and transfer of finished goods

[1] "The Young Mill-wright & Miller's Guide (Introduction to Part the Third)," by Oliver Evans. Printed for and sold by the author, Philadelphia, 1795.

Table 1—Levels of Mechanization Showing Character of Power and Control in Machinery

Initiating Control Source	Type of Machine Response	Power Source	Level Number	Level of Mechanization
From a variable in the environment	Responds with Action Modifies own action over a wide range of variation	Mechanical (Nonmanual)	17	Anticipates action required and adjusts to provide it
			16	Corrects performance while operating
			15	Corrects performance after operating
	Selects from a limited range of possible prefixed actions		14	Identifies and selects appropriate set of actions
			13	Segregates or rejects according to measurement
			12	Changes speed, position, direction according to measurement signal
	Responds with Signal		11	Records performance
			10	Signals preselected values of measurement (Includes error detection)
			9	Measures characteristic of work
From a control mechanism that directs a predetermined pattern of action	Fixed within the machine		8	Actuated by introduction of work piece or material
			7	Power tool, remote control
			6	Power tool, program control (sequence of fixed functions)
			5	Power tool, fixed cycle (single function)
From man	Variable		4	Power tool, hand control
			3	Powered hand tool
		Manual	2	Hand tool
			1	Hand



NOTE: (1) Dots on lines between operations represent the level of the materials-handling device. (2) If an additional "X" is shown in a column, it represents a distinctly different function at that particular station —that station is on the designated level. (4) Almost every machine tool on Level 8 is a Level 6 mechanism, actuated usually, inspection before or after operation. (3) The letter "P" means that only a part of the operation done upon receipt of work from the materials-handling device.

Fig. 1—Mechanization profile of cylinder block line

into storage and/or carriers is also done without human presence? Or do we mean that not only production, but storage, quality control, setup, and maintenance are done without people, or that product design, testing, laboratory analysis, scheduling, production control, and all the associated activities of manufacturing also are done without human assistance, or that there are no human beings whatsoever on the plant premises?

Obviously, these stipulations describe some impossible situations. However, they illustrate beautifully that before we can correctly describe a plant as being "automatic," we must agree on:

1. What activities must be mechanized.
2. Over how much of the manufacturing cycle the mechanization must extend.
3. How far back into the secondary operational and conceptual activities the mechanization must extend.
4. Exactly what kind, if any, human contribution to the plant and machinery may be permissible and still allow the system to be called "automatic."
5. What kind of mechanization is "automation."

Since such agreement or definition is nonexistent (and probably impossible), everyone is at liberty to use "automatic" and "automation" to mean any kind or amount of mechanization he chooses. And he does! The result is the grand confusion we now have in which "automatic factory" and "automated plant" mean just about anything at all in the way of an outstanding mechanization achievement.

It is rather clear that we need a better understanding of the nature of mechanization. Elsewhere I have presented a detailed analysis of mechanization showing that it has many different levels which differ significantly from each other by the character of the power and the control accomplishment.[2,3] Space will not permit explaining this here but Table 1 shows the 17 levels that I have derived. Whether this or some other method of establishing degrees of difference in mechanization is adopted, it suggests that we could take this system and apply it to every action in the factory from receiving dock to shipping dock. We would then create what I have called a Mechanization Profile, which will show the level at which each activity is performed and how the level varies from action to action.

A portion of a typical mechanization profile is shown here as Fig. 1. Through such an analysis it is evident that mechanization has at least three important qualities which must be understood to discuss intelligently the spread of automaticity.

1. There are many levels or degrees of sophistication in mechanical action. These vary widely from operation to operation. Automaticity seems to increase in richness of response as the higher levels are employed.

2. Mechanization also has a quality which I have described as "span." When the production system as a whole is studied, we see that mechanization appears in a very spotty manner. There are "clusters of automaticity" around certain activities, but the span of this mechanization definitely is not comprehensive from door to door. For instance, even in those plants where movement is completely mechanized and automatically controlled, as in an oil refinery, functions such as maintenance, sampling, and data logging are not done automatically (although progress is being made in this direction).

From such studies we can appreciate that automatic manufacturing or "automation" itself, no

2 *Mechanical Engineering*, Vol. 78, July, 1956, pp. 620–624: "Mechanization Profile—Tool for Measuring Automation," by J. R. Bright.
3 *Harvard Business Review*, Vol. 33, July, 1955, pp. 101–111: "How to Evaluate Automation," by J. R. Bright.

Table 2—Motives and Benefits of Automation in 13 Firms

Characteristic	Number of Plants Experiencing This Characteristic as: Primary Motive	Secondary Objective	Unexpected Benefit
Increased capacity	11	2	
Reduced direct labor content per unit of output	12		
Reduced indirect labor	8		3
Shorter lead time	2	1	3
Higher quality of product	8	1	2
Reduced scrap	1	2	7
Increased equipment utilization	2		2
Ability to expand production without major work force increases	1		
Improved safety	1	3	1
More attractive working conditions	1	2	5
Easier production control job		1	2
Easier plant administration		3	
Lower maintenance cost	1	1	1
Lower capital investment	1		
Increased flexibility in products manufactured	1		
Less skill and training required of some workers		1	5
Shorter setup time			2
Reduced floor space requirements		2	2
Reduced work in process inventory	1		2
Easier housekeeping	1		6
Reduced insurance cost		2	
Sales promotion value of new plant			3

matter what definition the enthusiast chooses to apply, becomes simply a matter of degree of mechanical sophistication and of the addition of more functions to the mechanization sequence.

Let us recognize, then, that while amazing and highly creditable technical accomplishments have created plants that are noteworthy for their degree of automaticity, it still is a matter of degree. The vast majority of so-called automatic plants have a substantial number of employees. The "fully automatic" factory still is a myth and appears to be a very limited industrial concept.

3. The prevalence of automatic production throughout an industry is a third aspect of its impact. There is a tendency to take a few exceptional plants and assume that they are typical when they are anything but!

— II —

Myth No. 2—Automatic manufacturing requires feedback control—A favorite phobia of the Sunday supplements is that automatic factories and automatic machinery have suddenly become possible and tremendously more significant because of feedback control. The reason this is the Second Industrial Revolution is that the "self-correcting" machine and the "decision-making" machine have been made possible by feedback control developments. Automatic production — automation — automaticity — are largely credited to the feedback concept. If we define feedback as the regulation of a machine by its own analysis of its output (closed loop control) we find that there is either none or remarkably little of it in most automatic production machines. This can be proved by taking a number of automatic sequences and examining them step by step to see if and where feedback appears. I have examined several hundred automatic machines in about 30 highly automatic production systems and found only oil refineries to show a significant amount of feedback control.

It is true that much automatic machinery does use elaborate and intricate electrical, hydraulic, pneumatic, or electronic control systems. However, these systems largely are programming devices: that is, they guide or limit the machinery to perform a sequence of prescribed operations. They very rarely correct machine adjustment in terms of output. Very, very few machines are truly self-correcting.

It is evident that feedback does not appear often because it is not necessary to much automatic production. Feedback control assumes that the machine needs to be corrected. But many machines do not need correction. They operate with quite adequate precision; their product falls well within the desired specifications. Since the need for self-correction is not great, feedback has no economic value.

Thus, we should not assume that automatic manufacturing always requires extremely sophisticated forms of control. Most highly automatic production systems are based upon the integration of production and handling devices, directed by control systems that direct a program of desired events.

This is not to deny the importance of feedback control in many of the continuous flow operations. There is no doubt that it will spread and make possible superior manufacturing systems and products. Yet control, alone, does not create automaticity.

— III —

Myth No. 3—The only motive for automation is cost reduction through the elimination of labor—It is perfectly true that many firms justify automation programs through labor savings alone; and it is equally true that this is a very short-sighted procedure. The experience in the 13 firms studied showed an astonishing range of motives as well as unanticipated benefits and secondary objectives that were not quantified. These are listed in Table 2. There are several important points to be made about this table (which is by no means complete for all plants).

The advantages of automation are far greater in number and often in value than the average firm realizes. I would estimate that not one firm in a hundred thoroughly explores the favorable (and negative) aspects of automation proposals. They rarely quantify anything but labor savings, and hence, overlook some splendid and valuable contributions.

Consider lead time, for instance. A parts firm cut lead time from 19 days to 2–3 days; an oil refinery from 4 days to 5 hr; a feed mill from 2 man hr to 5 man min. What would this mean to the average firm in customer service, in inventory reduction, and in production planning?

Another firm justified their automation program solely on reduction of scrap. Still another adopted automation solely to provide working condition im-

Table 3—Factors Influencing the Capital Cost of Automation as Contrasted to An Equivalent "Conventional" Production System (Installed)

Tending to raise the cost differential

1. Larger amounts of machinery because of mechanization of more functions and on a superior scale of speed, quality of output, etc
2. Greater complexity of machinery
3. Higher percentage of control devices
4. Higher proportion of novel elements requiring more engineering-development work
5. More testing and rebuilding expense in development
6. Larger and more difficult installation job
7. Difficulties encountered because stranger firms must integrate their physical systems and diverse technical arts
8. Longer debugging time
9. More scrap and lost production during the troubleshooting period
10. Need for extra production facilities while defects are corrected
11. Cost of correcting mistakes made in an overhasty program

Tending to lower the cost differential

1. Less volume of machinery due to
   - A. Compounding of more functions on fewer machine bases
   - B. Higher production obtainable from smaller but faster machines
   - C. Higher machine utilization and efficiency brought about by mechanical work-feeding and interlocking control
2. Less building space and facilities due to
   - A. Less volume of machinery
   - B. Reduced in-process storage space and storage equipment requirements
   - C. Less space required for operators
   - D. Fewer traffic aisles required since the material is mechanically moved within the production machine
3. Extent to which the "conventional" system is already highly automatic
4. Extent to which the automation system has been perfected by prior installations

provement. Automation made it possible for them to attract and hold women workers in a very tight labor market.

Engine plants have found that automation increases equipment utilization from about 60 to 80% This is a clear gain of one-third in production capacity without corresponding investment or plant expansion!

Some of these other motives appear quite strange—product flexibility, for instance. Yet the fact is that a fertilizer plant adopted automation because it enabled immediate pushbutton mixing of any of 500 formulations. Such product flexibility is totally beyond the conventional plant.

Reduction in capital investment also is completely contrary to traditional automotive-industry thinking about automation. The fact is that this was the major reason for automation in the most automatic plant I studied (an oil refinery). Here is a possibility that rarely enters management's or the engineer's calculations. We will discuss it below.

Lower maintenance cost is also a surprising advantage. We shall see why it, too, may arise out of automation.

The lesson is very clear: it is a mistake to think of automation as yielding a limited, fixed set of advantages and disadvantages. These vary widely and wildly depending upon a host of factors in the individual case. Management should carefully explore the full range of implications and quantify them wherever possible. Labor saving alone is a very naive and inadequate approach.

— IV —

Myth No. 4—Automation must cost more than a conventional system—This definitely is not always true. The oil refinery I studied adopted automation primarily because the conventional plant would have cost $1000 per bbl of daily output, while the automated plant cost only $700 per bbl. How can this be? Simply expressed, the automated plant was an integrated system eliminating much in-process tank storage and associated piping and pumping mechanism, as well as requiring only one control center instead of four. Thus automation costs less not because the equipment was cheaper, but because there was less of it.

This happens in many industries; and when the shrinkage of equipment volume is great enough, building and facilities requirements shrink too. Here additional great reductions may occur.

In any automation program we are contrasting the difference between a specific existing situation and a specific proposal. There is a mix of cost-raising factors and cost-reducing factors. The net impact on investment requirements is by no means certain to be greater. It all depends upon the relative weight of the factors shown in Table 3.

The automotive industry normally faces heavy development costs because each piece of automation is a unique bit of machine design. Consider, though, that many of the process industries advanced plants are unique assemblages of fairly standard components. When automatic production systems can be assembled out of relatively standard items, these engineering costs are enormously reduced. To the extent that this trend is occurring in the automotive industry, we can expect the differential between "conventional" and "automated" to shrink.

And it is only fair to remind ourselves that many so-called higher costs of automation actually are the costs of inadequately planned and organized automation efforts. Carelessness, ignorance, and plain human frailty may be at fault, not automation.

The truth here is that the cost of automation is not so much proportional to the amount of automaticity, but to the amount of novelty and uniqueness in what we are attempting.

— V —

Myth No. 5—Automation inevitably raises maintenance costs—Many automotive and electrical

Table 4—Factors Influencing the Severity of the Maintenance Problem of Highly Automatic Production Systems

Tending to Increase Maintenance

1. Increased mechanization across the span of the production system
2. Higher proportion of moving parts
3. Greater complexity of the machinery
4. Greater complexity of the controls
5. More novelty, uniqueness, or degree of innovation present in the mechanism (and its control)
6. Increased extent to which the machinery and control mechanism employs new technologies, particularly electronics
7. Higher proportions of electro-hydraulic-pneumatic actuation of controls and machinery as opposed to mechanical linkage
8. Increased extent to which the machinery operates near design limits of speed, temperature, precision, capacity
9. Inexperience of the maintenance force with the new machinery and controls

Tending to Decrease Maintenance

1. Reduction in total amount of machinery through smaller size of plant and compounding of production equipment
2. Development of radical improvements in controls
3. More design and engineering experience with the automated machinery concept
4. More operating and maintenance experience with the automated machinery
5. Increased proportions of "standard" components in the automatic machinery
6. Simplification and standardization of control mechanisms
7. Mechanization of trouble prevention (safety switches, etc) and use of trouble-signalling systems that forestall serious failure and damage
8. Mechanization of portions of maintenance job, such as automatic lubrication and adjustment for wear
9. Techniques that speed the maintenance man's comprehension of the control system such as color coding of circuits, testing circuits, detector panels, etc
10. Constructions that facilitate rapid removal and replacement of parts and assemblies or that permit easy observation of the functioning in actual use
11. More effective organization of the maintenance activity
12. The extent to which the automation reflects a substantial shift from old, worn equipment to new equipment
13. Adoption of a vigorous preventive maintenance program in the new plant

Note: This listing contrasts some of the more outstanding factors that affect maintenance in automated plants. The net impact on a given plant is a blend of the effects of these factors. The severity of the impact will vary in contrast to the net maintenance situation in the former plant. This list is not all-inclusive, of course, but it clearly shows that there are many things working to reduce maintenance, as well as to increase it, in the automated plant.

equipment manufacturers will insist that this is so, and it often is. Yet the two most automatic plants I studied actually reduced their maintenance costs and difficulties by automation!

Oil refineries generally expect that the maintenance force will be 50 to 60% of the total refinery work force. In the automated refinery mentioned above the maintenance force is about 20.5% of the total refinery force. Why? It is because there is significantly less machinery—pipes, valves, controls, and pumps —to maintain per unit of output. The maintenance is not directly proportional to the amount of automatic equipment or the degree of automaticity, but to the total amount of machinery. Reduce this machinery enough and maintenance will fall off, too.

The feed mill I studied is a pushbutton operation in which one foreman, one operator, and one truck driver receive, produce (mix), and distribute 90 tons of feed per 8-hr day. Formerly eight men were required by the same plant to produce 30 tons per 8-hr day. The control system is housed in a cage 7 ft high, 3 ft deep, and 8 ft long. It is a mass of wires, relays, and contactors required for the following operation.

The operator sets a dial to the right weight for each item to be mixed. He sets another dial to the total tonnage wanted. He sets a selector switch to divert the finished mix to the proper bin, and then pushes a button to start the action. The ingredients are weighed out into proper portions, mixed in 2-ton batches, moved into storage, and the cycle repeated until the desired tonnage is accumulated.

The effect of automation in this plant is that the one maintenance man formerly employed has been discharged! There has been no failure in 18 months of operation.

How is it that a small farmers' cooperative in an Indiana country town can "go automatic" with the greatest of ease, while the automotive giants literally spend fortunes on maintenance?

It is because maintenance is not proportional to automaticity. Maintenance, like capital cost, is made

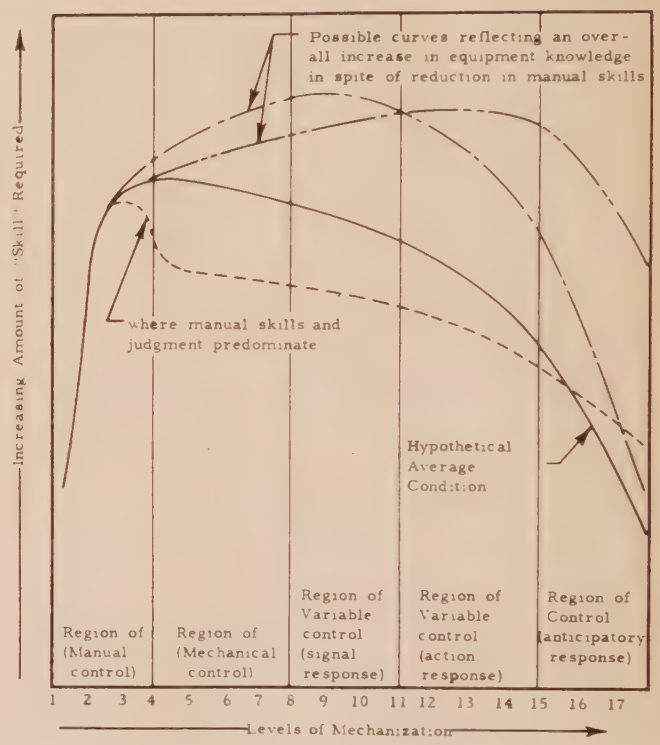


Fig. 2—Graph showing how skill requirements of operators vary with levels of mechanization. Skill means indefinite blend of special training, practice, job knowledge, decision making, and dexterity that are implicit in phrase "skilled worker"

up of many factors. Well-proved standard equipment can be highly automatic and quite trouble-free. It is the novelty in equipment that results in trial and error, and hence costly debugging and frequent failure until prefected. Automatic machinery can be perfected until it has a very high degree of reliability.

There are other factors which affect maintenance and these are shown in Table 4. We must not forget that maintenance techniques are improving, too. Management should try to appraise the mix of these factors in their own case before assuming dire main-

tenance troubles as the inevitable result of automation.

Another misleading thing about maintenance is that the figures and experiences reported are totally misleading. The thing called maintenance usually is a blend of: 1. Installation. 2. Debugging. 3. Routine servicing (lubrication, cleaning adjustment). 4. Downtime for tool replacement and the like. 5. Housekeeping. 6. Maintenance of tooling fixtures (as distinct from the production machine). 7. Breakdowns.

I have found no plant that has figures isolating each of these factors. They are all lumped together and charged to maintenance. Thus "maintenance troubles" in automated plants often turn out to be "installation" or "debugging" costs, and not a true picture of the normal maintenance load at all. Furthermore, we usually hear of "maintenance" as of one moment—the worst moment—and not after the plant has learned to live with the system. Thus maintenance is grossly exaggerated as a liability of automation.

– VI –

Myth No. 6—Automation inevitably means upgrading of labor skill requirements and severe retraining problems—It is one of the phobias of some elements of organized labor and some social scientists and industrialists that the average worker is unemployable in the highly automated plant. This study throws a great deal of doubt on the certainty of this end effect. The evidence in these 13 plants is mostly to the contrary. To the extent that employment opportunities are available (a) operators' jobs often are easier and more quickly mastered in the automated plant and (b) the impact on the quantity and quality of maintenance skills required may be much less than expected.

In fact, the prime management error in at least four plants studied was to assume higher skills were required on operating jobs than was so. Thus unrealistic wage rates were set on certain jobs that were unfair to other workers in terms of relative skills required.

Of course, there are many exceptions where knowledge, responsibility, education, and mental effort required of the worker increase. Nevertheless, analysis of many machines has preponderantly demonstrated that the higher levels of mechanization gradually or rapidly reduce the contribution by the operator.

Let us analyze the evolution shown in Table 1. First, there is the substitution of mechanical power for manual effort, which takes some burden off the worker. Then, as increasing degrees of fixed control yielding a desired machine action are provided, the worker does less and less guidance of the tool. As measurement is added to the machine, the operator has a portion (or much) of the control decision information obtained for him. As the machine is given higher levels of automaticity, more and more of the decision making and appropriate follow-up such as the selection of the necessary production action, control of the production action, and correction of performance are done by the machine. The more superbly automatic the machine, the less the operator has to do.

We must then recognize that a worker's contribution on a job embraces more than "skill," and that "skill" itself is a rather indefinite blending of several things—manual dexterity, knowledge of the art, knowledge of the theory, and comprehension and decision-making ability based upon experience.

The impact of higher levels of mechanization leads to what I have termed roughly as the "hump in skill requirements," as seen in Fig. 2. "Skill" is used broadly as meaning the combination of the necessary experience, dexterity, and requisite technical knowledge. I do not pretend that the shape of this curve is anything more than a crude suggestion of a general concept to which there are many exceptions. Nor do I contend that the starting and terminal points and peaks are relatively correct. It is true that the highest forms of skill frequently occur on Level 2, where the hand tool operator expresses the ultimate in craftmanship. (Is this not the level of the sculptor, the painter, the musician?) Furthermore, one could argue that there are separate curves for each of these elements of "skill," and that these curves are not identical or even similar. Thus, each kind of job would reflect its own peculiar blending of these individual element curves. These summation curves doubtless would vary considerably between various jobs.

Still, it seems evident that even a qualitative examination, as suggested in Table 5, will confirm a changing "skill" content—increasing severely, then gradually decreasing with the higher levels.

By its very definition the truly automatic machine needs no human assistance for its normal functioning. "Patrolling" becomes the main human contribution. The "operator," if he is still there, becomes a sort of watchman, a monitor. We might call him a liaison man between machine and operating management.

Thus, the progressive effect of automation is first to relieve the operator of manual effort and then to relieve him of the need to apply continuous mental effort. At times the mental effort is increased because of the alertness and overall responsibility required. Yet the evolution of machinery shows that eventually safety devices, and various recording and signalling systems are added to reduce or eliminate this demand.

The ultimate argument, of course, is to point out that this machine evolution leads to the very situation feared: automaticity grows until no operators are required. Then what employment opportunities exist? Only maintenance and design. The level of education required on some of these design jobs is of PhD caliber. The maintenance skills may require electronics

training. Now the unskilled or semiskilled worker is helpless. He does not even understand the language, let alone the job.

This grim ending neglects to account for several counteracting facts:

1. It is the trend of invention to mechanize difficult jobs. Thus the maintenance job is continually being simplified by innovations such as the trouble-shooting control panels which reduce skill and training requirements, and the routinizing of many theoretically complex jobs, such as had actually been experienced in computer work. The design job is simplified in part, too. For instance, fantastically difficult mathematical jobs have been made relatively easy by the application of computers. And, as previously shown, the maintenance job itself is simplified by improved machinery that is less failure-prone.

2. The economics of machinery bar the introduction of fully automatic machines in hundreds of thousands of installations. Man is a far cheaper and more flexible control device than the ultimate "feedback" mechanism for many machines. Thus, the productivity and automaticity increase, but people still are required to monitor machine action or provide minor production assists. In the inspection of visual flaws and a variety of observations it is hard to beat a man.

In brief, there is an enormous technological gulf between "automatic" and "fully automatic." Unless the machine really is "fully automatic," people are quite necessary to its operation.

3. Industry is by no means the mass production world so often pictured. The mixing of parts, products, and models required by many sales situations is absolutely astonishing. An electric motor manufacturer makes over 12,000 variations in six basic horsepower ratings from 1 to 15 hp. From ¾- through 3-in. diameter, a ball-bearing manufacturer makes 18,000 models. An engine plant makes 17 models of engines in what is theoretically a simple 2-model line. An oil refinery makes 400 kinds of oils. This range of variety will not support automatic machinery to the ultimate.

There are at least three factors that may increase skill requirements: (a) the assumption of responsibility for more of the production sequence (that is, a number of different kinds of machines), (b) the assumption of responsibility for higher caliber functions like setup and inspection, and (c) new kinds of preparation or "setup" jobs such as programming tapes for machine tool control.

On the whole, however, I found that automation generally reduces the demand for skill and knowledge on the part of the operator after reaching Mechanization Level 5 because the machine takes over more and more of these elements of the task.

Thus, automation does not tend to make people unemployable because they lack skill and specialized education. On the contrary, many (and perhaps most) *operator* jobs can be mastered more quickly, with less job experience, special skill, and education as mechanization increases past Level 4 and provides increasing degrees of automaticity.

While space will not permit a further discussion here, the study of labor in our 13 plants generally confirmed this theory. Perhaps the most valuable conclusion for today's management will result from the simplest of questions:

Did the skill requirements on these automated lines create any major problems for management?

1. Automation did not force management into drastic measures because of increases in operator skill requirements.

2. There was and is a scarcity of maintenance electricians trained in electronic work, but only something like a tenth, or perhaps a fourth, of all electricians currently need this training.

3. Hydraulic and pneumatic repairmen of adequate ability are not common, but relatively few firms ran into difficulty here.

4. Those firms that designed and built their own machinery see a distinct shortage of machine designers, engineers, and mechanics with machine-building skills.

Otherwise, there is little evidence that large numbers of distinctly superior skills are required to man the automated plant.

While we might discuss many theoretical interpretations of the causes and counteracting forces in the skill area, the practical questions for today's businessman are (a) "If I 'go automatic,' can I run the new plant with my present work force?" and (b) "Will I have to face a difficult retraining problem?"

Judging from our 13 plants the answer to the first question is, "yes" and to the second, "no" with due allowance for minor exceptions and some, but not all, of the maintenance job. Furthermore, there is a good possibility of lowering skill requirements and shortening the training period on many operating jobs.

— VII —

Myth No. 7—Automation has nothing to offer labor but unemployment—It seems true that many (and probably most) automation programs aim at reducing direct labor content per unit of output. However, most automation programs also are sparked by the need for increased capacity (Table 2). Depending upon the amount of labor reduction and the amount of capacity increase required, the net effect on labor can be anything from unemployment to far greater employment opportunities. (See Table 6.)

As an outstanding example, Ford has officially recognized and organized for "automation" since 1947; yet, today Ford employs about 40% more people than in 1947. This certainly is not because of automation, but it is in spite of it!

Another fact is that the labor impact often is distorted by the way the percentage labor savings are expressed relative to a minute part of the operation.

Table 5—Suggesting the Changing Contribution Required of the Operator with Advances in Levels of Mechanization

Worker Contribution[a] or Sacrifice Traditionally Receiving Compensation	Mechanization Levels			
	1-4 (Hand Controlled)	5-8 (Mechanically Controlled)	9-11 (Variable Control, Signal Response)	12-17 (Variable Control, Action Response)
Skill (Knowledge of Art)	Increasing	Increasing	Increasing-Decreasing	Decreasing-Nil
Skill (Dexterity)	Increasing	Increasing-Decreasing	Decreasing	Nil
Education (Knowledge of Theory)	Increasing	Increasing-Decreasing	Decreasing	Nil
Experience (Practice, Awareness, Comprehension)	Increasing	Increasing-Decreasing	Decreasing	Decreasing-Nil
Physical Effort	Increasing-Decreasing	Decreasing	Decreasing	Decreasing-Nil
Mental Effort (Concentration, Attention)	Increasing	Decreasing	Decreasing	Decreasing-Nil
Productivity[b]	Increasing	Increasing-Decreasing	Mostly (but not always) Decreasing	Nil
Exposure to Hazards	Increasing	Decreasing	Decreasing	Nil
Acceptance of Undesirable Job Surroundings	Increasing	Decreasing	Decreasing	Nil
Responsibility[c]	Increasing	Increasing	Increasing-Decreasing	Increasing, Decreasing or Nil
Seniority	Not affected	Not affected	Not affected	Not affected
Decision-Making (Alertness, Judgment)	Increasing	Increasing-Decreasing	Decreasing	Decreasing or Nil

[a] Refers to operators and not to setup men, maintenance men, engineers or supervisors
[b] Ability to influence amount of output
[c] Safety of equipment, of the product, of other people

Table 6—Employment Before and After Automation in 13 Plants[a]

Plant	Employment		
	Net Addition	Displaced but Absorbed Elsewhere in Firm	Net Displacement
Engine Plant A	5000 (approximate)		
Engine Plant B		350	
Parts Mfg. A		No Change	
Parts Mfg. B	2000 (in last 4 years)	70	
Parts Mfg. C		10% approximate	
Plating Plant A	2		
Plating Plant B		22	
Bakery			250[b]
Coal Mine	12		
Oil Refinery	160		
Feed Mill			5
Fertilizer Plant			2
Foam Rubber Plant		18	

[a] Note: It is a ridiculous oversimplification to assume that automation alone accounts for these changes. This table merely reflects the status after automation, but not the mix of factors causing the status (opening of new plants, product mix changes, new products, sales increases, new equipment, etc).
[b] Largely caused by consolidation of widely scattered old bakeries into one new plant. Employment today has crept up to approximately the original figure

In a foam-rubber mattress plant, for instance, displacement was quoted in the trade press as being 50% of the working force. However, on checking the figures closely, I found that displacement amounted to 45% of the people in the pouring operation only. They represented just 16% of the total employed on the foam-rubber mattress line. Further, this production line was only a piece of the entire plant's activities, and the number of displaced people amounted to about 1½ % of the total working force! (Absorbed, incidentally, in other operations.)

Again and again I have found that similar drastic percentage displacement or labor savings figures so widely quoted have been applied to very small segments of the total production activity. In the 13 plants studied, the amount of unemployment created was not significant except in one instance. Of course, this sample is by no means valid or necessarily typical for all industry; but as a sample it is suggestive.

Although the net effect on employment opportunities is uncertain, there is no doubt at all that automation offers the worker distinct job advantages. For operating labor, automation means:

1. Easier work, physically and mentally.
2. More interesting work through variety of job activity.
3. A more satisfying job through a sense of responsibility for a larger portion of the end result.
4. Pleasanter job surroundings.
5. Higher pay through increased productivity.
6. The satisfaction and prestige of working in an outstanding plant.
7. Greater job security against both the efforts of the firm's competition and the effects of old age on the individual's strength, dexterity, and endurance.
8. Greater continuity of employment.
9. A safer job.
10. The pride of running a fine machine.
11. Opportunity to learn more of the total production process and machinery.
12. Experience with advanced equipment that may be even more valuable as time goes on.

Of course, all these things do not apply in every situation. The point simply is that the worker has much to gain from automation. Why not tell him so?

— VIII —

Myth No. 8—Automatic machinery alone is responsible for sensational advances in automatic manufacturing and productivity increases—Study of manufacturing evolution in any industry will show that at least five major factors are involved in technological improvements:

1. Spatial rearrangement of machinery to provide a more efficient sequence of activity. As an example, consider Ford's original efforts to create moving lines for chassis assembly [4]

August, 1913—Stationary assembling locations for each chassis. Labor time: 12 hr and 28 min per chassis

August, 1913—Rope and windlass traction to pull chassis down a 250-ft assembly line; six assemblers traveling with the chassis. Labor time: 5 hr and 50 min per chassis

Oct. 7, 1913—Moving assembly line 150 ft long, 140 assemblers at fixed locations. Labor time: 2 hr and 5 min per chassis

[4] *Engineering Magazine*, Vols. 48, 49, March-June, 1915, pp. 859–876, 67–87, 184–201, 372–393: "Ford Methods and Ford Shops," by H. L. Arnold and F. L. Faurote.

Dec. 1, 1913—Assembly line lengthened to 300 ft, 177 assemblers. Labor time: 2 hr and 38 min per chassis

Dec. 30, 1913—Two assembly lines, cars pushed by hand past 19 assemblers. Labor time: 2 hr and 40 min per chassis

Jan. 14, 1914—Endless chain drive for chassis movement: Results: "favorable"

Feb. 27, 1914—Chassis mounted on rails, moved by endless chain. Labor time: 84 min per chassis

April 30, 1914—Three rail lines. Average labor time per chassis for 1212 chassis: 93 min

Notice that productivity gains were fantastic although there was no introduction of automatic machinery. The gain was simply the refinement of a concept of arrangement and the eventual mechanization of movement (although great gains were made without even this simple bit of mechanization).

2. Changes in materials to facilitate automatic production. Materials may block or advance the automatic aspect of production techniques. It was not until nonsag crystalline-structure tungsten wire was developed that mechanized mounting of electric lamp filaments became possible. Nonsag wire made it feasible to use coiled filaments which, being short, could easily be mounted mechanically (instead of hand-stringing long filaments over multiple supports to get adequate filament length).

Bread bakeries are not continuous flow systems today because of the need to have the dough mix sit for several hours until the yeast rises. A new type of yeast material for rapid fermentation is under development. If successful in eliminating this delay, it will make continuous flow—hence greater automaticity—possible.

The conversion of coal to a slurry that can be pumped is a typical material change which enables a superior mechanization of the function (in this case, transportation).

Notice that new automatic machinery has not created the improvement. Rather, the shift to a material that can be more readily mechanized is the catalyst that results in greater automaticity.

3. Changes in process to facilitate mechanization. Some of the most dramatic production gains come from the adoption of new processes; for instance, stampings and castings versus machined parts.

The idea of dip soldering of TV and radio chassis assemblies, instead of one-by-one hand soldering the connections is a typical noteworthy process change contributing to automaticity and productivity.

4. Changes in product design. The product designer probably contributes as much to automatic manufacturing as does the machine designer. For instance, the idea of exhausting the electric lamp through the base of the bulb while sealing, rather than through the tip using a separate exhaust tube, was a major factor in doubling worker productivity about the 1920–1925 period.

Automatic TV chassis assembly progress is not so much the result of a new concept in automatic machinery, but a new concept in design; to wit, the use of etched wiring on a flat board to connect the leads of mounted components. In other words, the "printed circuit" design has been the contribution that opened the way to automatic production.

5. Automatic machinery. This is the most spectacular aspect of automatic production. However, it often is not a radical machine concept but rather the mechanization of an act that was made simple (or eliminated) by changes in arrangement, process, material, or design, as described above.

Thus, to look to automatic machinery as the sole hero or villain of automatic production, productivity increases and/or labor displacement is industrial naivete of the first order. We must not forget that the product designer who eliminates a part does something for productivity that surpasses any mechanization we might dream of. And when he so much as eliminates two of four fasteners, he decreases labor content 50% without spending one cent on equipment!

There are other myths and fallacies of automation that should be exploded for the good of our economy and society, and to help industrialists develop the implications of their mechanization programs in an atmosphere of understanding.

Let me point out again that these myths and fallacies spring in part from reality. It is a grave mistake to assume or insist that there is no truth at all in them. These negative aspects of automatic production do sometimes happen—but they happen because of the mix of factors present in the particular situation, not because they are immutable laws of mechanization and inevitable drawbacks of every automation program. It behooves the industrialists to analyze those factors; then he can make surer judgments as to the end results of automation in his particular case.

DISCUSSION

Emphasizes Certain Points about Automation

—R. L. Kessler
General Motors Corp.

CONCERNING the excellent presentation just made on the "myths of automation," Prof. Bright is to be highly complimented on a fine survey and study of this phenomenon which a large segment of the public is calling "automation." His study has been a systematic and timely observance since considerable writing has been done in recent years on this subject but, unfortunately, too much of this written material has either been misleading or misunderstood.

The myths and fallacies concerning this subject have been ably described and explained by Prof. Bright, and widespread publication of his paper should help in quieting ugly rumors and statements used in connection with the word, automation.

While this presentation listed many of the advantages as well as the fallacies, three points perhaps need even more emphasis.

1. Quality is always improved. Experience has taught us that without exception the use of automatic methods to improve manufacturing processes will improve quality. For example, an automatic feeder of simple parts such as screws and rivets can-

not feed a bad part which would impair the quality of the product.

2. Safety. It is of prime importance to industry in general, and certainly the automotive industry is no exception, to strive constantly to improve the working conditions by the elimination of safety hazards. It is only natural to assume that the farther away we can keep any part of a person's body from equipment and machinery the chances are far better for a perfect safety record.

3. Cooperation between product design and production engineers. The necessity of mutual understanding between the product engineers and the folks responsible for the automatic manufacturing processes cannot be overemphasized. The degree of producing parts by this means oftentimes depends upon certain changes in product design.

Finally, the concept of doing or fabricating things by better means and methods has been the trend since the beginning of time. Man has always striven to improve means and methods. And his successes, particularly in America, have resulted in more and better things and jobs for all of us and has given us the highest standard of living of any nation in the world.

Until the word automation cropped up in our language, we referred to these achievements of man as "progress." Since Prof. Bright has so ably depicted the myths and fallacies connected with this new word, automation, perhaps it is of advantage to discard this misnomer because it is too often misunderstood. We should refer to this manufacturing phenomenon by its true meaning, "progress" or "technological progress."

Discusses Consequences Resulting from Automation

—S. K. Rudorf
A. O. Smith Corp.

I CAN'T base my remarks on a comprehensive study of many plants, such as Prof. Bright made so capably. Please forgive me if my viewpoint is based on a much more limited coverage (fortunately at rather close range) and confined to automotive parts production.

I am inclined to feel that discussion of fully automatic, "automated" or semiautomatic, factories can become mere exercises in semantics. I have never seen an automatic factory and don't expect I'll live long enough to see one. I have seen many outstanding examples of automatic material handling, which tied together automatic machines which not only produce the parts but inspect them as well. Such factories, however, incline to be full of people most of whom feel they are not superfluous. The continuing aim of the industry is to replace manual labor by machines, wherever production rates permit. Even though we have given this trend a new name, automation, it is neither new nor revolutionary—it is merely what we've chosen to call our present struggles along a path which leads from the village blacksmith shop and approaches, but will never quite reach, the automatic factory.

Our experience has been that parts produced on automatic lines are superior to those produced on our so-called "hand lines." I am thinking principally of our automotive frame lines. Here, we are not dealing with chunky parts such as rigid engine blocks but with flimsy, flexible members which deform considerably of their own weight until they have been completely assembled and welded together. Parts such as these present some real problems in automation. Our automatic lines produce a superior product, largely because the parts have to be better before the automatic lines can use them. When two parts don't quite assemble on a hand line, the operator merely taps one piece or the other, and they go together. The automatic line lights up like a tilted pinball machine and stumbles to a halt. This can't be tolerated, so we concentrate on better parts. In other words, the automatic machine forces us to make better parts and then assembles them to more exacting standards. This, we feel, is good for us and good for the industry.

As to the effect on manpower—as the primitive brute-force operations become mechanized, there will be less overall need for low levels of skill for a given rate of production. Today we would not hire a man to do nothing more than mechanically pull a lever all day long—we would use a motor or an air cylinder to do that job. But when we install the motor or the air cylinder, we create a need to service that piece of equipment, just as its conception required a higher degree of skill. The trend toward factory mechanization will reduce the total need for low skilled workers, and increase the need for engineers, technicians, and maintenance skills.

Ultimately, the increasing productivity of our industry, not based on automation alone but on the total industry effort, will produce more goods in less time and for less money. This will logically lead to a reduction in working hours and increased leisure hours.

Contrary to Prof. Bright's findings, our company urgently needs higher maintenance skills. We are carrying out elaborate plans to "make" some people such as these—and our feeling is that the industry would do well to concentrate on apprenticeships and training programs so that they can help themselves. We do not feel this shortage is a direct result of "automation"—it is a result of increasingly complicated equipment throughout industry and a tight labor market.

As to maintenance costs—I agree with Prof. Bright that no two plants seem to include the same ingredients in what they call "maintenance"—but it is painfully true that "debugging," "installation," "periodic lubrication and servicing" are all costs, as well as the equipment cost, the tool costs, and the obsolescence costs. To properly evaluate "automation," or for that matter any other shop improvement, we need to talk in terms of "total cost of the item produced"—taking into account the model run, the effect of engineering changes, life of the equipment, and all of the other costs, direct and indirect. I have seen automation which showed fantastic reductions in direct labor (widely quoted), and equally fantastic costs of designing, building, debugging, and scrapping at the end of the model run. These are sometimes not quoted!

I believe that each production requirement must be studied in the light of cold statistics, using the best experience and assumptions obtainable. We must keep in mind that no matter how many categories we put them into, costs are money—and money comes only from the customer. What really counts is the unit cost of the part being made for the life of the contract.

In conclusion, by the time honored examples of the sewing machine, the printing press, and the reaper, history shows that labor has nothing to fear from what we choose to call automation—it will enable us to produce more goods at prices which more people can afford. Such an expansion of our activities will improve our lot at home and make us better able to sell merchandise in foreign markets. I believe that countries which have been backward in adopting modern industrial methods have suffered a decline in world standing and in standards of living at home. Those who have been technological leaders have prospered. So let it be with us.

ORAL DISCUSSION

Reported by D. Reyna
Thompson Products, Inc.

E. F. Gibian, Thompson Products, Inc.: This paper gives us a workable measurement of the degree of automation present in an operation, but there are some other yardsticks or factors that must be considered.

We must ask ourselves, how much will the cost of the operation be decreased and how flexible will be the automated equipment. One of our divisions purchased a machine, and in four weeks it was superseded, because it lacked flexibility.

There is also the possibility that you may automate an operation or department and yet not obtain substantial savings. An example of this is our forging department. Further automation of this department will depend upon the development of die materials with a longer operating life and forging equipment more adaptable to automation.

One of the driving forces behind the advancement of automation in recent years, has been the yearly productivity raise. The only way industry can obtain the money for the annual productivity raise is by the application of automation to its operations.

HIGH-SPEED, HIGH-OUTPUT,

THIS paper deals with the problems connected with high-speed loop-scavenged 2-stroke diesel engines. The theoretical processes of scavenging are described, and methods for their calculation are discussed. It is shown that it is possible to predetermine, to a considerable extent, the most important characteristic values of scavenging and, thus, to shorten the development work.

Various arrangements for the supercharging of 2-stroke engines are treated, and the problem of turbocharging this type of engine is described.

IN this paper I shall consider only one group of high-output diesel engines, namely, high-speed, loop-scavenged 2-stroke engines. By high-output engines we mean engines which have a low weight/power ratio. Since weight is roughly proportional to cubic capacity, these engines also have high output per unit of swept volume. High-output engines are used where low weight and compactness is required and where only limited engine space is available. This means they can be used for automotive application, locomotives, and certain types of marine and semiportable installations.

Two methods to increase the power output of an internal-combustion engine are possible. The first consists of increasing the number of power strokes per unit of time, the second in increasing the output per power stroke. It is, of course, possible to increase the number of power strokes of both 4- and 2-cycle engines by raising the crankshaft speed. A substantial increase in power strokes can, however, be achieved by changing from a 4-cycle to a 2-cycle design.

The endeavor to increase the output per power stroke leads to supercharging, that is, to filling the cylinder with a charge at higher than atmospheric pressure. It is, of course, possible to supercharge a 4-cycle as well as a 2-cycle engine, but it is obvious that particularly intense utilization of the piston displacement is achieved by supercharging high-speed 2-cycle engines.

I shall confine my remarks mostly to one group of 2-stroke engines, namely to the loop-scavenged type. In these, the charging process is controlled by the pistons only, and valves are not used. The speed of these engines is not restricted by mechanical limitations and can, therefore, far exceed that of valve-controlled 2-stroke engines. Owing to the absence of cams, valves, and valve gears, the engines are simple and sturdy. Due to the simplicity of their cylinder-

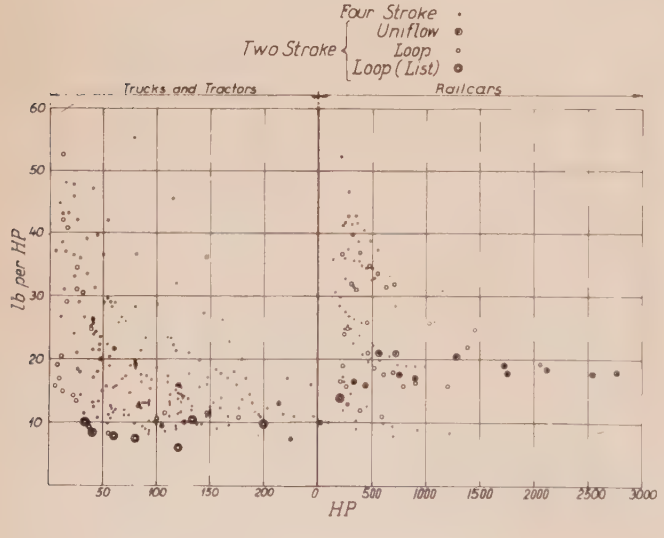


Fig. 1—Weight/power ratio

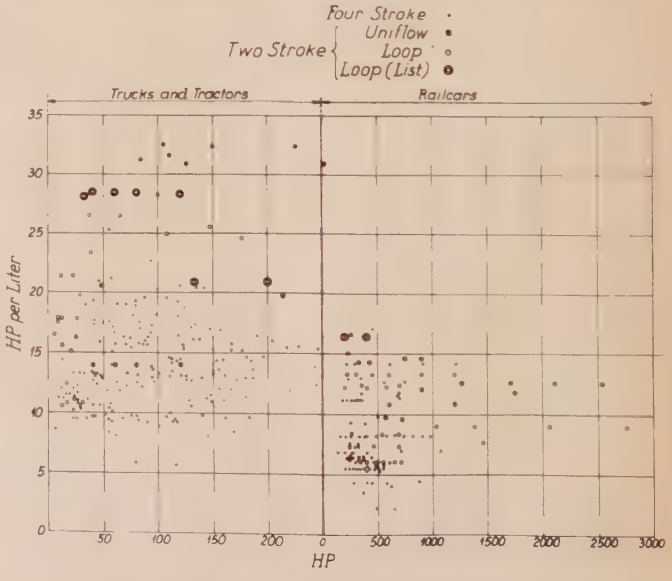


Fig. 2—Output per unit of swept volume

LOOP-SCAVENGED TWO-CYCLE DIESEL ENGINES

Prof. Dr. Hans List, Anstalt fur Verbrennungsmotoren

This paper was presented at the SAE National Diesel Engine Meeting, Chicago, Nov. 1, 1956.

head design, they have a high resistance to thermal stresses, and thus, are well suited to higher supercharge.

The values of weight/power ratio and output per unit of swept volume obtained from different engine types in some applications requiring high-output engines, are shown in Figs. 1 and 2. It can be seen that the results obtained from high-speed loop-scavenged 2-stroke engines are relatively high. This design, therefore, is suitable for high-performance engines, particularly as it combines high performance with simple and sturdy design.

This simplicity, or the small number of moving parts, is demonstrated in Fig. 3, which shows a comparison of the moving parts contained in two engines having nearly the same output. One is a 4-stroke diesel, the other a 2-stroke diesel; both give approximately 40 hp at 3000 rpm. The first engine needs four, the second two cylinders to give this output; the torque curves of both engines are approximately the

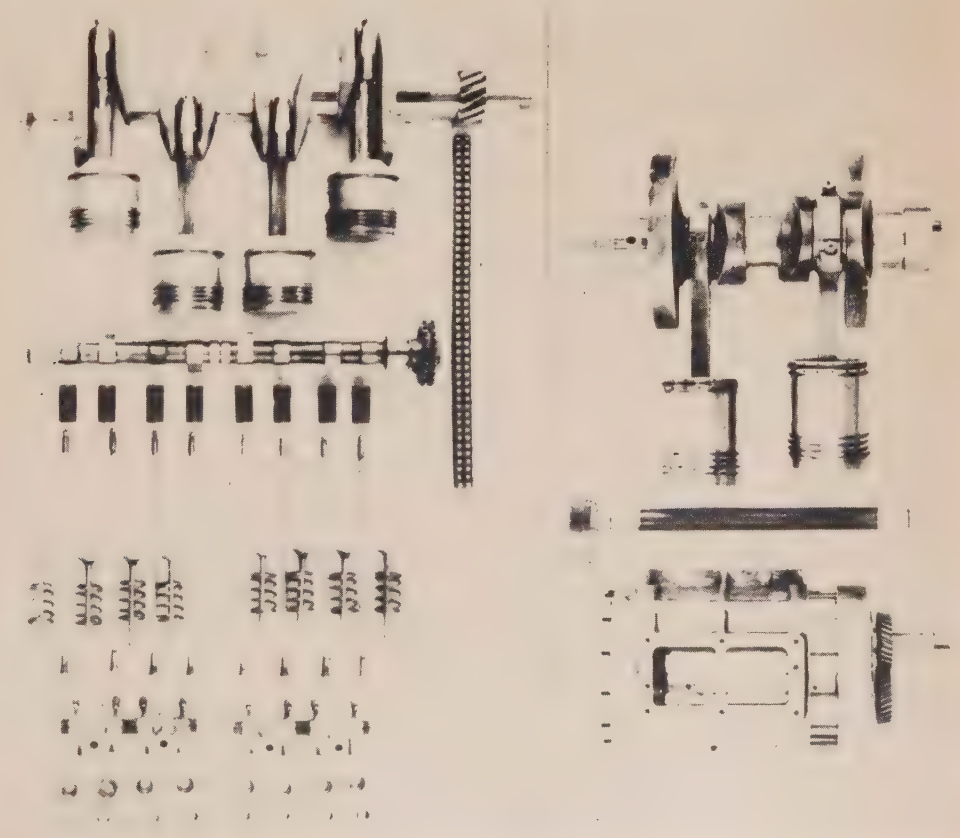


Fig. 3—Comparison of moving parts of 4- and 2-stroke engines of same output

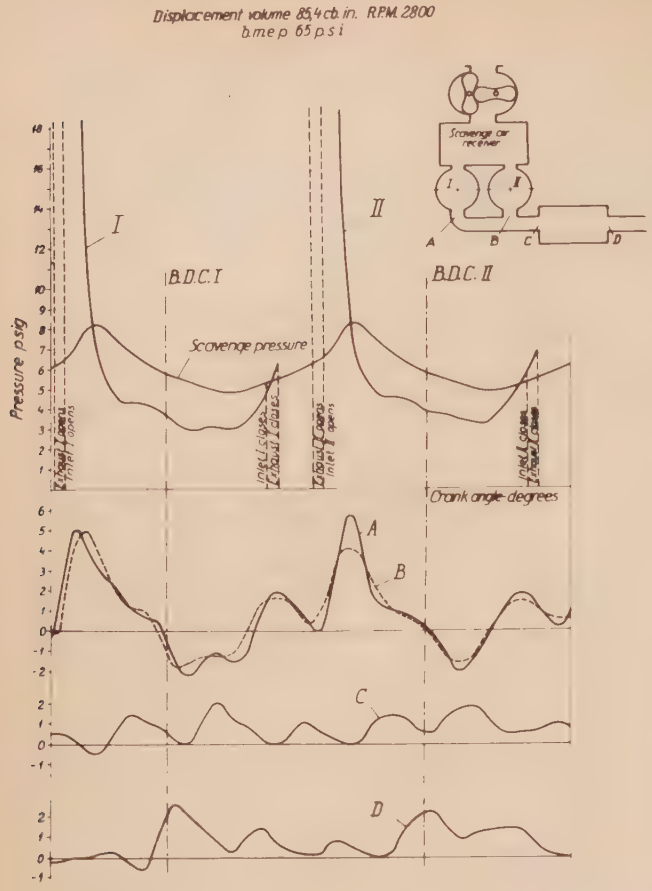


Fig. 4—Calculated exhaust pressure pulses and schematic scavenging system

same, and so are the inertia forces. In the case of the 2-cyl 2-stroke engine, this was accomplished by the use of a balancing shaft.

It can be seen from these brief statements that the high-speed loop-scavenged engine has its advantages and that it deserves to be studied to a greater extent than has been done heretofore. More development work on this type is clearly worth while.

The apparent simplicity of the engine's layout does, however, not mean that it is easy to design. Whenever a single part has to perform several different functions, it is almost always more difficult to design correctly such a part than if it had to fulfill only a single function.

A relatively large number of functions is concentrated on each of certain parts of a loop-scavenged engine so that these parts are not too easy to design. These difficulties become greater, the higher the engine speed and the higher the thermal stresses.

When the Institute for Internal Combustion Engines, Graz, Austria, was first formed with rather limited facilities and finances, it was by no means certain that satisfactory solutions could be found to all problems posed by high-speed 2-cycle engines. Under these circumstances, the decision to concern ourselves primarily with these problems required a certain amount of courage and a good deal of conviction. The same statement, of course, applies equally well to the formerly existing group headed by the late Dr. Schnürle, whose organization concerned itself exclusively with high-speed 2-stroke engines.

I can say here that, thanks to cooperative effort, most of the important problems of the high-speed 2-stroke engine have been solved so that it is now well able to compete with the 4-stroke engine. As has been proven by the latest developments in this field, the former can now claim some advantages over the latter.

I shall place the work of my institute in the foreground. We have a certain right to this due to our long experience with high-speed 2-stroke engines and due to several completed developments in this field. Furthermore, I feel that it is better to talk about problems one has solved oneself, instead conveying experiences of others.

Theoretical Basis

Among the problems inherent in loop-scavenged 2-stroke engines, the charging processes are the most important. The burned gases contained in the cylinder must be replaced by fresh air within a relatively short period of time. In order that this process may take place with the greatest possible efficiency, a number of factors must be adjusted to one another. With the increasing number of possibilities the difficulty of making the right choice increases also. Since we, in Europe, are forced to carry out development work at minimust cost and since, in general, calculation is cheaper than experimentation, it was obvious that mathematics should be employed for the investigation of the charging processes.

The charging process consists essentially of three groups of events: first, the gas-flow process, that is, the outflow of the exhaust gases from the cylinder and the transit of the scavenging air from its container through the cylinder to the exhaust; second, the mixing and displacement process in the cylinder by which the gases in the cylinder are replaced by fresh air; and finally, the heat-exchange process, by which heat is transferred from the hot gases to the cylinder walls, and inversely, the cylinder walls warm up the incoming scavenging air.

Gas-Flow Process—A great number of calculations, as well as experiments carried out to check their accuracy, have shown that it is possible to calculate flow-processes with sufficient accuracy by a step-by-step calculation. Calculation is not only applicable to the simplest case, where large containers are used ahead and behind the cylinder, but they also aid in the comprehension of the sometimes very complicated flow and oscillation processes which result when the duct system, through which the gases flow during the gas exchange, consists of a combination of tubes, orifices, and containers having variable cross-sections. The calculation can also be applied to multicylinder engines of different designs, Vee-form or in-line, and it also affords a very good insight into the gas move-

ment through the passages. To save time I must forego an exact description of the calculation, but I would like to show its elementary basis, to describe briefly the mathematical processes, and to give some results.

For the purpose of this analysis, one starts with two differential equations which show the principal character of the calculation. One equation makes it possible to calculate the change of pressure in the cylinder, P_c, taking into account the varying inlet and exhaust port areas and the piston movement. For use in the calculation these differential equations are transformed into a difference equation which has the following form: (See Appendix for nomenclature.)

$$\Delta P_c = P''_c \frac{k}{z}\Bigg[\overbrace{\frac{\tau_i W''_i}{180\, W_{mi}} \cdot \frac{P''_i}{P''_c} \cdot \left(\frac{P_{sc}}{P''_c}\right)^{\frac{k-1}{k}} (\mu_i \sigma_i)'' \Delta\alpha}^{\Delta V_i} - \overbrace{\frac{W''_e}{180\, W_{me}} \cdot \left(\frac{P''_e}{P''_c}\right)^{\frac{1}{k}} (\mu_e \sigma_e)'' \Delta\alpha}^{\Delta V_e} - \Delta z\Bigg]$$

The first term in brackets, ΔV_i, corresponds to the admission into the cylinder, the second term, ΔV_e, corresponds to the discharge from the cylinder, and the third, Δz, represents the piston movement. The change in pressure in containers, inserted in the duct system through which the scavenging gases flow, is calculated by using the same equations, but omitting the term for the piston movement, Δz.

The pressure and velocity-waves in pipes can be calculated by the following system of equations:

$$P = \bar{P} + \frac{1}{v}\left[F\left(t - \frac{x}{a}\right) - f\left(t + \frac{x}{a}\right)\right]$$

$$W = \bar{W} + \frac{g}{a}\left[F\left(t - \frac{x}{a}\right) + f\left(t + \frac{x}{a}\right)\right]$$

This system shows that the pressure at a certain point in the pipe consists of the sum of a constant pressure and of the pressures due to outward and returning pressure waves. If one knows the waves leaving from one pipe and the reflection conditions at the other, it is possible to determine the reciprocating waves and, thus, to determine the pressure at any point of the pipe.

The reflection conditions occurring at points of change, as at orifices or at changes in cross-sectional area, as well as the passage of the waves through conical parts of the pipe, can be calculated, and the necessary equations can be determined. For the purpose of calculation it is assumed that the contents of the entire duct system are at rest and that the engine suddenly begins to operate at a certain crankshaft speed. For a 2-stroke engine, for instance, one would begin with the following assumption:

The scavenging system, as shown in Fig. 4, would be filled with air at a scavenging pressure P_{sc}, the combustion would have taken place in the cylinder, and the working gases (charge) would now expand in the cylinder. The exhaust system would be filled with exhaust gases at mean temperature and, at first, at external pressure. The exhaust ports open and the outflow of the high-pressure exhaust gases into the exhaust duct begins. A pressure wave is set up, which is propagated in the exhaust pipes and which finally reaches the exhaust container. There it causes an increase in pressure and finally causes the outflow of the exhaust gases to the atmosphere through the tail pipe. The course of these waves and of their reflections and the variation of pressure in the exhaust container can be determined by calculation.

A certain pressure exists in the cylinder when the inlet port opens, and, according to the magnitude of this pressure, the exhaust gases will either backfire from the cylinder into the scavenging system, or the inflow of scavenging air into the cylinder will begin at once. If the residual gases backfire, the pressure in the cylinder decreases until it is equal to the scavenging pressure. The movement is then reversed, and the backfired gases, whose quantity can be determined by calculation, will reenter the cylinder. After all gases are pushed back into the cylinder, scavenging begins. Immediately after the opening of the inlet ports, a pressure wave and a velocity wave expand from the inlet ports into the inlet system. However, scavenging-air admission into the cylinder also influences the pressure variations within the cylinder and the pressure and velocity conditions in the exhaust ducts. By calculation of the whole charging process the variations of pressure and velocity are obtained for any point in the system.

When the charging process is completed, that is after the ports have closed, the gases in the inlet and exhaust systems are not at rest but are in movement. These movements can be followed by calculation for both systems and, thus, the state of motion at the beginning of the new gas exchange may be determined. At the second gas exchange the system will be near the actual working condition and, if the calculation is continued long enough, the latter condition will be obtained. In general, it will suffice to calculate two or three cycles.

The calculation can now be evaluated in different ways: the pressure variations, before, within, and after the cylinder and in the different containers are obtained; also, the variations of charge and discharge in the cylinder and the velocity at any time and at any point of the systems are determined. The accuracy of the calculation and its agreement with reality can be checked by indicators installed at certain points, and from the conformity of measurable values with calculated results the exactness of those values which are not measurable can be judged.

The calculation, therefore, furnishes the designer with valuable information as it shows the different processes in a kind of slow-motion picture. The insight into the processes, obtained from repeated calculations, will frequently be sufficient to let one judge correctly the results which will be produced by cer-

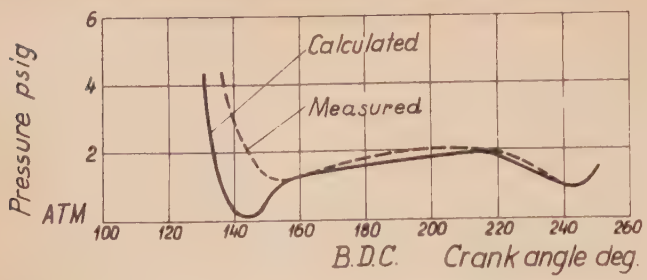


Fig. 5—Pressure in cylinder during scavenging for 2-stroke engine

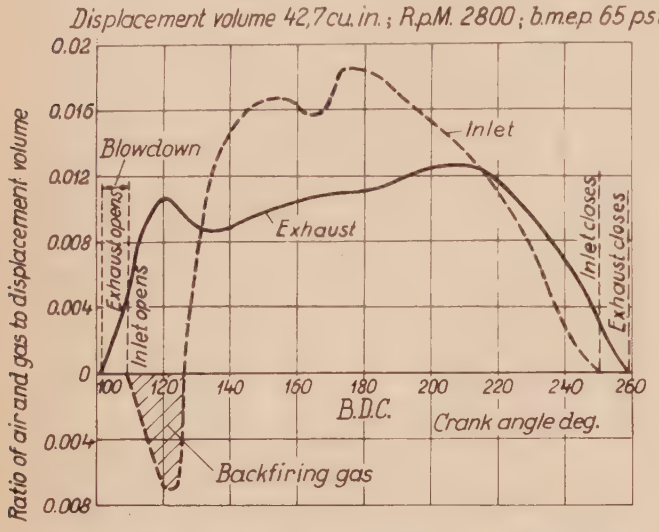


Fig. 6—Air and gas flow through inlet and exhaust ports

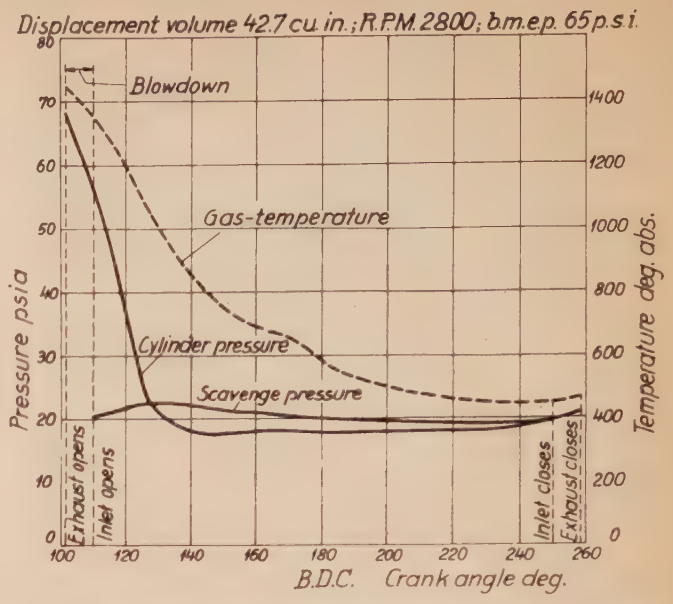


Fig. 7—Calculated pressure and gas temperature in cylinder during scavenging

tain measures, even though a specific calculation may not have been performed.

In the following figures I should like to show the results of such a calculation and in this way to demonstate its efficiency and usefulness.

Fig. 5 shows the measured and the calculated pressure variations during the scavenging process in a low-speed 2-stroke diesel engine.

Figs. 6 and 7 show the variations in the inflowing and outflowing masses and the variations of pressure and temperature in a 2-stroke engine.

Fig. 8 shows the results of a calculation for a more difficult process, namely the pressure variations in the suction pipe of a 4-stroke engine. Here also, the measured and calculated pressures are in good agreement.

Scavenging Processes—One of the most important processes in the engine is the scavenging process. In this process the scavenging air enters the cylinder and drives out the exhaust gases. It is clear that the effects of this process cannot be calculated and that the determination of the scavenging efficiency must be left to experiment.

Corresponding tests have been made with the models as well as with running engines. Here it was found that results of model tests, if correctly applied to the engine, lead to quite useful conclusions as to the processes in the working cylinder.

We have carried out many scavenging tests on a model cylinder which was filled with carbon dioxide and then was scavenged with air. Through these tests we obtained invaluable knowledge, not only of the efficacy of different scavenging systems, but also of the effect of the different variables upon these systems. The results have been plotted as fractions of the cylinder contents, with the quantity of the entering air as abscissa and with the air remaining in the cylinder as ordinate.

These scavenging curves permit judgment of a scavenging system. "Scavenging system" is understood to mean the arrangement of the inlet and exhaust ports within the cylinder. I shall show you only an extract from many thousands of tests, a diagram which is well-known and which has been used by many research engineers. This diagram shows the scavenging curves for three entirely different scavenging systems (Fig. 9).

The "cross-scavenging system" employs inlet and exhaust ports placed in opposite sides of the cylinder wall, the "loop-scavenging system" inlet and exhaust ports in the same side of the cylinder wall, and "uniflow-scavenging system," inlet and exhaust ports at opposite ends of the cylinder. You can see that uniflow scavenging gives by far the best scavenging, that loop-scavenging is good, and that, in general, cross-scavenging is the worst.

The quantitative application of these results to the actual engine has been tested by exact measurement. Since the gas in the cylinder is stratified after scavenging, small gas samples lead to error. We have, therefore, developed a method for extracting larger samples from the cylinder and, by the arrangement shown in

Fig. 10, have even emptied the whole cylinder contents into a rubber balloon. By this method the purity of the charge and its quantity can be accurately determined. This method is too complex for the normal examination of an engine, but it gave us the basis for improving the procedure of calculating 2-stroke engines.

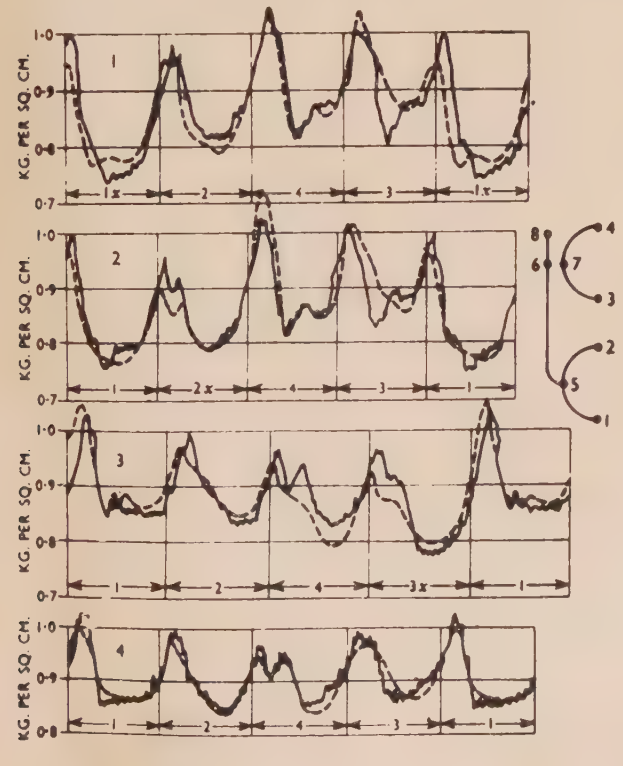
For the development of actual scavenging systems it is sufficient to extract samples of residual gases from the blow-down part of the exhaust either by means of a controlled valve or by a small relief valve. From these samples and from the measured fuel quantity, it is possible to determine the air quantity in the cylinder which is available for combustion. In this fashion the "delivery ratio" is obtained.

Thus, proceeding from the so-called "dilution scavenging," the introduction of scavenging curves into the calculations is possible. In dilution scavenging, a theoretical concept, the incoming air mixes at once and thoroughly with the residual gases, and an equal volume of this mixture is then pushed out of the exhaust. The course of scavenge can be calculated and is represented by the following equation:

$$\eta_{sc} = 1 - e^{-L_c}$$

In engines with inlet and exhaust at one end of the cylinder, the scavenging curves are, in general, near those of dilution scavenging. With good loop-scavenging they are above and with cross-scavenging they are generally below. In many cases, the course of the scavenging can be accurately expressed by calculations following the assumption of dilution scavenging. In order to introduce into the calculations scavenging curves deviating from dilution scavenging, a coefficient a is used and multiplied by the air consumption L. If a is greater than 1, the scavenging is better than dilution scavenging, but if a is less than 1, the scavenging curve is below that for dilution scavenging.

So far as their influences on the course of scaveng-



— — — — Calculated. ———— Measured.
x = admission stroke.

Fig. 8—Calculated and measured pressures in admission pipe

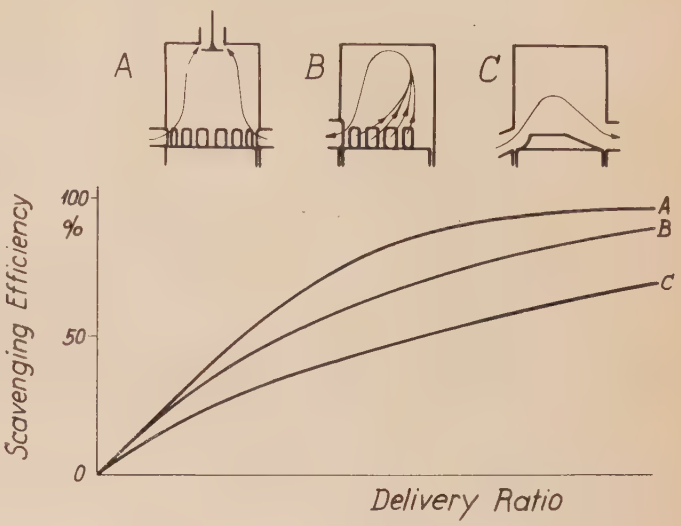
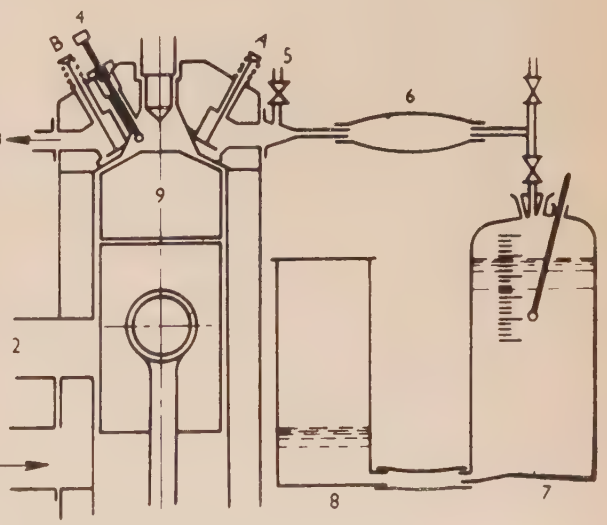


Fig. 9—Scavenging systems



1. Scavenging air.
2. Exhaust pipe.
3. Gas taken here for analysis.
4. Thermocouple.
5. Connexion for clearing.
6. Rubber bag.
7. Measuring vessel.
8. Levelling vessel.
9. Loose piston head.

Fig. 10—Apparatus for testing total charge and scavenging efficiency

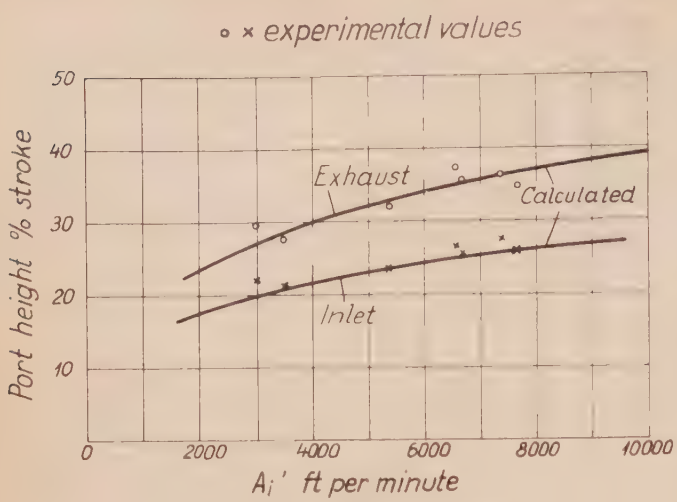


Fig. 11—Optimum port heights

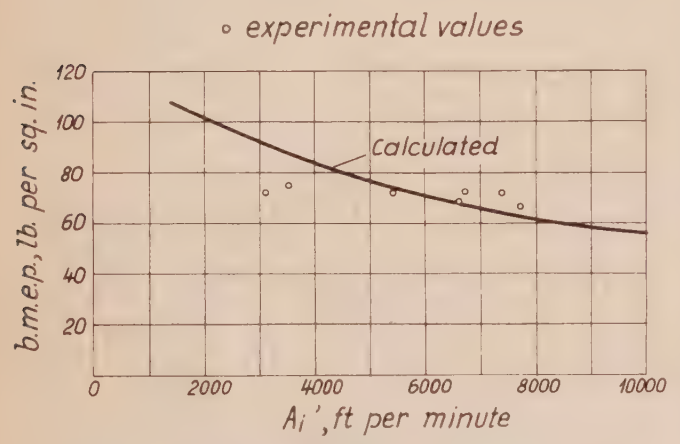


Fig. 12—Optimum bmep values

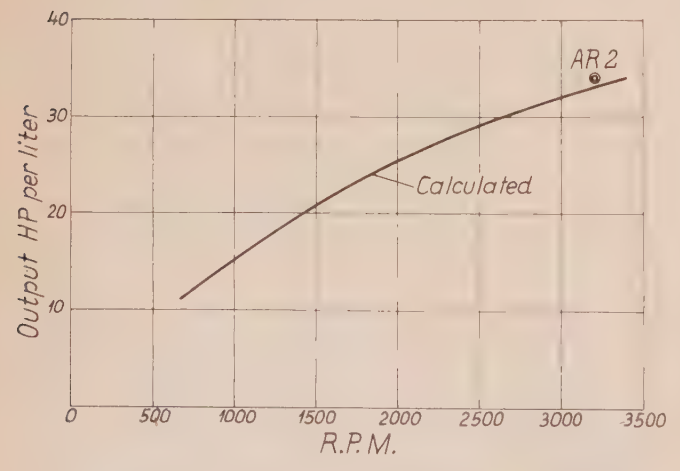


Fig. 13—Maximum output of 2-stroke engines with 3.35-in. bore

ing are concerned, it can be said that the arrangement of the ports is of greatest importance while the layout of the cylinder head is of minor consequence. This also applies to the shape of the piston crown in loop-scavenged engines. With increasing stroke/bore ratio, the delivery ratio becomes somewhat worse, but this effect is slight in loop-scavenging, and it also gets worse with increasing scavenging pressure owing to the greater turbulence of flow.

Today, fundamental examinations of scavenging arrangements have lost their interest, as the general results are already known. Furthermore, as the pertinent patents have expired, it is generally possible to design good scavenging systems. Scavenging arrangements which have the exhaust ports in the center and the inlet ports on each side so that the intake air impinges on the cylinder wall opposite the exhaust ports are, according to our experience, the most successful for high-speed 2-stroke engines in general. This design combines best scavenging efficiency with relatively ample port areas. We have used this scavenging arrangement, which was first introduced by Dr. Schnürle, for all our engine designs. With good scavenging layouts a scavenging efficiency of 90% is obtained (at an air-consumption volume ratio usually of 1.4), that is, the charge in the cylinder contains approximately 10% residual gases. By careful design of the ports following extensive tests, even the scavenging efficiency of loop-scavenged engines can be improved further.

Approximation Method for the Calculation of Port Dimensions—It is a disadvantage of the step-by-step calculation that the pipe system, the manifolds; and the timing of the engine must first be assumed. This method, therefore, does not show an immediate way for obtaining most favorable values for an actual scavenging system, but serves for comparison with the chosen system.

A start for the layout of ports is offered by a considerably simplified calculation method. Here, one proceeds from a simplified system with large containers before and after the cylinder. An orifice, multiplied by a discharge coefficient which includes the other resistances, is substituted for the controlled ports.

To determine the most favorable port lengths, one proceeds from an approximate expression for the brake mean effective pressure:

$$\text{bmep} = K(1 - \sigma_i - 0.00314\sqrt{A_e} + 7.3 \cdot 10^{-6} A_e)(1 - e^{-a'L}) - \frac{8 \cdot A_i^2 L^3}{\sigma_i^3 \cdot \eta_b \cdot 10^9} - P_{fr}$$

By partial differentiation with relation to L and σ_i the conditions for P_e maximum are obtained.

A represents the high-speed figure of the engine. The greater A is, the more difficult are the gas-exchange processes. Basically, the product $D \times n$ is contained in this A, as are some values of ratios, especially that of the width of ports in relation to the cir-

cumference of the cylinder and also the coefficient of discharge.

Results show that the most favorable inlet- and exhaust-port lengths of our engines are shown in Fig. 11. These were obtained experimentally, starting from the calculated port until the optimum values were obtained. Variations of port length were investigated in both directions. As you can see, these values agree well with our calculations. This method, therefore, offers a useful starting point for the determination of port length, but assumes that certain coefficients are known, especially the discharge coefficients of the scavenging system. In Fig. 12 the bmep is plotted on a base of the high-speed figure. As you see, bmep gets smaller with increasing high-speed figures, as an increasing part of the stroke is lost to the working process owing to the ports, and an increasing part of the output must be used to drive the scavenging pump.

The characteristic figure for the output of an internal-combustion engine is the output per liter of swept volume (specific output). If medium values for the characteristics of a good scavenging system, such as the Schnürle, are substituted in the high-speed figure A, the bmep may be given as a function of $D \times n$. The output per liter is proportional to bmep $\times n$ so that as the output per liter is higher, the cylinder diameter is smaller. However, diesel engines with very small cylinders start badly. The smallest bore size which produces both good fuel consumption and good starting, is in the neighborhood of 3⅜ in. This statement does not apply to some special engines like those used in model airplanes. The curve of output per liter for a cylinder diameter of 3.35 in. (85 mm) is given in Fig. 13 and shows a value of 34, or nearly 0.56 hp per cu in. of piston displacement. It is of interest that the engines we developed for Alfa Romeo (3.35-in. bore) give, in the latest stage of development, an output of 34 hp per liter which corresponds well with the calculated results. This specific output is relatively high but nevertheless can be obtained with a loop-scavenged 2-stroke engine of comparatively simple design.

Mechanical and Thermal Problems

The major mechanical problem with a loop-scavenged 2-stroke engine is that of obtaining a sufficient, but not excessive, oil supply to the cylinder walls and obtaining reasonable lubricating oil consumption together with the smallest possible cylinder wear. Conditions here are considerably more unfavorable than in a 4-stroke engine. When the piston is at top dead-center (Fig. 14), only a comparatively narrow sealing belt prevents the leakage of oil from the crankcase into the exhaust ports. Of course, high-performance engines must have a well lubricated mechanism, and in many cases oil is also used for spray-cooling the underside of the piston crown. The crankcase is thus filled with oil-saturated air. Only

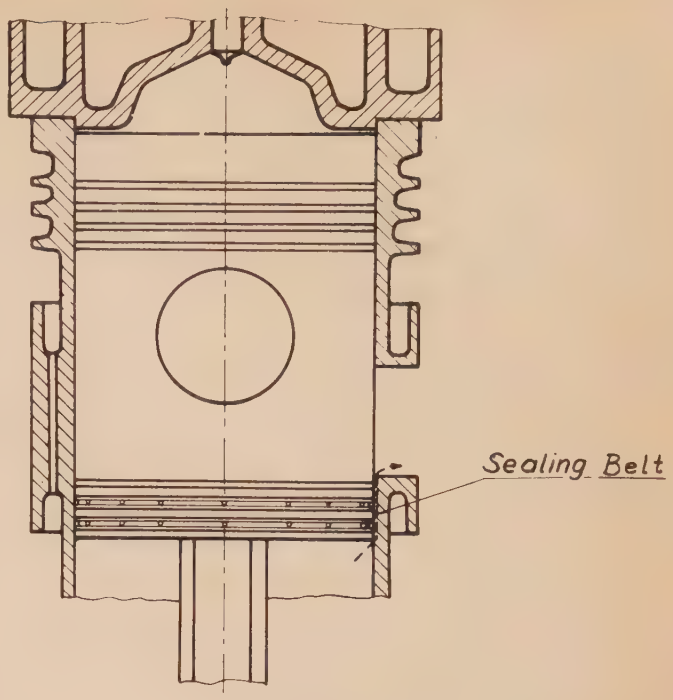


Fig. 14—Leakage path

by very accurate design of the sealing areas at the bottom of the piston can the oil distribution be kept within reasonable limits.

Variables affecting the performance of the sealing areas are the finish of the piston surface, the clearances of the piston skirt, the finish of the oil-scraper rings, and the finish of the liners. Tests on various kinds of oil-scraper rings, which extended over a long period of time, with variations in such things as ring width, oil drains, and ring tension, as well as varations in piston-skirt layout, were necessary before practical solutions were found. It is now possible to control the leakage of oil through the sealing areas.

This control of oil leakage was imperative for the practical application of high-speed 2-stroke engines, especially for speeds over 2000 rpm. The oil consumption of loop-scavenged 2-stroke engines is not only a question of economy but also of reliability in operation. In this respect, there is a difference between 4- and 2-stroke engines. The exhaust gases of a 2-stroke engine, even at full load, contain considerable amounts of oxygen from the scavenging air which passes directly into the exhaust ports, and the danger exists that quantities of lubricating oil may ignite in the exhaust ducts. This has happened to 2-stroke engines with excessive lubricating oil consumption. It usually does not harm the engine but certainly is unpleasant.

When we began our development on high-speed 2-stroke engines such burning exhaust pipes sometimes caused us difficulties. But now that we have mastered the problem of oil consumption, this belongs

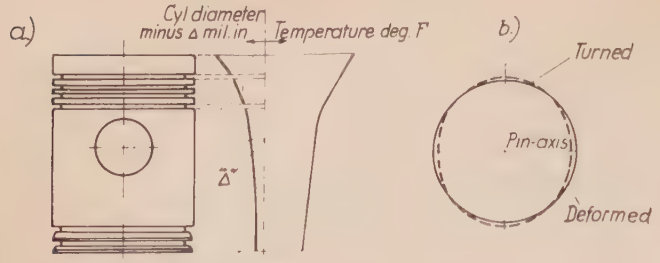


Fig. 15—Deformation of piston

to the past. I have not heard of such a case with our engines for a long time. With small- and medium-sized engines there is, in general, no separate oil supply to the cylinder, since this would complicate the engine and make it more costly. The lubricating oil leaking through the sealing area at the bottom of the piston must, thus, be so adjusted that it provides just enough oil for the lubrication of the cylinder but so that it prevents undesirable oil leakage into the exhaust system.

We found that the danger of a burning exhaust began at an oil-leakage rate of 3 to 4 gal per hp-hr. A rate of 0.5 to 1 gal per hp-hr is sufficient to form an oil film. Thus, there exists a sufficiently wide difference between necessary and objectionable rates of oil supply. In general, we design our engines with an oil-leakage rate of 0.8 to 1.5 gal per hp-hr. This oil consumption is only a trifle higher than that of a high-speed 4-stroke engine, and it ensures effective lubrication of the cylinder without any disagreeable white oil smoke in the exhaust.

In larger engines, the oil leakage through the sealing areas is made less than that necessary for the lubrication of the cylinder, and the required additional amount of oil is supplied separately. Such an arrangement has the advantage that the oil supply can be made dependent on the load so that at smaller loads less oil burns or vaporizes from the oil film and less oil needs to be supplied.

It is important that the oil be evenly distributed on the cylinder surface and that an even lubrication of the whole piston travel be effected. A special problem is the lubrication of the upper compression ring. This ring works with extremely high surface pressure and, owing to the efficiency of the scraper rings situated below, it receives the smallest amount of oil. The part of the cylinder over which the top piston rings travel is subject to the greatest wear. The oil quantity reaching the upper piston ring can be influenced by the tension of the compression rings. In order to prevent ring and piston scuffing, uniform contact between rings and cylinder wall must be ensured. Exact ring shapes, best obtained by form-turning, are necessary.

Special attention has to be paid to the exact outside shape of the piston so that the side pressure, to which the piston is subjected, is distributed as evenly as possible over the sliding surface. A combination of crowned and oval piston-skirt shape is used. In this way, the deformation of the skirt by thermal and mechanical forces is taken into account. The form of the piston skirt is considerably influenced, in longitudinal direction (Fig. 15), by the temperature gradient. The oval contour of piston circumference is changed by mechanical deformation. In short, the shape has to be such that, after deformation, a close and even contact of the piston surface with the cylinder is produced. The more even this contact, the better the operating conditions of the piston.

The determination of the shape of the piston, therefore, requires specific experience, and tests have to be made in each individual case. We have obtained satisfactory results with simple aluminum pistons. These were, however, cast of an alloy specially developed in Europe for the purpose and which, I believe, is similar to A-142.

Special attention must be paid to the cylinder wall in way of the ports. The wall is interrupted by orifices which, in high-speed engines, cover a great part of the circumference, so that only comparatively small lands remain as supporting areas.

When cylinder and piston have normal proportions, the piston pin and, thus, the point of application of the side pressure, coincides with the ports at the time when the side pressure is greatest. (Fig. 16.) The cylindrical surface area of the liner, in combination with the deformed piston, results in considerable bearing pressures which are imposed on the lands between the ports. This may lead to scuffing at these points. To prevent the possibility of scuffing, the port area is

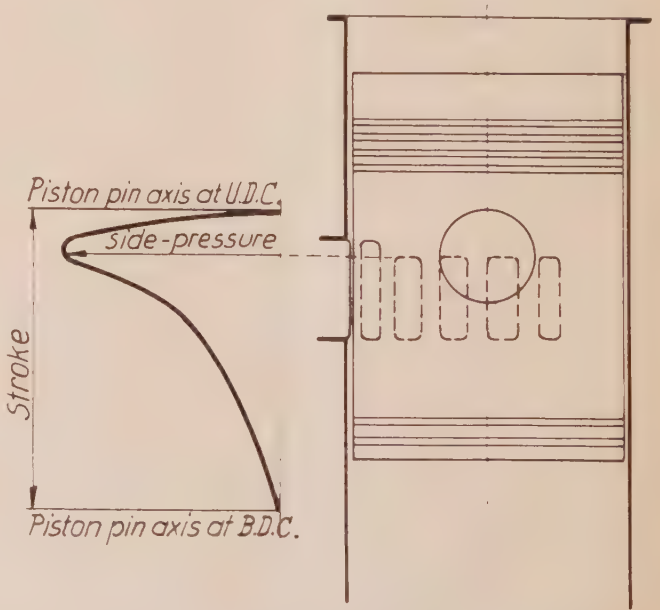


Fig. 16—Course of side pressure relative to cylinder axis

very slightly barrel-shaped. In this way these areas are freed of pressure or at least relieved, and the side pressure is distributed over the uninterrupted part of the cylinder liner with the result that maximum bearing pressures are substantially reduced.

Since the most effective sealing occurs in areas which are exposed only to slight wear, the oil consumption of a loop-scavenged 2-stroke engine does not, in general, increase with continued operation. Usually the replacement of a cylinder liner is necessitated, not by high oil consumption, but by wear in its upper part. If this is the case, the compression rings seal insufficiently and cause hard starting. A specific rate of cylinder wear is the result of a particular condition of cylinder-liner lubrication. Since purely hydrodynamic lubrication of the cylinder running surface is not possible, wear also depends on piston and cylinder material and on the quality of their finish.

It is now an established fact that the lubrication condition in the cylinder of a loop-scavenged 2-stroke engine, as obtained by the means described, results in reasonable wear figures and that the 2-stroke is no longer at a considerable disadvantage compared with a 4-stroke engine.

In Europe, high-speed loop-scavenged 2-stroke engines were first installed by the firm of Graf & Stift in their buses and heavy trucks. This firm has limited production but, up to now, has the most extensive experience in Europe in the application of this type of engine to automotive purpose. Graf & Stift report cylinder wear as shown in Fig. 17. These values were obtained under the severe operating conditions encountered in the mountainous areas of Austria and in short-distance runs. The firm of Hanomag who use in their tractors 2-stroke diesel engines rated 12–14 hp at 2200 rpm have not had any complaints about excessive cylinder wear in three years of operation. Approximately 25,000 cylinders are in use in these tractors.

The experience with the test vehicles of the Ford Works, Cologne, Germany, in which engines of our design are installed, is also favorable. After 32,000 miles a cylinder wear of 0.05 mm (0.00197 in.) was measured.

We have tried to obtain comparative material on 4-stroke engines. Steyr-Daimler-Puch, the biggest firm in Austria producing light vans, gave us data as shown in Fig. 17.

The Swiss Post Office authorities found, after extensive tests, that their 4-stroke engines have a cylinder wear from 0.1 to 0.3 mm (0.0039 to 0.0118 in.) after 32,000 miles of service.

It follows from these results, which I have intentionally limited to such material as could be obtained from nearby sources, that the cylinder wear of high-speed, loop-scavenged 2-stroke engines compares favorably with that of 4-stroke engines.

So far it has not been necessary to introduce special means to improve wear conditions, such as chromium-plated or -hardened cylinder liners. This does not, of course, mean that such measures could not bring advantages since they would, no doubt, further increase the life of the cylinders.

The mechanism of a 2-stroke engine presents no special problems and is subject to the same forces as that of a 4-stroke engine. The forces which, in general, always act in the same direction on the piston pin, produce an unfavorable load condition on the small end bearing. This condition can, however, be rendered harmless by a special groove design.

Scavenging Blowers for 2-stroke Engines

The design of a 2-stroke engine is basically influenced by the kind of scavenging blower used. The engine is simplest and cheapest if the piston is used to displace the scavenge air. "Crankcase-scavenging," however, has great disadvantages and is used less and less and is only applied to small, low-cost engines. Its disadvantages are: high oil consumption and, in spite of this, a type of bearing lubrication not sufficient to cope with the large forces acting on the running gear of a diesel engine, and, in addition, the amount of air so conveyed is limited and is not enough for good scavenging. High-output 2-stroke engines must, therefore, be equipped with a scavenging pump.

Piston-type blowers can be considered only for low-speed and single- or 2-cyl engines. In all other cases, either Roots- or centrifugal blowers are used. We use Roots-blowers for engines of small and medium output. We have found that the Roots-blower, due to its steep characteristic, is only slightly sensitive to such changes in flow resistance as can be caused by de-

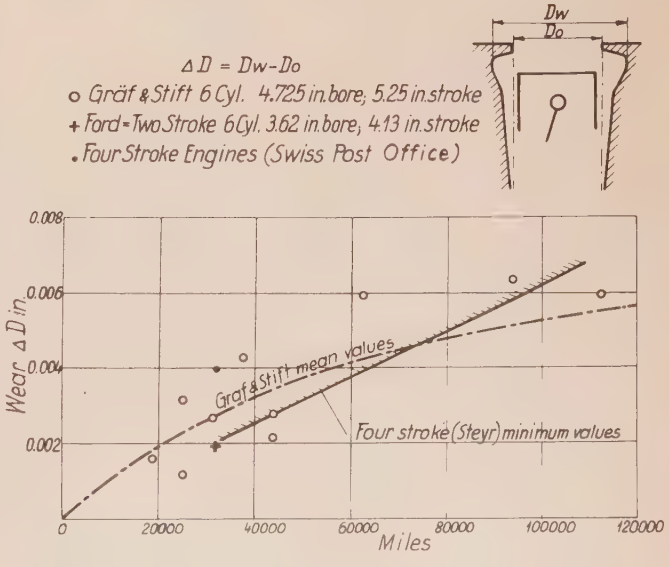


Fig. 17—Comparison of cylinder wear for three engines

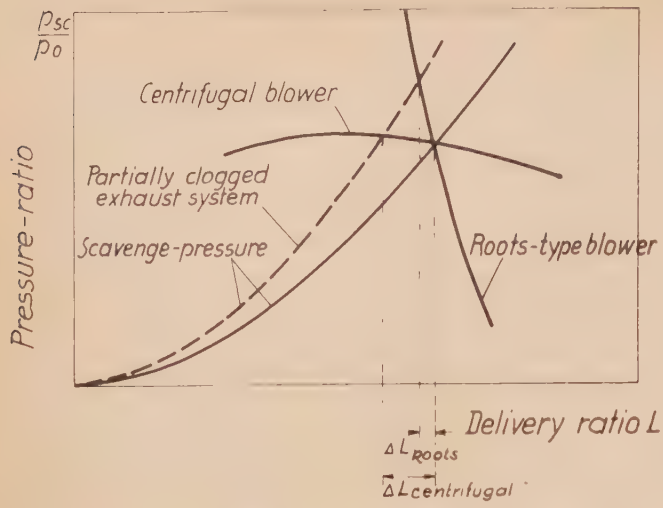


Fig. 18—Interaction of 2-stroke engine with centrifugal blower and Roots-blower

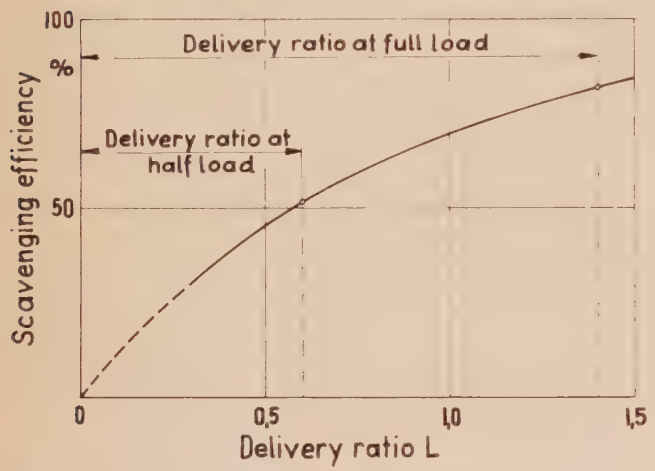


Fig. 19—Scavenge-air quantity at full and part load at equal excess air factor of combustion

posits in the exhaust ducts. Fig. 18 shows a comparison with a centrifugal blower which has a flat characteristic. An increase of flow resistance has a much greater effect on the scavenging-air output of a centrifugal blower than on that of a Roots-blower. If deposits accumulate in the exhaust system, an engine equipped with a centrifugal blower will, therefore, acquire a smoky exhaust earlier than if it had a Roots-blower.

For this reason it seems to us that whenever service maintenance is not too regular or the exhaust system has to be kept small because of space limitations the Roots-blower is to be preferred. Furthermore, flexibility as well as a torque curve, desirable for automotive applications, can be obtained more easily with a Roots-blower.

If engines operate over long periods at part load, like vehicles in city traffic, it is desirable to reduce the part-load fuel consumption. This possibility exists in 2-stroke engines by decreasing the scavenge-air quantity at low load. The shape of the scavenging curve (Fig. 19) shows that, at part load, a considerable reduction in the quantity of scavenge air is possible. Such a reduction could, for instance, be accomplished by a change-gear between scavenge pump and engine, actuated according to the engine load. However, such change-gears are too expensive for automotive engines.

A simple method of controlling the air output of a Roots-blower consists in a throttle-actuated bypass between blower inlet and outlet (Fig. 20). So far as the airflow through the engine is concerned, the latter can be likened to an orifice with variable area.

If the bypass has the same effective section as this orifice, the airstream will divide into two halves, and only one half will pass through the engine. The scavenge pressure, at same delivery ratio, will then decrease to ¼ with approximately a proportionate reduction in blower power requirement. Thus, ¾ of

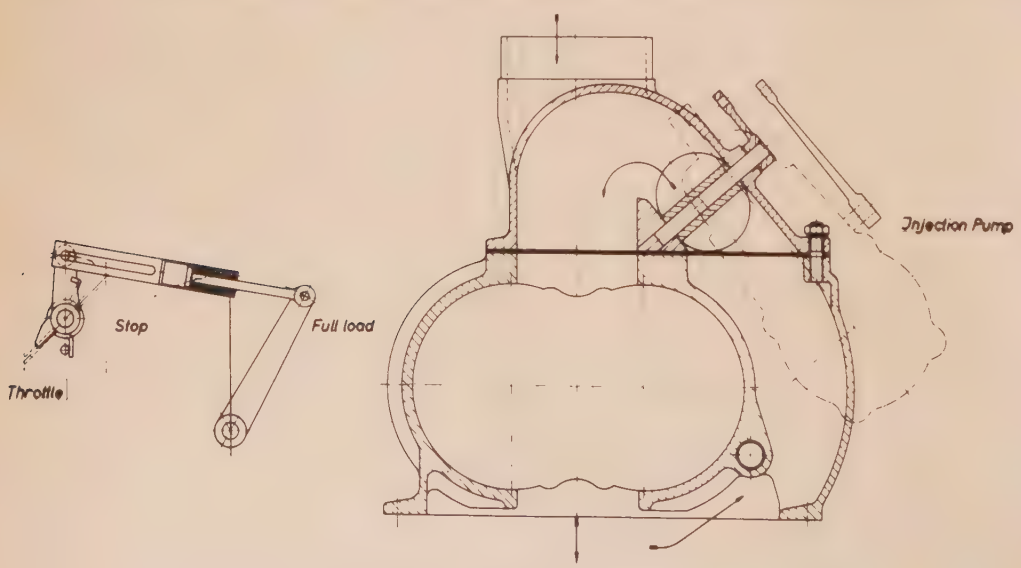


Fig. 20—Installation of bypass valve in scavenging system to adapt delivery ratio to load

the blower power input is saved, producing a substantial reduction in fuel consumption at part load. If a change-gear were used, the savings would be ⅞ of the blower input, since, in that case, only half of the scavenge air would have to be delivered against ¼ of the pressure. The difference in these savings, ⅞ for the change-gear against ¾ for the bypass, is insignificant and does not justify the considerable additional cost of a change-gear. In the actual design the bypass valve is controlled by the rack of the fuel pump. The most favorable combination is determined by tests. The Graf & Stift bus engines, used in city operation, are equipped with such a device, and their fuel consumption is reduced by approximately 17%.

Control of the air delivery of centrifugal blowers can be effected by throttling the air on the intake side. In this case, if the air quantity is reduced by one half, the saving will be one half of the blower input and is thus, smaller than the 75% saved in the case of the Roots-blower.

Description of Existing High-Speed 2-Stroke Engines

For reasons stated earlier I shall restrict my report to engines of our own design and shall describe them in the order in which they were developed.

Graf & Stift Engine—The engine is Vee-form, and 4- and 6-cyl engines have been designed. The main dimensions are: 4.725-in. bore by 5.52-in. stroke, giving an output of 30 to 35 hp per cylinder at 2000 rpm. A definite limit in output has, however, not yet been reached. Fig. 21 shows longitudinal and transverse sections, and Fig. 22 shows the consumption curves. In this design a plate covers the top of the Vee, forming the scavenge-air container. The auxiliary machinery, namely blower, injection pump, generator and brake compressor are mounted on this plate. Blower and injection pump are driven by gearing located at the fly-wheel end of the engine.

I shall discuss only the special parts of a 2-stroke engine, particularly the cylinders. The type which we have chosen for all our designs has a double wall in the port area. This design provides adequate duct length for the inlet ports, and thus, gives good control of the airflow. It also permits surrounding the exhaust ducts with water and, particularly, allows water-cooling the lands between the ports.

The water jacket below the ports is connected with that above the ports through passages in his double wall. We consider the water jacket below the ports to be essential for the success of a loop-scavenged 2-stroke engine. The piston is below the ported section for a considerable time, so that cooling of this part of the liner is important for the transfer of heat from piston to coolant. Since the heat entering the piston is primarily transferred from the piston to the liner, the cooling of the latter's lower part is important to effective piston cooling. Tests show that approximately 20% of the transmitted heat passes to the cooling water through the lower part of the cylinder liner.

The limit to the output of a 2-stroke engine is, in general, defined by the thermal load on the piston. This limit is increased considerably by cooling the piston and the lower liner. If a simple piston design is desired, the heat transfer from the gases to the piston must be kept as low as possible. We, therefore, always use a flat piston, that is, a piston crown having minimum surface. The amount of heat to be transferred also depends, of course, on the combustion system used.

Some time ago we investigated the heat rejection of certain 4-stroke engines having different combustion systems. The tests were condutced under identical operating conditions. Results of these tests are shown in Fig. 23.

The heat-rejection rate of different direct-injection combustion systems also differs substantially. Prof. Pischinger in Graz established experimentally that the heat-transfer rate is greatly affected by the

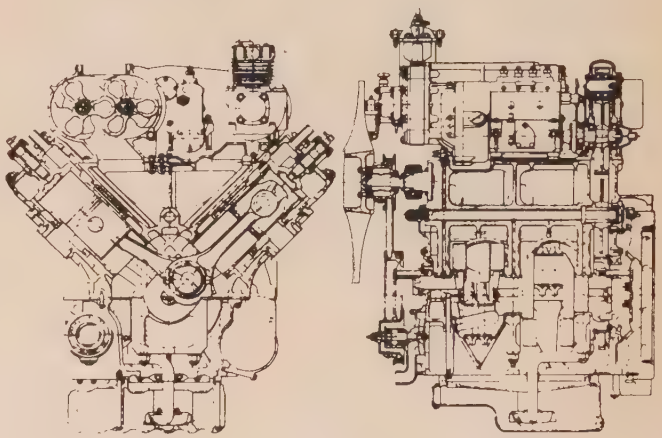


Fig. 21—Longitudinal and transverse section of 4-cyl, 2-stroke Graf & Stift engine

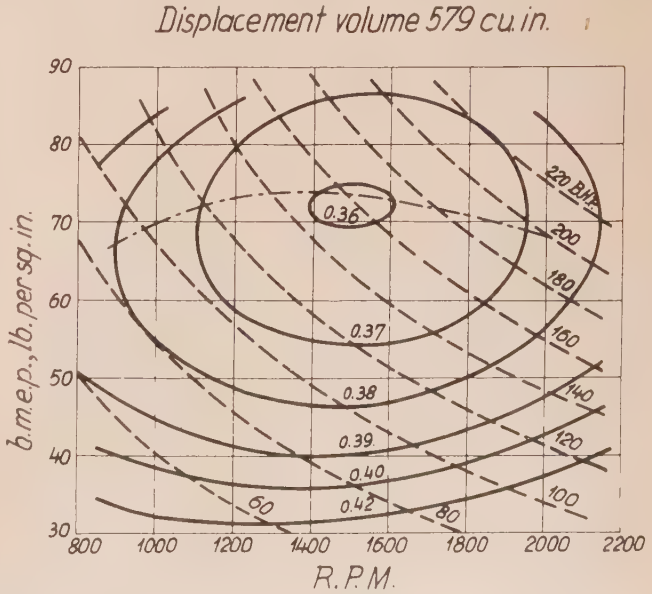


Fig. 22—Characteristics of 6-cyl 2-stroke Graf & Stift engine

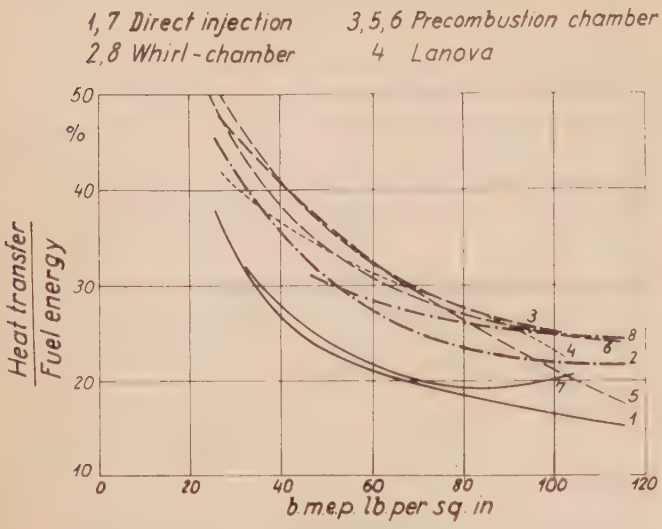


Fig. 23—Comparison of characteristics of heat transfer to cylinder wall during combustion of 4-stroke engines

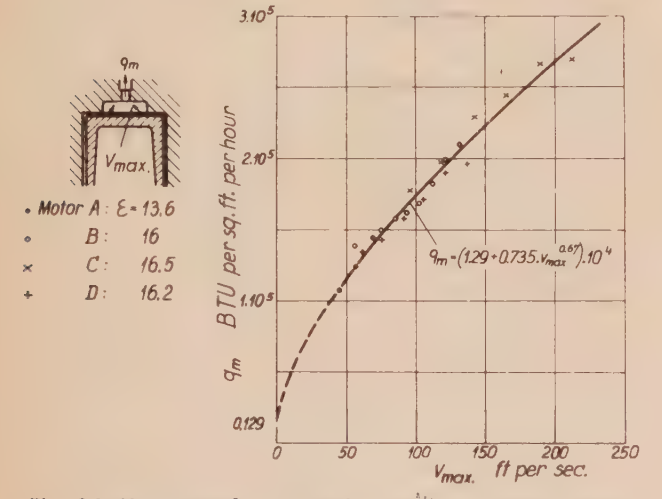


Fig. 24—Heat transfer to nozzle relative to displacement velocity

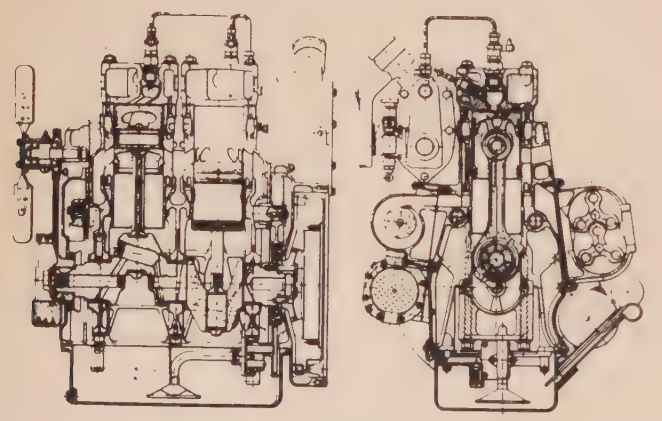


Fig. 25—Transverse and longitudinal section of 2-cyl 2-stroke engine

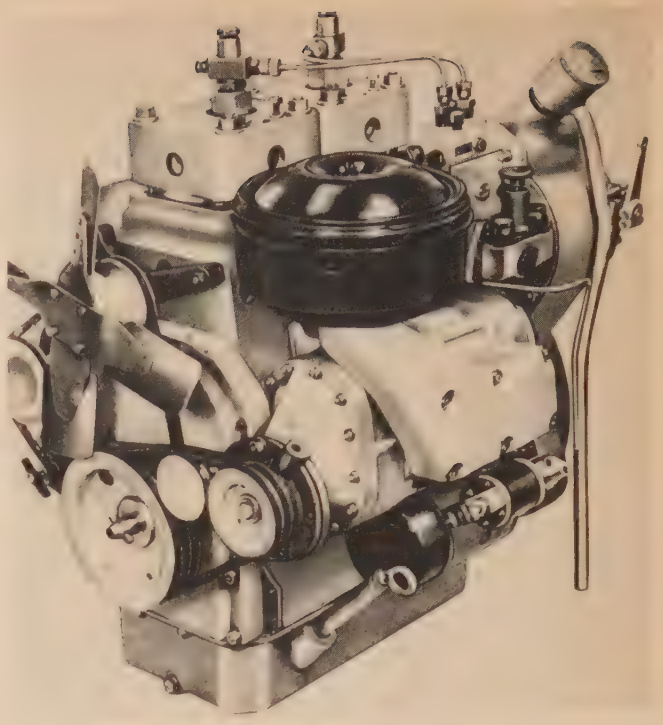
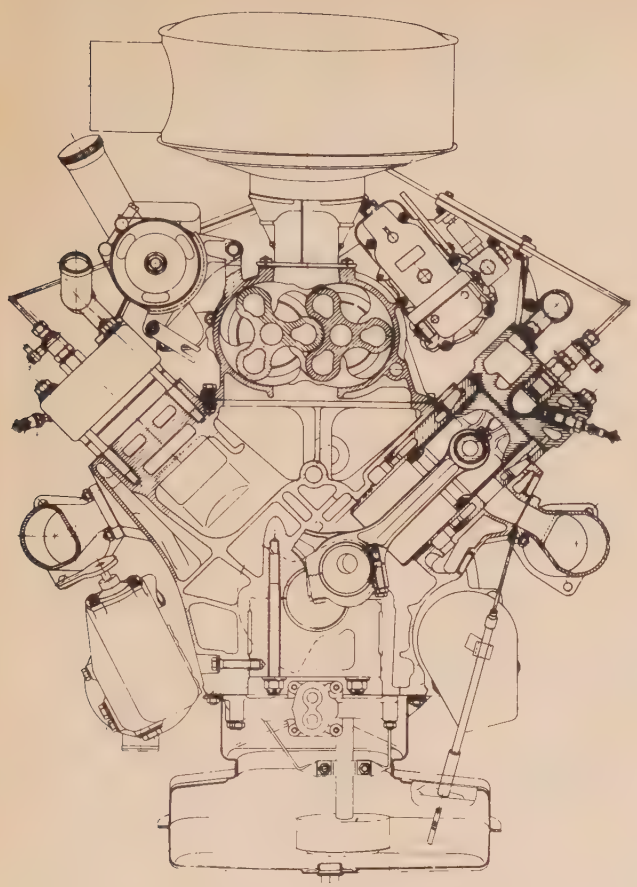


Fig. 26—JW 2-cyl 2-stroke engine

squish velocity: (V_{max}), in Fig. 24. In these experiments the rate of heat transferred to the nozzle was measured, but the findings apply equally well to the piston. As far as heat-transfer rate is concerned, flat pistons are, therefore, better than those of more involved shape.

In developing the design of the Graf & Stift engine, we first used a compact combustion chamber but finally a flat and less compact design was chosen. It resulted in lower thermal load on the piston and in better efficiency. It was also possible to increase substantially the engine output.

Engines of Ford, Cologne, Germany, Jenbacher Works, Jenbach, Austria, and Turner Manufacturing Co., England—This group of engines has a cylinder bore of 3.62 in. and a stroke of 4.13 in. At 2800 to 3000 rpm they have an output of 20 hp per cylinder. The 2- and 3-cyl engines are in-line, the 4- and 6-cyl engines are in Vee-form. The 1- to 4-cyl engines were designed by my organization, the 6-cyl engines by Amok, Vaduz. These engines are primarily intended for automotive purposes. It is planned to run them at still higher operating speeds. Figs. 25 and 26 show the 2-cylinder engine, Fig. 27 sections of the 6-cylinder engine. To control combustion noise and to reduce fuel-quality requirements, we designed the engine with a swirl chamber. We knew, of course, that heat transfer to the piston would be considerably greater than with direct injection. But since the danger of excessive thermal stresses decreases with decreasing cylinder diameter and since this cylinder is small, we found the use of a swirl chamber permissible. Despite the relatively high engine speed, the control of heat transfer to the piston proved effective,



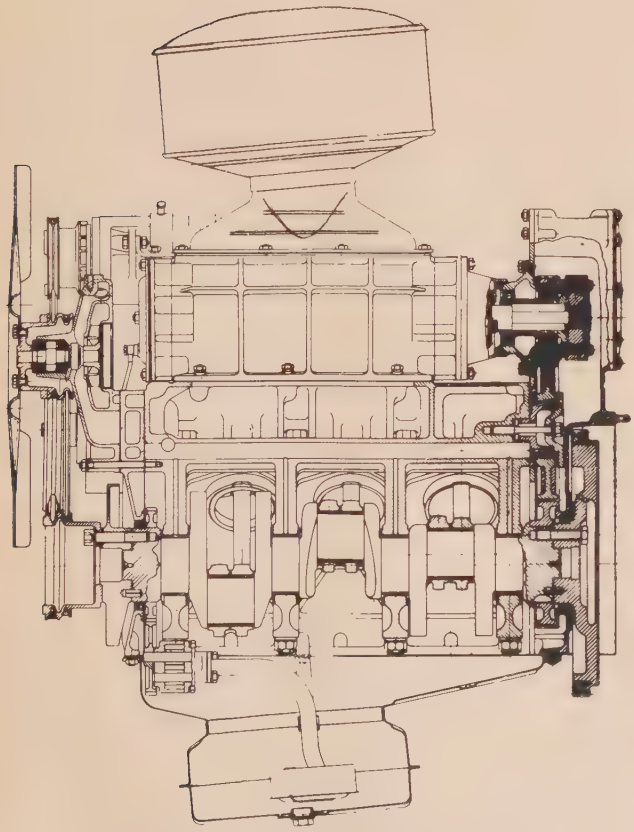


Fig. 27—Ford 6-cyl 2-stroke engine

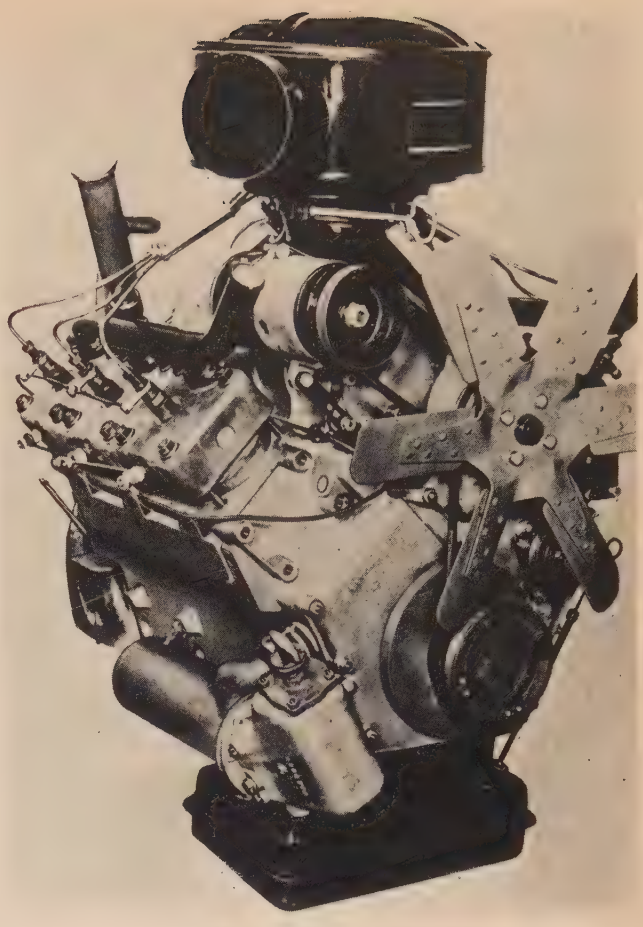


Fig. 28—Ford 6-cyl 2-stroke engine

and a simple piston design could be retained. The piston crown is, however, cooled from below by an oil spray from the small end of the connecting rod. The fuel consumption of this engine is, of course, somewhat higher than if direct injection had been chosen. This increased fuel consumption is due not only to the energy loss caused by the constriction of the throat, but also to the increased heat transfer following the intensified gas movement in the cylinder. Despite this, fuel consumption is affected only slightly by engine speed, as can be seen from Fig. 28. The accompanying curve, Fig. 29, of a modern European 4-stroke engine, having about the same combustion system, shows that the fuel consumption of both engines is approximately the same, but that the consumption of the 2-stroke engine is better in the area of lower engine speeds.

The layout of the Vee-engine is, in principle, the same as that of the Graf & Stift engine. Attempts were made, however, to build the engine more compactly. The power/weight ratio, at 2.6 kg per hp (5.73 lb per hp) for the 6-cyl engine, is lower than that of any other automotive diesel engine now on the European market. The block is aluminum. The in-line engines are equipped with balance shafts, so that the balance of inertia forces of a 2-cyl engine is the same as that of a 4-cyl 4-stroke engine and that of a 3-cyl engine

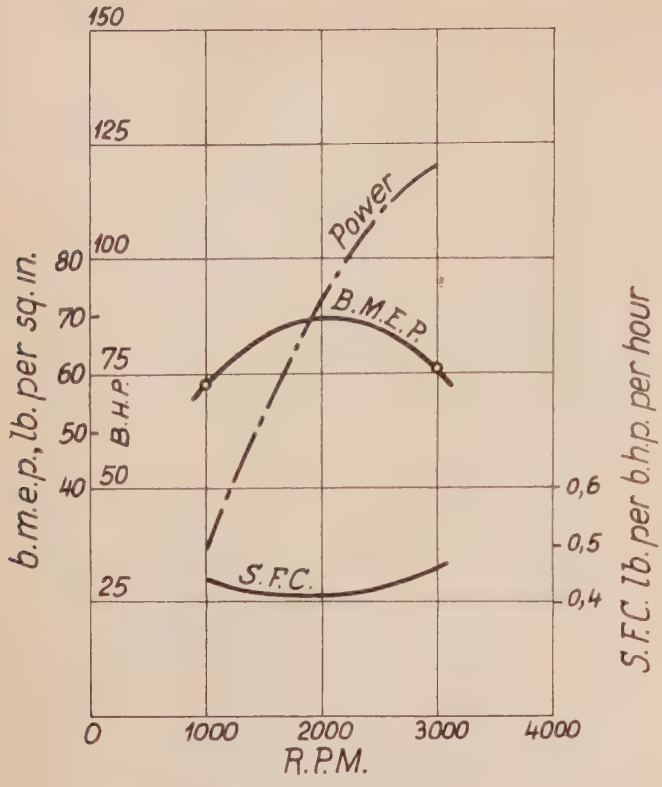


Fig. 29—Fuel consumption of Ford 6-cyl 2-stroke engine, 256-cu in. displacement

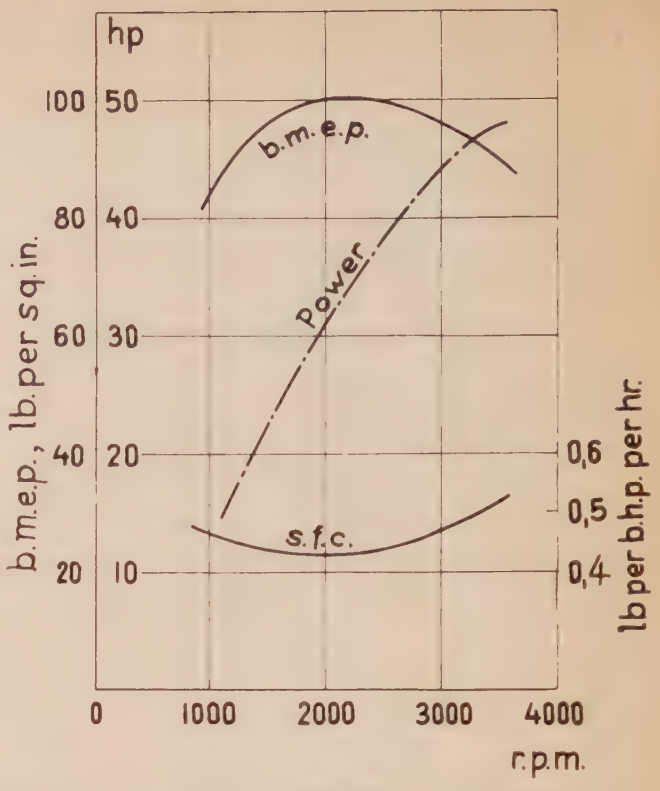


Fig. 30—Fuel consumption of Alfa-Romeo 4-cyl 4-stroke engine, 121-cu in. displacement

is the same as that of a 6-cyl 4-stroke engine. Fig. 30 shows a comparison between a 2-cyl 2-stroke and a 4-cyl 4-stroke engine in respect to heat transfer. It is somewhat lower in the 2-stroke.

Engines JW 50 to JW 400 of Jenbach Works, Jenbach, Austria—A series of relatively high-speed 2-stroke diesels was developed for installation in shunting locomotives and as small stationary units. The engines have a cylinder diameter of 5.91 in. and a stroke of 6.70 in. and run up to 1500 rpm. The 1-, 2-, and 3-cyl engines are in-line and are scavenged by Roots-blowers. The 4-, 8-, and 12-cyl engines are in Vee-form with a 90-deg Vee-angle and are scavenged by centrifugal blowers. (Fig. 31, 32). Due to the type of application for which these engines are intended the use of centrifugal blowers would be possible. Nevertheless, the engines up to 3-cyl are equipped with Roots-blowers because centrifugal blowers only work effectively if the air delivery out of the scavenging container is fairly uniform. If this is not the case, they require very large scavenging containers to counteract uneven air delivery, and such containers cannot be accommodated properly. The centrifugal blower is designed to be assembled as a separate unit and then mounted on the engine. Fig. 33 shows the "inflexible" torque curve which, however, has no adverse effect on the practical operation of these engines, as electrical or hydraulic torque converters are used with them.

Andritz Engine—Although the Andritz engine, with its low engine speed, cannot be called a high-performance engine, it is described briefly here in order to point out that the constructional principles given above also have their application in relatively low-speed marine engines. Fig. 34 shows the 4-cyl and Fig. 35 the 6-cyl Vee-form. The Vee-angle is 45 deg, bore and stroke are 9.05 × 13.4 in. At 530 rpm the cylinder output is 75 hp. Fig. 36 shows the curves for output and consumption. A particularly simple reversing gear makes these engines very suitable for heavy-duty marine application. Centrifugal blowers are used to supply the scavenging air.

So far, I have dealt only with unsupercharged loop-

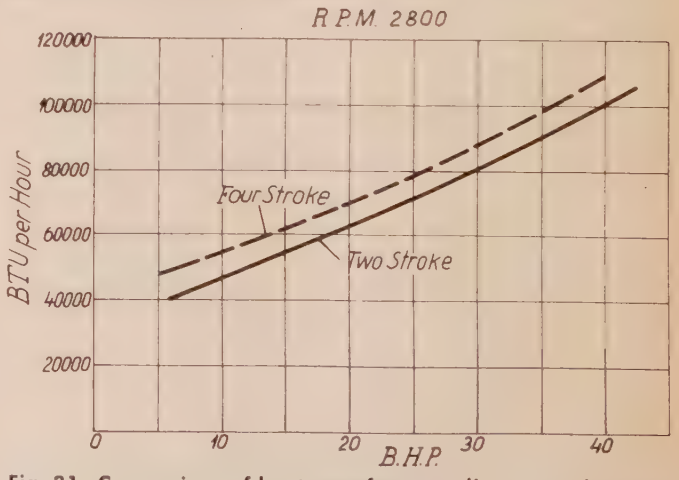


Fig. 31—Comparison of heat transfer to cooling water between 2-stroke and 4-stroke engines of same output

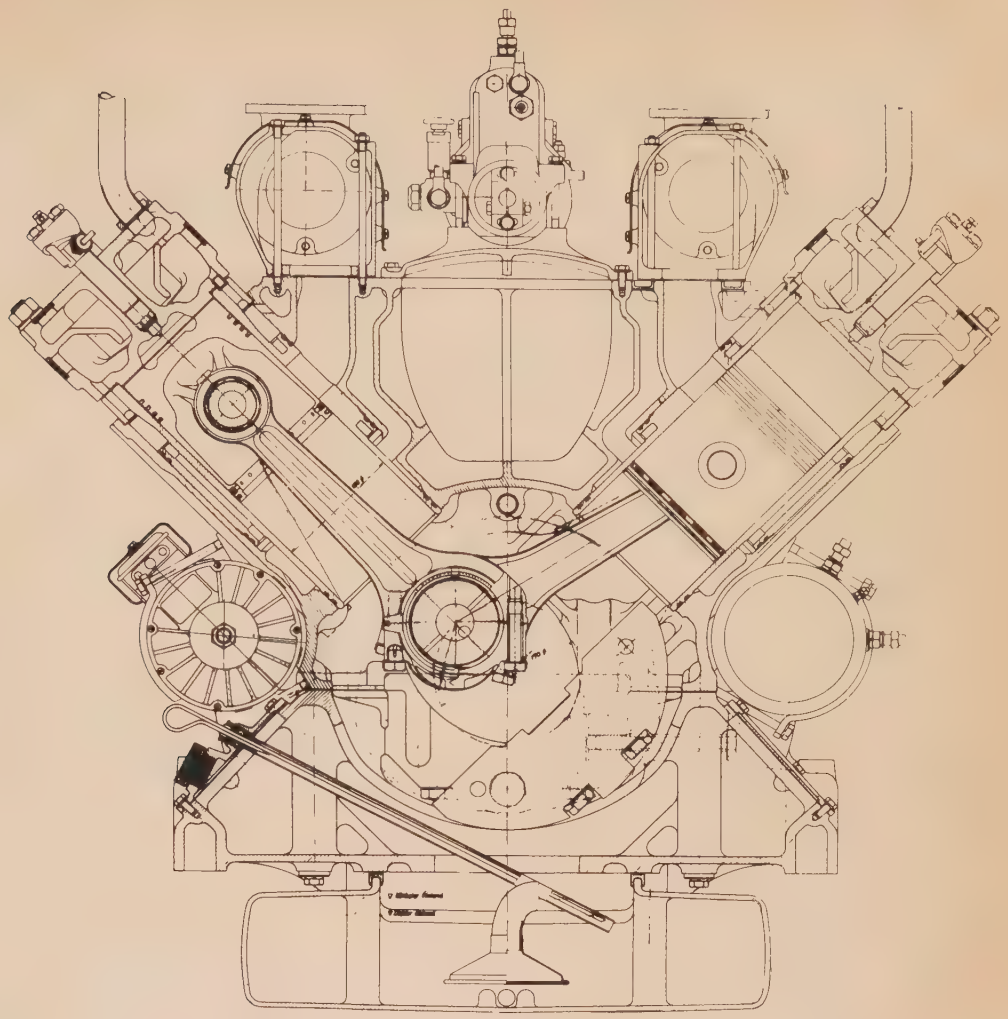


Fig. 32—Cross-section of 4-cyl 2-stroke engine JW 200

scavenged engines, and I showed that engines of this design can produce relatively high output per unit of swept volume. As with 4-stroke engines, an increase of output can equally be obtained by supercharging 2-stroke engines.

The loop-scavenged 2-stroke engine can be supercharged in two ways:

1. *Supercharging by additional valve gear*—By the addition of some valve arrangement, the symmetrical timing diagram which is not favorable to effective charging, can be changed into a nonsymmetrical one and, thus, a relatively limited increase in output can be obtained.

An ideal layout with respect to the gas exchange requires that the exhaust not only open earlier than the inlet but also that the exhaust passage close earlier than the inlet. This can be achieved by providing an additional timing element either ahead of the inlet or after the outlet. Fig. 37 shows such an arrangement, and Fig. 38 shows the increase in brake mean pressure obtained from a relatively low-speed engine which was tested in Graz. More effective and, therefore, more often used is a rotary valve inserted after the exhaust, a valve which either rotates uniformly or oscillates. This design is often applied to slow-speed engines, but it has not yet passed the experimental stage so far as high-speed engines are concerned. The latter are usually small engines, so that simplicity counts. The considerable complication cannot be economically justified when it is compared with the relatively small increase in output obtainable. In addition, the high-speed rotary valve limits reliability since soot and oil may easily clog the sealing surfaces. A final decision as to the applicability of these rotary valves for high-speed engines will depend on the results of future tests.

2. *Supercharging with turboblowers*—For high-duty engines of medium and high power, supercharging by means of a turboblower is much more important than the rotary valve. In this case the pressure level of the gas exchange is raised and, theoretically, supercharge up to any desired level can be obtained.

In some cases it is possible to compress the scavenging air by the turboblower alone, and this results in a very simple engine.

If the compression of the scavenging air requires relatively high power, either because greater air quan-

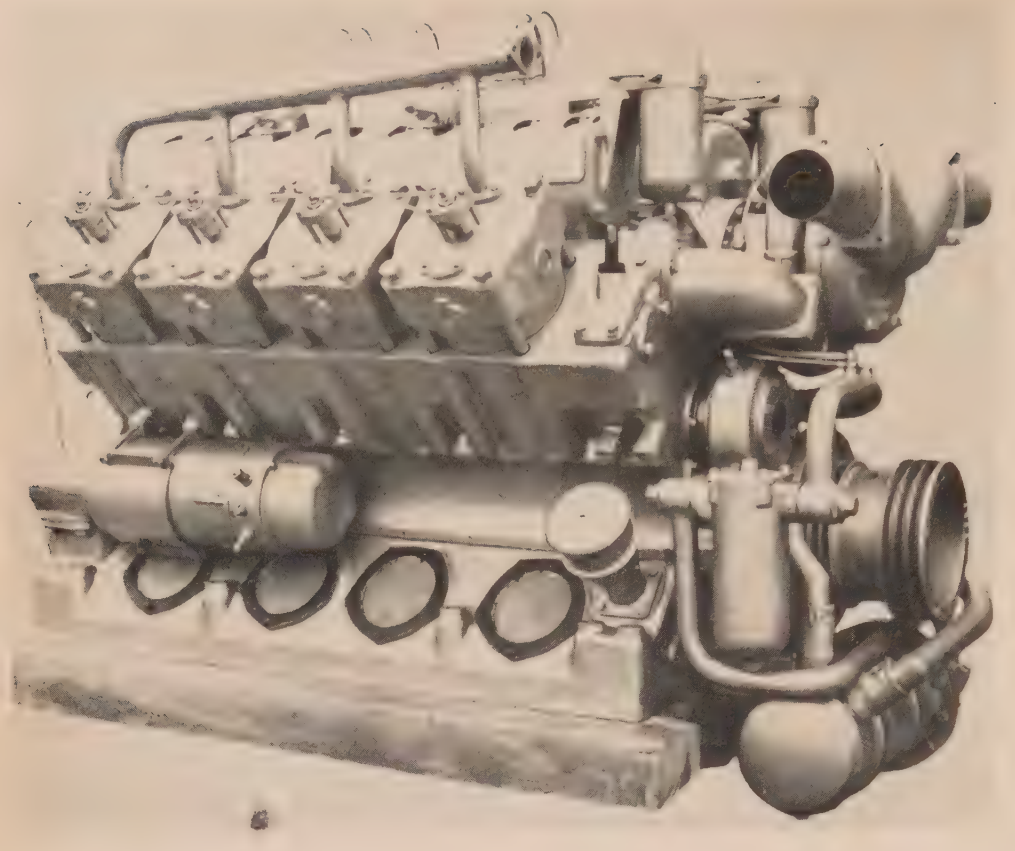


Fig. 33—View of 8-cyl 2-stroke engine JW 400

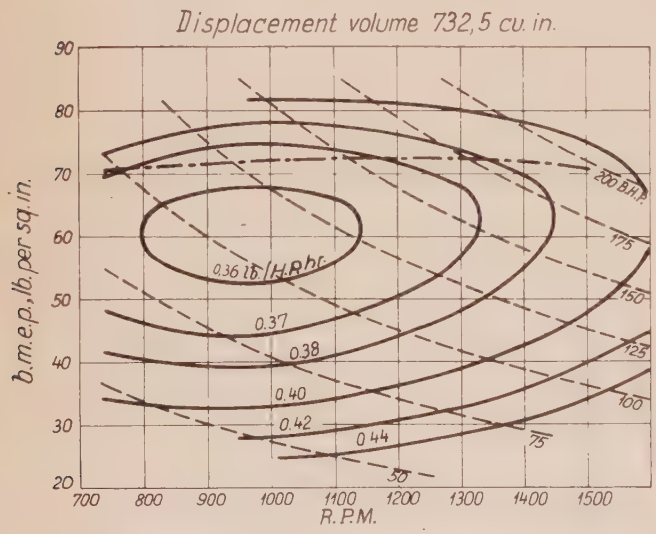


Fig. 34—Characteristics of 4-cyl 2-stroke engine JW 200

tities are required for scavenging or because high engine speed and short port opening periods necessitate a greater pressure drop, the scavenge air must be compressed by a mechanically driven blower in addition to the turboblower.

Fig. 39 shows the arrangement with turboblower alone, and some of the possible combinations of mechanically driven blowers and turbochargers.

The most usual arrangement is seen in Fig. 40, in which the turboblower is arranged ahead of the mechanically driven blower. As the arrangement with turboblower alone allows a very simple engine design, it is of interest to determine the field in which it can be applied. Although only tests will permit a final decision, the following calculation gives a good insight into the basic relationshp. This procedure is based on the method of examining processes in 2-stroke engines by calculation developed by us in Graz.

The calculation for the compression of scavenging air by the turboblower alone is derived from the fact that the output of the turboblower, to which gas at pressure P_t and absolute temperature T_t is supplied, must be sufficient to compress the equivalent scavenging air quantity to pressure P_{sc}. This results in:

$$K_t = \frac{(P_{sc}/P_o)^{\frac{k-1}{k}} - 1}{1 - (P_o/P_t)^{\frac{k_e-1}{k_e}}} = \frac{T_t}{T_o} \cdot \eta_{bt} \cdot \frac{k-1}{k} \cdot \frac{k_e}{k_e - 1} \cdot \frac{R_e}{R} \cdot \delta$$

For air delivery the following relationship exists:

$$L = 11360 \frac{\sigma_v^{3/2}}{A_i} \cdot \left(\frac{P_{sc}}{P_o}\right)^{\frac{k-1}{k}} \left(\frac{P_t}{P_o}\right)^{\frac{1}{k}} \cdot \frac{T_o}{\sqrt{T_{sc}}} \sqrt{1 - (P_t/P_{sc})^{\frac{k-1}{k}}}$$

and for the temperature of the residual gases, t_t:

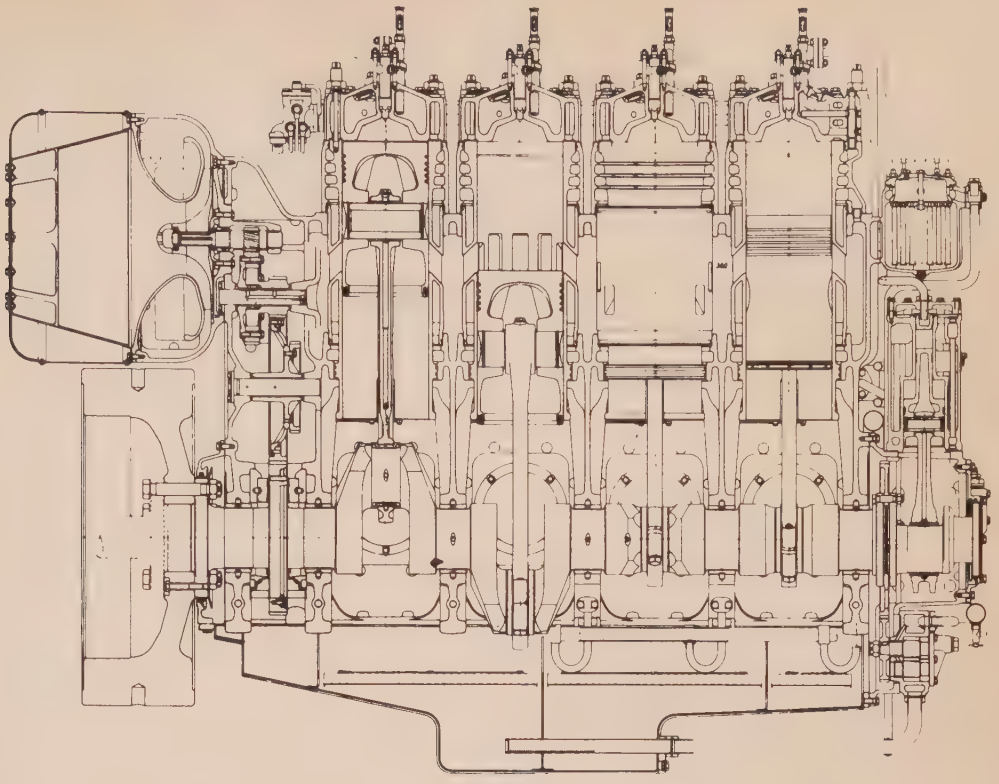


Fig. 35—Longitudinal section of 4-cyl 2-stroke engine UZ4A (Andritz)

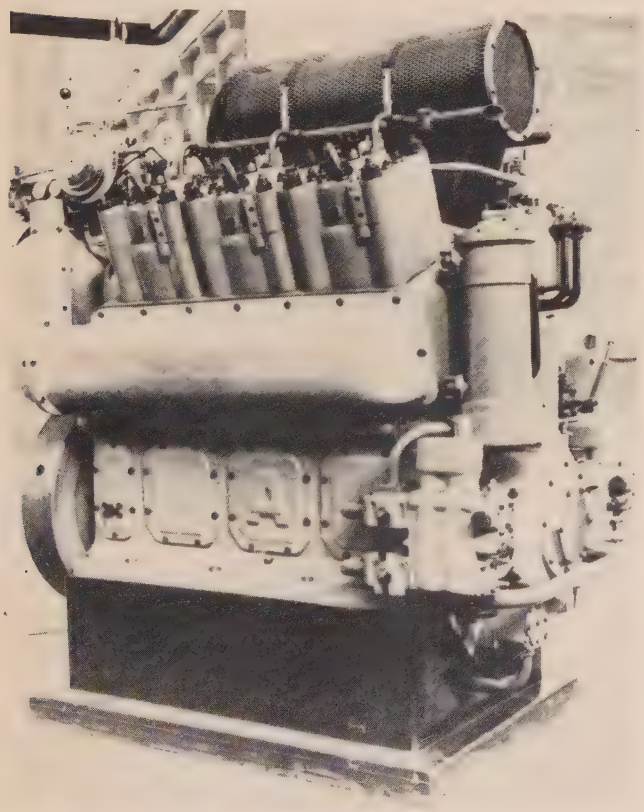


Fig. 36—View of 6-cyl 2-stroke engine UZ6A (Andritz)

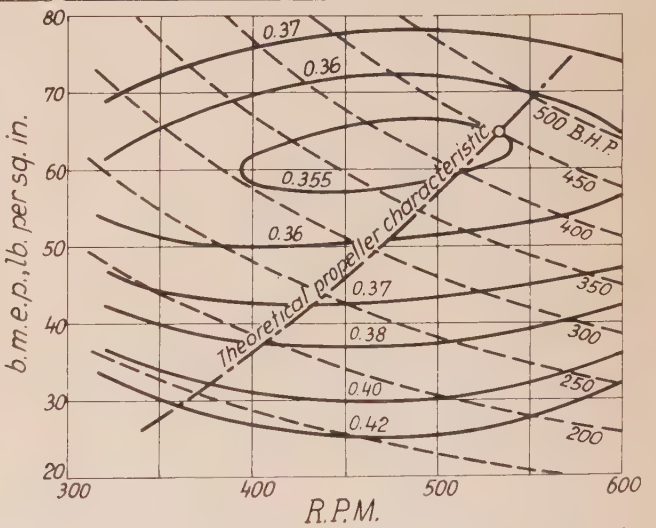


Fig. 37—Characteristics of 6-cyl 2-stroke engine UZ6A, displacement = 5160 cu in.

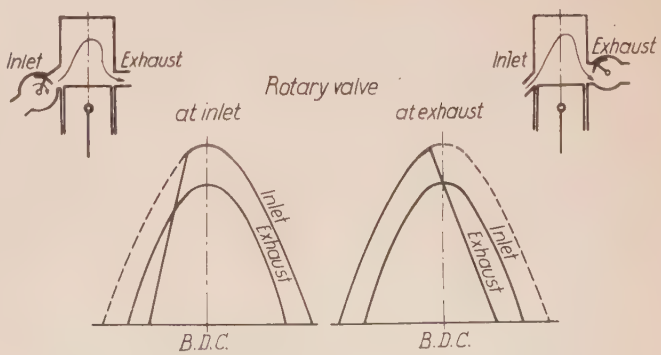


Fig. 38—Nonsymmetrical port area diagrams

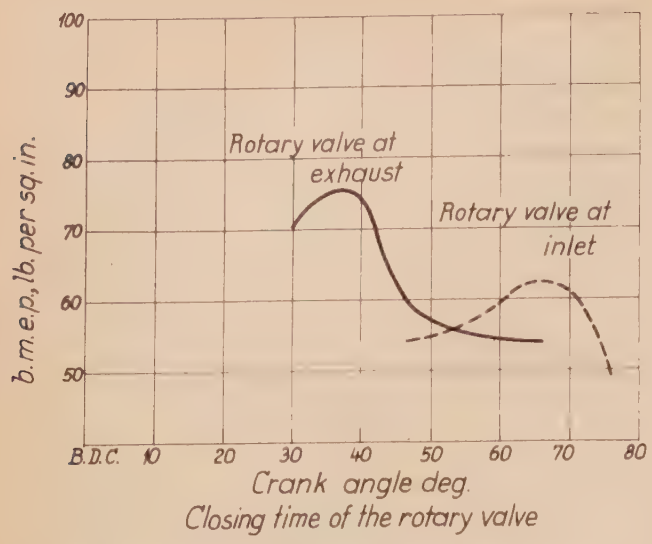


Fig. 39—Supercharge by rotary valve (Niedermayer)

$$\delta t_l - t_o = \frac{1510 \cdot H_u \cdot q_v}{\lambda V_{th} C p' {}_o{}^{t_l}} \cdot \frac{\eta_{ch}}{L} \cdot 10^3$$

For the possible useful pressure the factor

$$\text{bmep} = K(1 - \sigma_e)(1 - e^{-a'L}) - P_{fr}$$

is obtained, and for the scavenging efficiency

$$\eta_{sc} = 1 - e^{-a'L\frac{p_o}{p_l}} \cdot \left(\frac{P'_{ce}}{P_{ce}}\right)^{\frac{1}{k}} \cdot \delta'$$

is obtained.

Fig. 41 shows the evaluation of these equations for scavenge air cooled to 50 C, assuming average conditions. For this evaluation scavenging curves with $a = 1.2$, corresponding to fair loop-scavenging, and with $a = 1.5$, corresponding to uniflow scavenging, were assumed.

To limit the practical range, it can be assumed that the mixture of residual gases with air must not exceed a certain percentage. If one takes approximately 10%, giving a scavenging efficiency of 90%, the shaded area shows the practical range. From this it can be seen that exclusive turboblower scavenging is possibly only up to a certain speed, and this limiting value of A, and its practical equivalent $D \times n$, increases with increasing scavenging efficiency (increasing a).

In determining the curves, an efficiency of the turboblower of $\eta_{bt} = 55\%$, was assumed. An improvement of blower efficiency moves the limit curves to the right, a deterioration to the left.

The relationship represented by the curves may be evaluated in various ways, and the conditions at part load are of particular interest.

The character of the curves shows that, in the field of smaller load, the speed limit is lower than at higher loads. This leads to the conclusion that difficulties occur at part-load operation, unless, as is the case with propeller drive, the engine speed decreases during part-load operation.

Basically it must be stated that pure turbosupercharge is only applicable for relatively low speeds, as for instance with low-speed marine engines directly coupled to their propellers. Real high-output engines, therefore, cannot be designed for pure turboscavenging.

Fig. 40 shows the arrangement with exhaust-turbocompression and with a mechanically driven blower. It also shows the relationship between back-pressure and supercharge. The increase in output by supercharging and the fuel consumption as shown in Fig. 42, resulted from tests on a high-speed, high-output engine. One sees that in the given case, at an efficiency of about 50%, the fuel consumption remains approximately the same with increasing supercharge. If turbine and blower efficiency is improved, the fuel consumption decreases with increasing supercharge.

These efficiencies for turbocharging are not at all Utopian, but require careful layout of turbine, blower, and gas and air passages within the engine.

In my opinion the matching of turboblowers to engine should be made individually so as to obtain really satisfactory results. In Germany, several engine builders produce their own turboblowers, which enable them to produce excellent matching.

The limit to supercharging is established by the increasing pressures and the increasing thermal stresses which result from supercharging.

First, the conditions resulting from mechanical stresses will be examined.

If P_{cmax} is the maximum cylinder pressure and assuming that the specific load on bearings and the specific stress of material must not exceed certain values, the weight of the running gear and of other force-transmitting parts and, thus, the engine weight, is roughly proportional to:

$$C_1 P_{cmax} \frac{D^2\pi}{4} \cdot S$$

At a given speed the output is:

$$C_2 \text{bmep} \cdot \frac{D^2\pi}{4} \cdot S$$

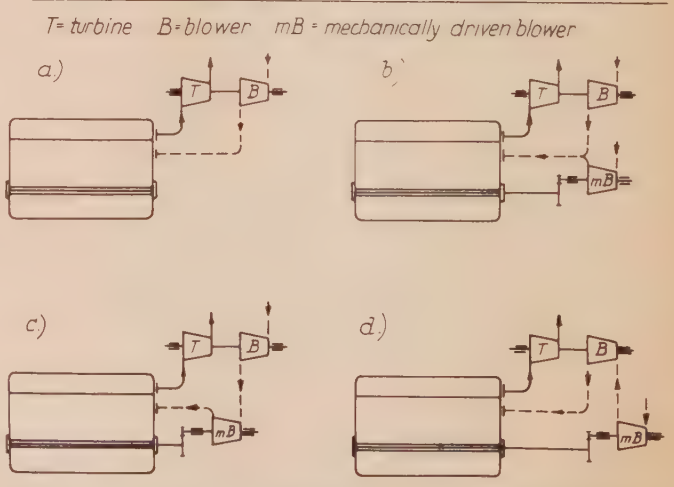


Fig. 40—Schematic arrangements of exhaust-driven blowers

Thereby, at given engine speed, the weight per bhp is:

$$\frac{\text{lb}}{\text{hp}} = C\,\frac{P_{e\max}}{2\text{bmep}}\,(\text{2-stroke}),\ \text{or}\ C\,\frac{P_{e\max}}{\text{bmep}}\,(\text{4-stroke})$$

Considerable advantages in weight and cost result from supercharging only if the ratio $P_{e\max}$/bmep decreases.

Fig. 43 shows some results of tests on Deutz and Sulzer engines. Since the curves will later be used for comparison with conditions existing in 4-stroke engines, 2 bmep has been plotted as abscissa for the 2-stroke engine. One sees that with increasing bmep, curves tend to fall so that reduction in the ratio of weight to power results from supercharging. In the range of higher supercharge, the course of the curves become flatter. Therefore, from the point of view of mechanical loads on the mechanism and of weight/power ratio, there does not exist a definite limit to supercharge. It is, however, assumed that, with increasing supercharge, the driving gear must be relatively heavy, that is, relatively small cylinders must be mounted on a large engine block so that the cylinder center distance becomes greater.

In the loop scavenged-engine the greater cylinder distance is necessary also to accommodate the scavenge-air passages between the cylinders. This added space also permits the use of the strong connecting rod and crankshaft needed by the supercharged engine. It should be pointed out in this connection that special care must be taken with the design of the piston-pin bearing.

The fact that the bearing load always works in the same direction can be used to advantage. This leads

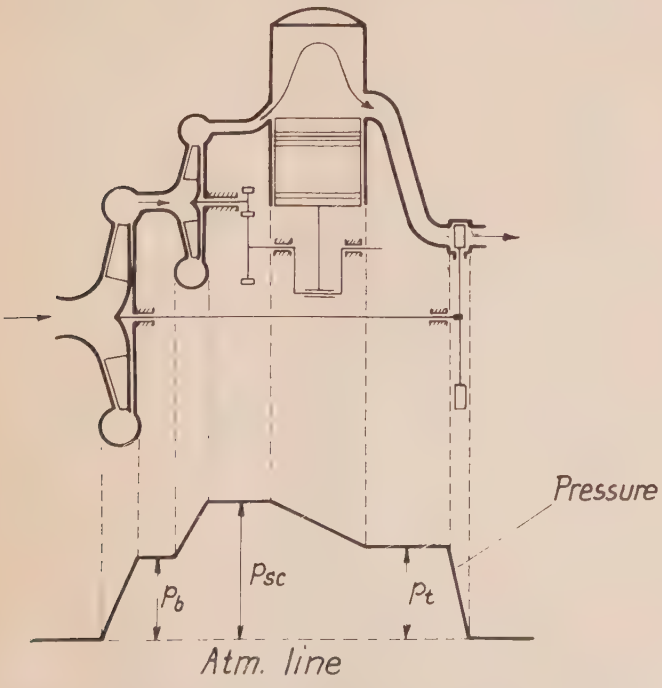


Fig. 41—Turbocharge of 2-stroke engine with BBC arrangement

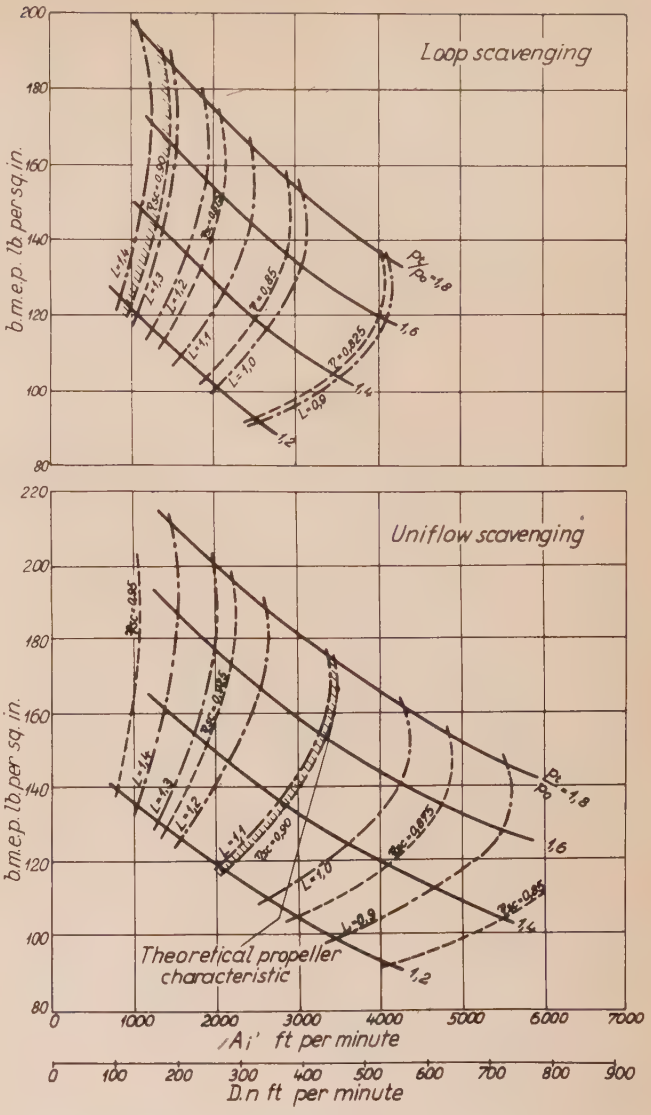


Fig. 42—Limits of scavenging in 2-stroke engine with pure turbocharging

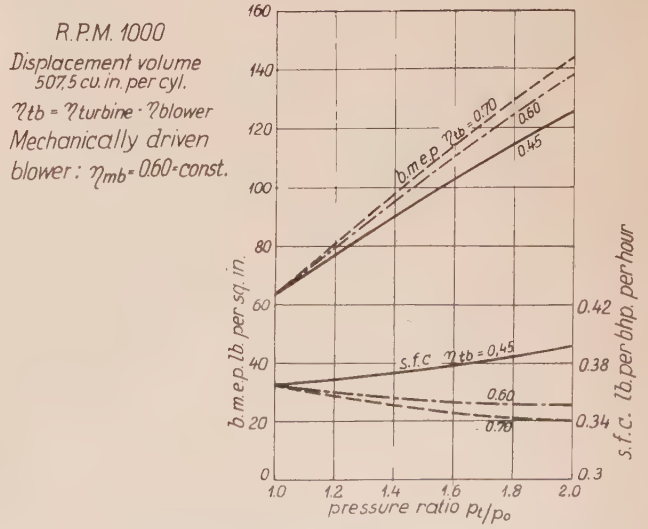


Fig. 43—Calculated effect of efficiency η_{bt} on turbocharged 2-stroke engine

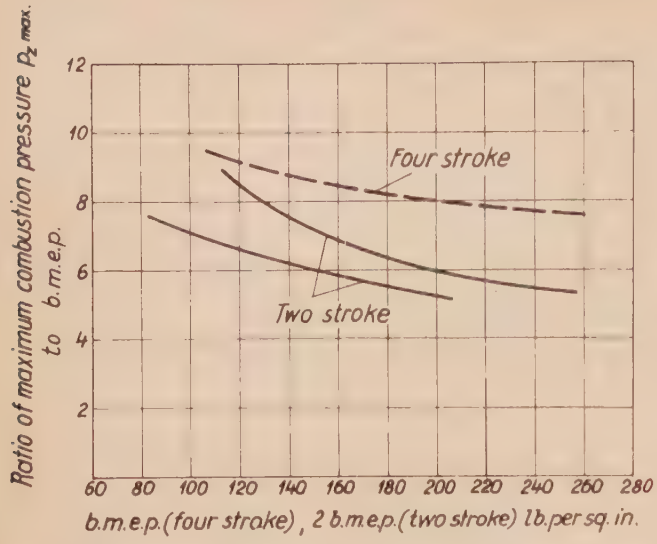


Fig. 44—Comparison of ratio $P_{z\,max}$/bmep of 4- and 2-stroke engines

to the design shown in Fig. 44. Thus, the greater cylinder center distance actually is not a disadvantage so far as highly supercharged engines are concerned. As one can avoid difficulties due to mechanical stresses by using a heavy running gear and still profit in weight, (as is shown in Fig. 43), the limit to the degree of supercharge is imposed by the thermal stresses in the engine, particularly in parts subject to continuous load, such as pistons and cylinders.

For loop scavenged engines a reliable cylinder head can be designed even for high supercharge since the cylinder is not pierced by large valves so that one has freedom in choosing the most suitable design with respect to heat flow. Care must be taken in designing the combustion chamber, and especially in designing the squish corners of the combustion chamber, since these, under certain conditions, may be subject to cracking. Cooling of the injection nozzles is of special significance but, in general, sufficient space is available for this purpose.

It has been found, concerning thermal conditions of the piston, that high supercharge is not possible with uncooled pistons. Simple piston cooling is possible, however, by spraying the underside of the piston crown through little holes in the small end of the connecting rod or by separate nozzles. The latter method is usually more effective. High supercharge at high speeds would require pistons with ducts for forced-oil circulation. Developments in this respect are in full swing and we cannot yet see where the limit for supercharge, as established by the piston, will be. It may be assumed that a 50% to 100% increase in output will be reached in the near future.

Finally, I would like to try to compare supercharged loop-scavenged and uniflow 2-cycle engines with the highly supercharged 4-stroke engine.

Loop-Scavenged 2-Stroke and Uniflow 2-Stroke Engines—When not supercharged, the uniflow 2-stroke engine has the advantage of considerably higher mean effective pressure, but it has a limited maximum speed owing to relatively high-acceleration forces in the valve gear. The freedom in designing the combustion chamber of loop-scavenged engines, in general, results in low fuel consumption and, furthermore, loop-scavenged engines are simple to make and can be produced cheaply.

In both the loop-scavenged and the uniflow-scavenged 2-stroke engine, the limit in output is set by the thermal stresses. It seems, from results obtained with large marine engines, that these limits are approximately the same with both scavenging systems and that a higher output is obtained sooner with the loop-scavenged engine (Fig. 45).

It is possible that an improvement in piston cooling or, in cases where the thermal load on the piston does not represent the limit, the loop-scavenged engine, with its simple cylinder head, can better withstand higher thermal stresses, so that a higher specific output can be obtained.

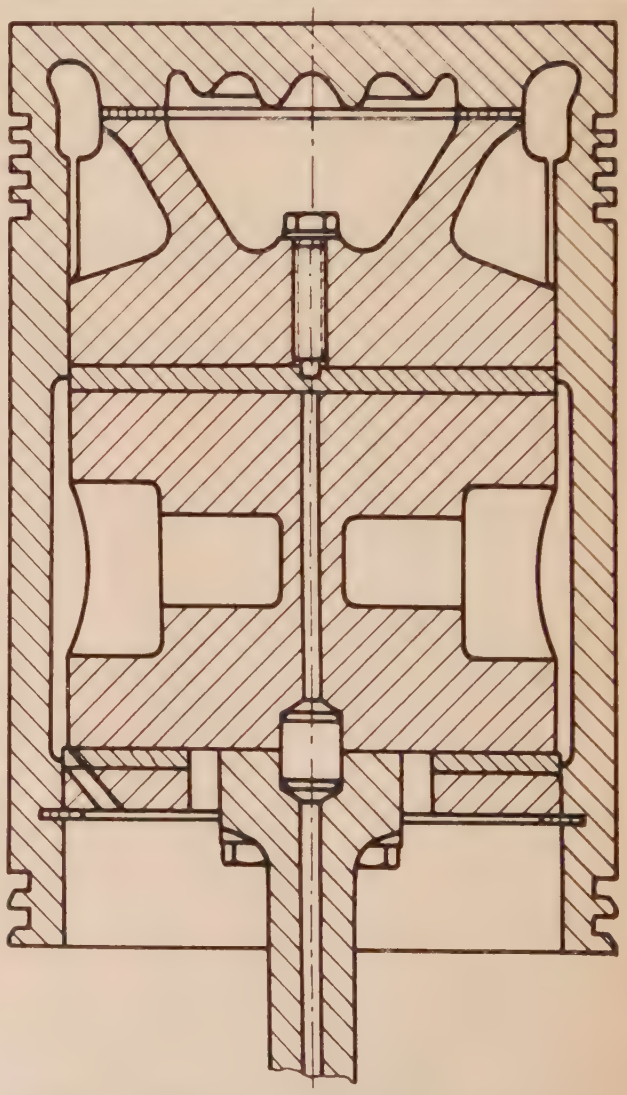


Fig. 45—Piston-pin bearing for 2-stroke engine

Uniflow scavenging has the advantage over a loop-scavenged engine that it can function over a wider range of $(D \times n)$ values by pure turbocharging, that is, without a mechanical blower. The usual service requirements for high output engines call for a mechanical blower in addition to the turboblower so that this particular advantage of uniflow scavenging does not apply in this case.

Supercharged Loop-Scavenged 2-Stroke Versus Highly Supercharged 4-Stroke Engines—Good insight into conditions existing in both engine types is obtained by evaluating the results of extensive tests which were carried out by Leiker on a Deutz loop-scavenged engine and by Eichelberg & Pflaum on a highly supercharged MAN 4-stroke engine.

Fig. 43 shows that the relation P_{cmax}/bmep or $P_{cmax}/2$ bmep is lower for the 2-stroke engine. Thus, at the same output higher cylinder pressures are necessary in the 4-stroke engine, and, as a result, the weight per horsepower of the loop-scavenged 2-stroke engine will, in general, be lower than that of the 4-stroke engine.

It can be shown that the heat transferred per unit of cylinder-wall area and thus the thermal load is roughly:

$$q_w = C \times D \times n \text{ bmep} \times q_n$$

If the values of q, derived from these tests, are plotted against bmep or (2 bmep) respectively, the curve shown in Fig. 45 is obtained. It shows that the thermal stresses are higher in the 2-cycle engine than in the supercharged 4-cycle. The pistons were uncooled in both cases.

The temperature in the center of the piston crown of the 2-stroke engine is higher at the same output, but the difference is not very large. It may be assumed that the greater thermal robustness of the valveless cylinder head of the loop-scavenged engine outweighs the difference in thermal load.

I am, therefore, of the opinion that the supercharged 2-stroke engine can well compete with the highly supercharged 4-stroke and will, very likely, be superior in cases where simple design and low weight per rated output are important.

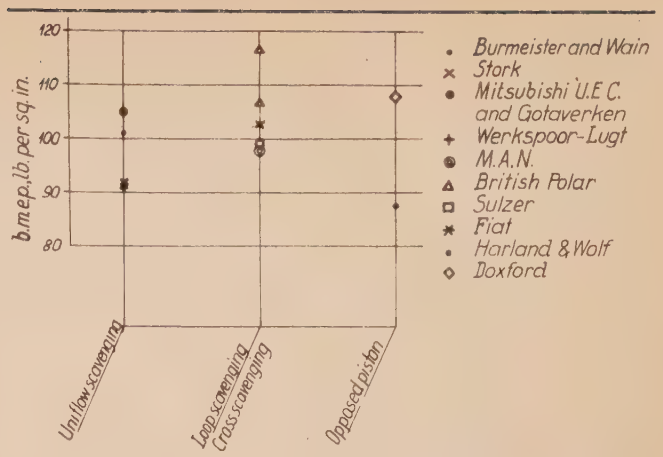


Fig. 46—Comparison of bmep of turbocharged 2-stroke engines

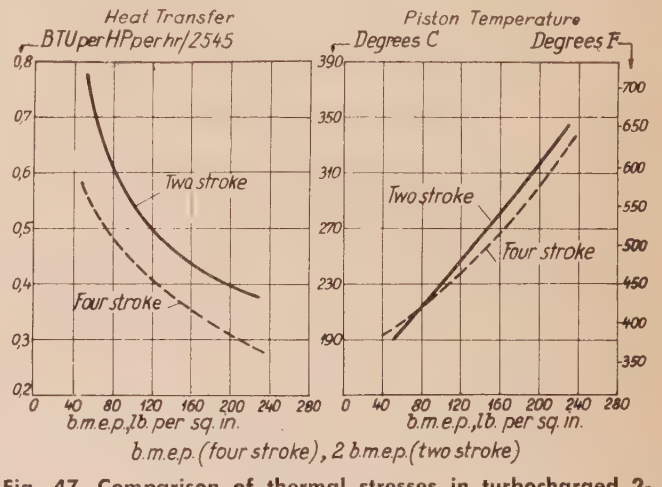


Fig. 47—Comparison of thermal stresses in turbocharged 2- and 4-stroke engines

APPENDIX

Index of Symbols

$A_i = C_i \times Dn$; C_i dependent on relative width of inlet ports and their mean flow coefficient, fpm

$A_e = C_e \times Du$; C_e dependent on the relative width of exhaust ports and their mean flow coefficient, fpm

a = Velocity of sound, fps

a' = Coefficient of L in the scavenging curve, dimensionless

bmep = Brake mean effective pressure, psi

bmip = Brake mean indicated pressure, psi

$C_p/_o{}^{t'}$ = Mean specific heat per mol at constant pressure, Btu per mol F

D = Bore, in.

f = Specific fuel consumption (effective hp), lb per bhp per hr

g = Acceleration of gravity, fps per sec

H_u = Caloric value, btu per lb

K = Mean indicated pressure with perfect scavenging and without loss of stroke by ports, psi

k = Ratio of specific heat C_p/C_v for air, dimensionless

k_e = Ratio of specific heat for exhaust gases, dimensionless

L = Delivery ratio, dimensionless

L_c = Delivery ratio at cylinder condition, dimensionless

n = Rpm

P = Absolute pressure, psi

P_c = Pressure in the cylinder, psia

P_{ee} = Pressure in the cylinder at opening of exhaust ports, psia

P'_{ee} = Pressure in the cylinder at closing of exhaust ports, psia

P_{cmax} = Maximum pressure in the cylinder, psia

P_{sc} = Scavenge pressure, psia

APPENDIX continued on following page.

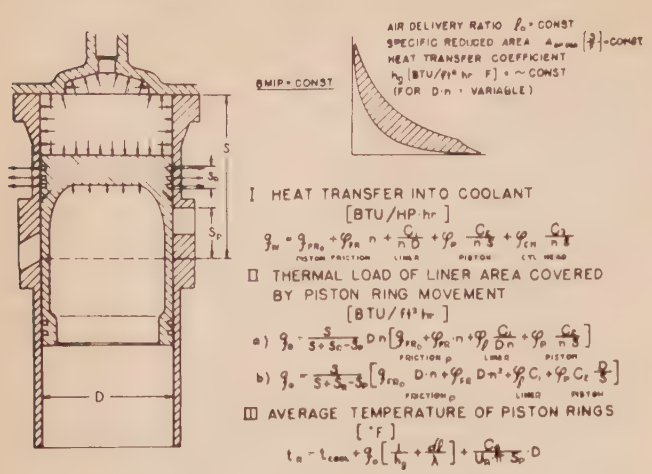


Fig. A—Thermal load of liner and piston

P_t = Pressure at the entrance of turbine, psia

P_a = Ambient pressure, psi

P_{fr} = Mean effective pressure of friction, psi

q_r = Part of heat in the exhaust gases, dimensionless

q_w = Heat transfer to unit of surface of cylinder wall, Btu per sq in. per hr

q_n = Heat transfer to cooling water, Btu per bhp per hr/2545

R = Gas constant for air ft per F

R_r = Gas constant for exhaust gases ft per F

s = Stroke, in.

t = Time, sec

t_a = Ambient temperature, F

t_t = Temperature of exhaust gases at entrance of turbine, F

T_a = Absolute ambient temperature, R

T_{sc} = Absolute temperature of scavenge air, R

T_t = Absolute temperature of exhaust gases at entrance of turbine, R

v = Specific volume, cu in. per lb

V_{th} = Volume of air theoretically necessary for the combustion of 1 lb of fuel, cu in. per lb

W_i = Velocity in the inlet ports, fps

W_{mi} = Mean velocity in the inlet ports, fps

W_e = Velocity in the exhaust ports, fps

W_{me} = Mean velocity in the exhaust ports, fps

x = Way of gas particle, in.

z = Relative volume of the cylinder, dimensionless

P_i = Absolute pressure in the inlet ports, psi

P_e = Absolute pressure in the exhaust ports, psi

α = Crank angle, deg

δ = Weight ratio of air to exhaust gases, dimensionless

δ' = Ratio of volume of one mol air to one mol exhaust gases, dimensionless

ϵ = Compression ratio, dimensionless

η_b = Adiabatic blower efficiency, dimensionless

η_{bt} = Product of the efficiencies of turbine and blower, dimensionless

η_{ch} = Charging efficiency, dimensionless

η_{sc} = Scavenging efficiency, dimensionless

λ = Excess air factor, dimensionless

μ_i = Flow coefficient of inlet ports, dimensionless

μ_e = Flow coefficient of exhaust ports, dimensionless

σ_i = Relative height of inlet ports dependent on piston position, dimensionless

σ_e = Relative height of exhaust ports dependent on piston position, dimensionless

τ_i = Coefficient of heating, dimensionless

DISCUSSION

Questions Advantage of Loop-Scavenged Two-Cycle Engine over Present-Day Powerplants

—C. G. A. Rosen
Caterpillar Tractor Co.

IN reviewing Dr. List's paper, we gain the impression that the 2-cycle engine in the loop-scavenged configuration has achieved a status where it is directly competitive to the 4-cycle engine in high-output performance. This is most interesting in evaluating present-day powerplants. The remarks I wish to make will be focused on the future, on some crystal-ball gazing. It is essential that we evaluate the future in terms of multiple improvements in performance in all types of powerplants. What is the outlook on the broad horizon with regard to the 2-cycle loop-scavenged engine as a contender for supremacy in the powerplant field? Our interests are directed toward powerplants which can take full advantage of high turbosupercharging and achieve not only comparable results but superior performance in the powerplant field.

It is apparent that Dr. List has made a notable contribution to the 2-cycle loop-scavenged engine by achieving an extremely short period of glow-down or travel of the crank between the time the exhaust port opens and the inlet port opens. The graph in Fig. 6 indicates that this period is less than 10 deg of crank angle.

This is achieved by a rather courageous design based upon a knowledge of the pulsating flows wherein the backfiring gases are even permitted to dilute the mixture of the incoming fresh air. This fact, plus the situation in which the exhaust gas temperatures are relatively much lower than in the 4-cycle supercharged engine, should permit a higher degree of supercharged boost in a 2-cycle loop-scavenged turbocharged engine. The question is, what is the limiting factor? Is the limiting factor in comparing 4-cycle turbocharged against 2-cycle loop-scavenged turbocharged engines, (1) the exhaust gas temperature determining the critical blade temperatures, or (2) the thermal loading of the piston? There is reason to believe that the 2-cycle loop-scavenged engine has an advantage in this regard. Is this true?

If we are to consider higher supercharged engines, say over 100% boost, what sort of problems would be anticipated with rings riding over ports in the liner? It is always known that an interrupted cylinder surface or bearing surface is difficult to lubricate. Has anything been done in the way of utilizing an inverted loop-scavenged system in which the tiers of ports are in the upper portion of the liner, out of range of ring travel and controlled by a sleeve valve. It is to be expected that the simplicity of the loop-scavenged 2-cycle engine is its principal virtue; however, when high performance is a factor, this simplicity might be deviated from to the extent of providing mechanisms which would permit higher boost than are now possible with the conventional loop-scavenged engine.

In considering these factors and anticipating the development of 2-cycle loop-scavenged engines in multiple ratios of power or

performance than now found available, one must consider, for instance, the factor of oil control. In Fig. 14 the sealing belt is indicated in which oil rings are providing effective control of the lubrication to achieve the desired lubricating oil consumption. However, it is also significant that this sealing belt has a reverse function when high boost pressures are employed in sealing the crankcase of the engine from the scavenge ports or the higher air pressure from the scavenge box.

The writer is familiar with earlier developments in large-bore 2-cycle engines where cylinder lubricators were employed and various methods were attempted to avoid bridging the ports in the distribution of an oil supply from the lower portion of the cylinder to the upper portion of the cylinder. Some of these methods employed timed lube oil injection pumps above the ports and some attempted even to lubricate through the piston into the ring belt as the piston approached top center. All these efforts were directed toward reducing cylinder wear. In Fig. 17 the mean wear values given for the Graf & Stift are considered quite satisfactory for ordinary cylinder irons when compared with the 4-stroke minimum values. However, the consideration for higher supercharge pressures may provide a problem in permitting oil to travel across the ports and distribute the film properly in the upper portion of the cylinder.

Fig. 16 indicates that the maximum side pressure of the piston is in the port areas. It is recognized that the bridges between the ports are mae-wested in order to provide additional clearance against the growth of the bridges into the bore diameter. However, the heavy side thrust in the port area would indicate the skirt's opportunity to provide maximum heat transfer in this portion of the cylinder where the skirt is permitted to have contact under pressure on the cylinder wall.

The other factor in higher output or multiple outputs of present engines is the port-clogging problem which again is dependent on the lubricating oil consumption and the means of distributing oil across the ports for the lubrication of the upper cylinder surface. Would, therefore, the conventional-type port-scavenged engines indicate higher port-clogging problems in higher output engines than is presently permissible.

In Fig. 47, the heat transfer to the cooling water at 200 bmep indicates a 30% rise for the 2-stroke cycle as against the 4-stroke cycle, whereas the piston-crown temperature for the same bmep only shows a 6% increase in the center of the piston crown. Isn't this predicated on a high rate of heat transfer from the piston to the cooling oil and not on any significant heat transfer to the scavenge air during the scavenging period? Higher supercharge means higher pressure levels and higher charging temperatures which in turn would provide less cooling effect on the piston crown. In this connection isn't the heat-transfer picture incomplete unless the heat rejected to the oil is given consideration in comparing the 2- and 4-stroke cycle engines to get a balanced picture?

It was known to the author recently that an aircooled 2-cycle engine showed as much heat rejection to the piston cooling oil as would normally be expected in heat rejection to the water jacket of a conventional engine. The total heat rejection should therefore be evaluated in the comparison of the 2- and 4-stroke-cycle engines or Fig. 47.

It is interesting to note that the piston temperature curve for Fig. 47 is taken in the center of the piston crown. This temperature naturally would be a factor in the life of the piston. However, is this the critical area rather than the top ring belt? Under ordinary circumstances it would be considered that the temperature of the piston ring belt would have a greater influence on engine durability than would the center of the piston crown.

In Fig. 46, it is interesting to note the high performance, high bmep achieved in the British Polar loop-scavenged engine. What is the reason for this outstanding accomplishment, and how can it be accounted for? Its bmep values are so much more significant than the other engines illustrated in the graph.

In Fig. 24 the work of Prof. Pischinger indicating the Btu heat transfer in relation to squish velocity is of much interest. It would be valuable to know what is the definition and the value of ϵ ranging from 13.6 to 16.2. In some of the author's review of the experiments conducted by Triebnigg it would seem as though the ϵ values could be compression ratios. If this is true, what values are kept constant during the experiment? Is the area of the combustion chamber or diameter of combustion chamber to diameter of cylinder kept constant and is the clearance between the piston and cylinder head kept constant? It would be interesting to know what factors were controlled to achieve the velocities indicated and what compression ratios were applied.

It is of interest to note that, in the various engines which have been designed by Dr. List, three different combustion chambers have been employed. In two cases the piston is of the flat-top design. In one, the swirl chamber is employed, and in the second the direct-injection fuel system employed. The Ford engine shows a swirl chamber. In this connection, what is the ratio of the swirl chamber volume to the main chamber volume? In the direct-injection engine what optimum number of orifices are best suited for the type of combustion chamber employed? The third type of combustion chamber in Fig. 32 shows a depression in the cylinder head of a torous-type design, and the piston is fitted with a circular groove. Is this for purposes of encouraging directed velocities to the fuel spray from the squish between the head and the piston or are there other factors which are significant in this combustion chamber?

Where are the limits determining the selection of a swirl chamber, a direct-injection system, and the combustion chamber fitted with a depression in the piston?

Discusses Development of Future High-Output 2-Stroke Diesel Engines

—K. C. Karde
Harnischfeger Corp.

THE author's paper covers the entire field of the major problems related to the design and operation of the high-output loop-scavenged 2-stroke diesel engine. I would like to add a few remarks covering, in a more detailed way, some of the essentials in the future development of the *exhaust-turbocharged* version of this engine. Dr. List's paper clearly indicates the necessity of an additional mechanically driven blower supplying sufficient air for starting, low load, and for operation under sudden load and speed changes, as they prevail in the automotive application.

For this type of operation the combination of an exhaust-turboblower plus a positive displacement blower of the Roots type, arranged in series, seems to be a good solution for today and the near future. Calculations and tests indicate that cutting down the Roots-blower capacity to about 65–80% of the nonsupercharged version will, above certain load and speed conditions, result in feeding energy back into the Roots-blower. Thus the disadvantage of the rather low adiabatic efficiency of this blower will be approximately eliminated in the higher bmep and speed range.

As already stated by Dr. List, the upper limitation of the specific output will be imposed by the thermal load, which is approximately proportional to $D \times n$. In view of the paramount influence of the thermal load upon the limitation of the highest allowable bmep a more accurate approach to this problem seems to be worth-while.

The schematic view on the left side of Fig. A illustrates the heat dissipation at the shown piston position. At any small time interval, that is, at any piston position, heat is transferred from the high-temperature gas to the surrounding walls at approximately the same rate. It is apparent that both the cylinder-head surface and the piston crown are exposed to the cylinder gas during the entire cycle. The direct heat transfer to the cylinder walls, however, is a function of the piston stroke, the upper end of these walls absorbing heat during nearly the full cycle, the lower end receiving only a small fraction, since the piston crown uncovers this area for a short period only. In some instances about 80% of the piston-crown heat is transferred through the ring zone into the cylinder walls. The remainder is dissipated by the lube oil and air at the underside of the piston. Friction between the piston and cylinder walls disappears as heat into the portion of the cylinder walls covered by the piston-ring movement.

Assuming the indicator diagram being constant, the heat transfer into the coolant, the thermal load of the liner area covered by piston ring movement, and the average temperature of the piston rings can be expressed for any size of an engine as it is shown in the formulas 1, 2, and 3, in Fig. A.

For different engine types the piston friction and the constant factors C_1, C_2, C_3, and C_4 can be determined by engine tests. Evaluating such test data and the terms of the formulas of Fig. A

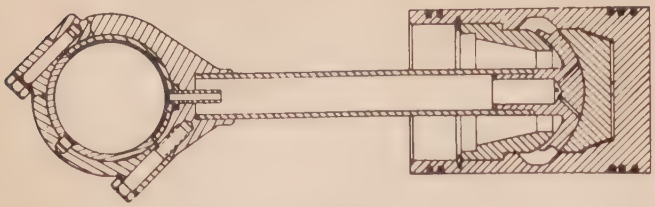


Fig. B—Mushroom piston connecting-rod assembly

enables us to appraise the thermal conditions of a new engine design as a function of the piston diameter D, stroke s, and the engine rpm n.

Essential for the dependable function of an engine is a low piston-ring temperature. It can be lowered by providing a sufficiently high stroke s/bore D ratio according to formula 3, and by decreasing the heat flow to the rings. Decreasing the heat flow can be achieved by two basic means:

1. Restricting the heat transfer into the piston crown by selecting a high-temperature material with a low coefficient of conductivity for the top of the piston.

2. Artificial cooling of the underside of the piston crown by arranging a sufficient flow of coolant, such as lube oil.

Exploring all possible means of diminishing the heat transfer into the piston will be a must in the future of the highly supercharged diesel engine, since the surface coefficient of heat transfer from the gas into the surrounding walls is about proportional $P^{2/3} \times T^{1/3}$. What can be achieved in this manner is best illustrated by the example of the Junkers aircraft diesel engine, Jumo 207D. Without any special protection of the piston crown it is impossible to operate the engine under higher loads. When using the high-temperature fireplate on top of the piston (average temperature 1000 C or 1800 F) the heat flow into the ring zone still amounts to about 30% of the total heat dissipation into the coolant. Using, for example, quartz glass as plate material will reduce the amount of heat going into the ring zone to about 5%. Thus proper function of the piston rings and a minimum of wear is fully obtained. This result has been confirmed by engine tests.

Recent and future development of materials in the field of metals and ceramics will be the source for the piston crown material of the highly turbocharged 2-stroke diesel engine.

In anticipation of the high mechanical piston loads of the highly supercharged engine, we have successfully put into reality an old engineering goal, namely the spherical-type piston support. As shown in Fig. B, the piston connecting-rod assembly consists of the following parts: the piston body itself which is very simple in design, a spacer plate, a concave bearing cup, a convex rod retainer, and the connecting rod itself. The upper end of the rod is a mushroom-shaped carrier, suitably connected to the lower end by a light-weight tube.

The advantages of this design can be enumerated as follows:

1. The mushroom design results in a bearing-area increase of approximately 150 through 250%. Additionally the load distribution is completely uniform.

2. The spherical piston support eliminates all misalignment problems.

3. Rotational movement of the piston provides uniform thrust and ring load distribution. Consequently, wear of the involved parts is reduced to the lowest possible level.

4. The spacer plate allows a wide latitude with respect to the thermal load conditions: it can be made of various materials, having either a high or a low coefficient of heat conductivity. Thus the heat flow can be adjusted to the best operating conditions. It can be easily modified to intensive piston cooling by the lubricant, if desired.

5. The weight reduction of this design amounts to about 15%, resulting in lower disturbing forces with regard to balancing and vibration.

Test results of the first units over a period of more than 4000 hr under extremely high-load conditions on our standard production engine show exceptionally low wear. The aluminum bearing carrier and the mushroom had no measurable wear at all and were in perfect condition. This new design also proved its dependability in supercharger test runs up to a bmep of 170 psi at 1800 rpm in our 4-cyl uniflow-scavenged engine.

In view of these excellent results we immediately started a rather extensive tooling program, to enable us to produce this mushroom piston connecting-rod assembly for our line of 2-stroke diesel engines.

Evaluates Claim of Lower Weight Per Horsepower for 2-Cycle Engine

—Ralph Miller
Nordberg Mfg. Co.

OBSERVATIONS and experience with both 2- and 4-cycle engines confirm Dr. List's conclusion that the loop-scavenged 2-cycle engine is superior from a standpoint of simplicity.

The claim of lower weight per horsepower is less readily accepted.

Let us look at the horsepower equation:

For 4-cycle:

$$\text{ihp} = \frac{P\,L\,A\,N}{33{,}000 \times 2} \qquad (1)$$

For 2-cycle:

$$\text{ihp} = \frac{P\,L\,A\,N}{33{,}000} \qquad (2)$$

This gives rise to the popular misconception, founded on ignorance of other factors affecting output, that 2-cycle engines develop twice the ihp of the 4-cycle engine of equal dimensions.

If we now compare engines of equal cylinder dimensions, piston area A, can be dropped and the values of P, L, and N only need be considered.

However, the evaluation must be based upon equal air manifold pressure and temperature and the fact that the clearance volume is such that the ratio of compression pressure to manifold pressure is the same in both engines. Further, consider that the 4-cycle engine is supercharged and operates with scavenging.

The mean indicated pressure P_i is a function of the weight of oxygen in the cylinder charge, and we have:

$$P_i = \frac{37{,}150 \cdot P_c \cdot L_{sc}}{(R-1) \times T_c \times W} \qquad (3)$$

or

$$P_i = \frac{37{,}150 \times P_c^{.72} \times P_{mf}^{.28} \times E_{sc}}{(R-1) \times T_{mf} \times W} \qquad (4)$$

where:

P_c = Compression pressure, psia
P_{mf} = Scavenging or supercharging pressure, psia
R = Actual volume compression ratio
T_{mf} = Temperature of scavenging or supercharging air, R
E_{sc} = Scavenging efficiency (purity factor)
W = Air, lb per ihp per hr

Showing, what is obvious, that at constant W the imep is proportional to the scavenging efficiency.

Properly designed supercharged 4-cycle engines operate with 100% scavenging efficiency.

A supercharged port-scavenged engine of the types being discussed is shown to have a scavenging efficiency of 90% (Fig. 9). We then have 2-cycle imep = 90% of 4-cycle imep, and where the power stroke in a port-scavenged engine terminates when the scavenging ports open.

From Fig. 11 and the equation for calculation of port height, an inlet port height of 25% of the stroke (also confirmed by scaling Fig. 25) is used in these high-speed engines.

The imep expressed on the basis of 100% stroke and used in the horsepower equation (2) is then reduced to 75% of the actual imep calculated from equation (1).

The factor $L \times N$ expresses piston speed in feet per minute. When L is increased in the 2-cycle engine, scavenging efficiency suffers. When N is increased, port height must be increased, and effective stroke decreases.

The 4-cycle engine does not suffer loss of scavenging efficiency with increase in stroke or rpm. Other factors limit $L \times N$ in 4-cycle engines at much higher values.

A review of successful engines in commercial use shows that 2-cycle engines in this class operate at about 75% of 4-cycle piston speeds or less.

We now have:

1. 2-cycle scavenging efficiency = 0.9 of 4-cycle
2. 2-cycle effective stroke = 0.75 of 4-cycle
3. 2-cycle piston speed = 0.75 of 4-cycle

Hence, the product $P\ L\ N$ in the 2-cycle horsepower equation (2) must be multiplied by $0.9 \times 0.75 \times 0.75$, or 0.507.

This example, representing modern advanced design in both 2- and 4-cycle engines, proves that the output per unit cylinder volume is about equal.

On the basis of rating at equal thermal load rather than equal air/fuel ratio as above, the 4-cycle engine is far superior to the 2-cycle in output per unit cylinder volume. The heat-transfer curve in Fig. 47 confirms this.

Dr. List's statement to the effect that the loop-scavenged engine has greater thermal robustness is questionable, in view of experience which shows that piston rings and liners and not the cylinder head are first to fail with increasing thermal load. This is borne out by a comparison of the bmep ratings of two well-known high-output 2-cycle locomotive diesel engines. One is an $8\frac{1}{2} \times 10$ uniflow, the other an $8\frac{1}{8} \times 10$ opposed-piston engine. The latter has no cylinder head, yet it is rated at a lower bmep and hence is operating at a lower thermal load than the former.

In the Miller System of Supercharging 2- and 4-cycle diesel engines, the weight of air entrapped in the combustion chamber is increased by reducing the compression temperature which in the conventional engine with good cold-starting characteristics far exceeds the temperature required for ignition when operating at high loads due to dilution and hot cylinder walls.

The weight of air in the combustion chamber is:

$$W = \frac{V_c \times P_2 \times 144 \times E_{sc}}{R \times T_2} \qquad (5)$$

where:

V_c = Clearance volume
P_2 = Compression pressure
R = Air constant
T_2 = Compression temperature, R

The mean indicated pressure is then inversely proportional to the compression temperature or directly proportional to the manifold pressure to the exponent $\frac{n-1}{n}$. (See equation 4.)

In the conventional system, the ratio of compression to manifold pressure is fixed for the engine.

In the turbocharged Miller System, the manifold pressure is independent of the compression pressure which remains substantially constant as the manifold pressure is raised to increase the weight of air and thereby the bmep.

Then, since the maximum combustion pressure is a function of compression pressure and only to a small degree increases with bmep, the Miller System of turbocharging 2- and 4-cycle engines fulfills better than any other system the requirement which Dr. List correctly lays down, namely, "Considerable advantage in weight and cost results from supercharging only if the ratio P_{cmax}/bmep decreases."

Questions Aspects Of 2-Stroke Engine

—D. S. Sanborn
Consulting Engineer

DR. LIST'S pioneering in the analytical research and practical development of the 2-stroke engine has done much to raise "Performance Barriers." The methods of scavenging analysis that he has developed will become even more important now that mechanical and thermal limitations of the engine have been largely overcome and problems of flow-dynamics become all-important at high speeds and outputs.

Dr. List's consideration of the following questions that arise from reading the paper, will be greatly appreciated:

1. With regard to the proper reference volume upon which to base scavenging efficiency; in the author's book: "The Charge Exchange of Internal-Combustion Engines" (Vol. 4, Part 2), "stroke volume" is apparently assumed as reference volume for scavenging efficiency, while "average cylinder volume during scavenging" is used as the reference volume for the air expenditure which accomplishes this scavenging. Will this lead to erroneous impressions of scavenging performance in modern high-speed engines having very wide port timings?

2. Have any investigations of Schnürle scavenging systems disclosed whether or not the vortex system generated in the cylinder during scavenging persists throughout compression and combustion?

3. Has Dr. List any experience with or comments on the possibilities of "Exhaust Pulse Supercharging" as developed by H. D. Carter of Crossley Bros., Ltd.?

4. Comparing the effect of inlet rotary valve versus exhaust rotary valve in Fig. 38, it would appear that both valve systems reduce the losses that emanate from the exhaust port closing after the inlet port. Inlet control, however, increases the expansion ratio compared to the compression ratio and might be expected to improve thermal efficiency. Exhaust control, on the other hand, increases the compression ratio relative to the expansion ratio, permits earlier closing of the cylinder (for a given scavenging time area), and provides an effective supercharge. Are these the significant factors in the output comparison of Fig. 39?

5. How important is the rate of closing when a rotary exhaust valve is used?

6. Does Dr. List recommend the pinning of compression rings?

7. Should round port upper edges be used only from a production consideration or does the larger allowable unsupported ring arc justify the adverse effect on timing?

8. What maximum unsupported ring arcs are considered advisable in the combinations of pinned and unpinned rings with round and square port upper edges?

Discusses Work on Scavenging Efficiency, Piston Construction

—G. Flynn, Jr.
General Motors Corp.

I WAS particularly impressed with the success of the mathematical analysis and with the excellent correlation of calculated results and experimental data, as evidenced by Figs. 8 and 11 in the paper. Having recently been associated with a somewhat similar analysis of a two-stroke engine aided by a very modern and expensive IBM calculator, I can fully appreciate the magnitude and value of the calculations Dr. List presents.

We have done considerable work in our laboratory using models to evaluate scavenging efficiency of various port and cylinder configurations. We used all sorts of air, smoke, and water analogies and finally resorted to the quantitative method described by W. H. Percival in his paper presented at the 1954 Diesel Engine Meeting. Here, however, we run into the problem of defining scavenging efficiency. Does Dr. List use the volume basis for determining scavenging efficiency as defined by Schweitzer, namely, the volume of fresh air trapped, divided by the total volume of trapped gas; or does he use the weight method advocated by Taylor and Taylor—that is, weight of fresh air trapped divided by the total cylinder volume multiplied by a density factor based on inlet air temperature and exhaust pressure. This latter method, we feel, is more realistic since power output depends on the weight of oxygen burned and not the volume. For instance, in two identical cylinders with different port heights the scavenging efficiency based on volume might show the cylinder with the high exhaust ports to be superior. However, on a weight basis the lower port cylinder would give a higher scavenging efficiency, better fuel/air ratio, more optimum combustion, and thus, more power.

One other point I would like to mention is in regard to preformed pistons. A piston can be a complex structure subjected to unknown loads and thermal stresses. Therefore, it is most difficult to predict what shape the piston wants to assume under temperature, gas, inertia, and friction loading. Our experience has shown that it is better in high output engines to take wristpin loads

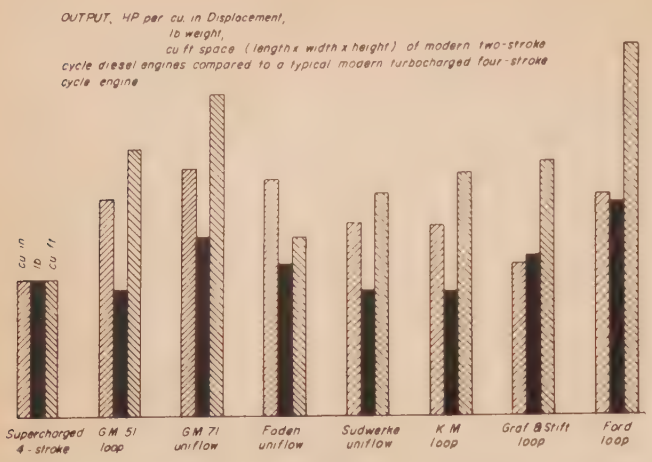


Fig. C—Bar diagram for comparing performance of modern engines

directly under the piston head and make the piston skirt flimsy enough to conform to the cylinder, rather than to tell the piston by cam grinding what shape to take. As Boss Kettering used to say, "the piston is usually smarter than the piston engineer."

This type of construction is used on many engines and tends to minimize the tendency to scuff at the port area. However, it has been our experience that most port bridge scuffing is due to thermal distortion of the cylinder, rather than piston thrust forces. We have had this problem even in free-piston engines, which have no piston thrust forces and have easily corrected it by relieving or "Mae Westing," as we call it, the port belt in the cylinders.

I realize that Dr. List has presented a great deal of material, but there was no mention of combustion problems in the paper. In concluding, I would like to ask if, after having satisfactorily worked out scavenging, was combustion control much of a problem? Also, what is Dr. List's reaction to the claim of MAN that combustion should be controlled by spraying the fuel directly on the piston heads?

Further Discusses Aspects of 2-Stroke Engine

—P. H. Schweitzer
Pennsylvania State University

IN the middle of the Thirties I was engaged in 2-stroke-cycle engine development and needed information on the scavenging efficiency of such engines. Where did I find it? In a research publication of a technical institute in Woosung, China. The author of those reports was a Dr. Hans List who conducted an advanced research on 2-stroke-cycle engines in China.

Dr. List was a pioneer then as he is a pioneer now. After the war he has developed no fewer than eight loop-scavenged, 2-stroke-cycle engines, and all of them are in production now in four different countries.

To you it is not new how partial I am to the loop-scavenged engine. This preference is mainly based on the simplicity of port scavenging. When there are no valves, rockers, pushrods, tappets, and camshafts, these parts cannot go wrong, and they don't have to be made in the first place. These parts do not add to the engine displacement, but nevertheless they have bulk and weight. Therefore, the valveless port-scavenged engine may have smaller specific weight and installed volume than the uniflow engine in spite of its 10–15-lb lower bmep, as shown in Fig. C.

The first bar is always the horsepower per cubic inch, the second the horsepower per pound weight and the third the horsepower per installed volume. The higher the bar the better. they all relate to a modern turbocharged 4-stroke-cycle engine. The second bar represents uniflow engines, and the next bars, loop-scavenged engines. You see the latter bars are generally higher than the former, and both are higher than the first. The highest bars are those on the right end which represents the Ford-Amok engine to which Dr. List referred.

Many people visualize the uniflow engine as having perfect scavenging. The fresh air just pushes out the residual gases as if it were a gas piston. This is not so. A uniflow engine generally has angular intake ports for swirl. Then the scavenge airflow has a center core where the flow is in the reverse (Fig. D). This is the nature of the semifree vortex encountered in tornados. So the scavenge air does not act as a gas piston in a uniflow engine.

On the other hand, a well designed loop-scavenged engine with advanced Schnürle type porting has scavenge airflow similar to the one shown in Fig. E. This is the result of tests we have made at Penn State. Little short-circuiting was observed and most of the scavenge air went up and turned down later.

A little-used stunt to reduce the fuel consumption of a 2-stroke-cycle engine is recirculation of scavenge air as explained by Dr. List. The 17% saving in a city bus by such simple means is really worth going after.

The part I have found most interesting in the paper is List's prediction on the turbocharged 2-stroke engine. You notice from his Fig. 42 that when the pressure ratio is doubled with 60% combined turbocharger efficiency the bmep is more than doubled, and starting out with 64 bmep you can get 96 bmep with only 1.4 pressure ratio. We have an engine under development which will weigh a little under 3 lb per hp with 50% supercharge so it will be lighter and occupy less space than present-day automobile engines.

The future of the loop-scavenged diesel engine seems very bright indeed. We are indebted to Dr. List for his pioneering work and for this enlightening paper.

Questions Certain Aspects Of Design of This Engine

—A. R. Schrader
USN Engineering Experiment Station

DR. LIST is to be congratulated for his excellent and comprehensive paper on European practice as regards high-speed loop-scavenged diesel engines. Dr. List's contributions to the design of these engines are obviously considerable. The high-speed loop-scavenged engine has been largely neglected in this country. The basic simplicity of the engine would seem to warrant greater attention by American manufacturers.

Dr. List's use of mathematical analysis in studying various design factors and in forecasting engine performance is of great interest. We have made use of this technique, although not in such detail, in studying the application of various methods of turbocharging to a 2-stroke-cycle diesel engine. We found reasonably good correlation between the calculated results and subsequent actual test operation. However, in calculations of this nature, the final accuracy is dependent to a large degree on the validity of the necessary assumptions and the correctness of the constants used. A certain amount of empirical data from the engine in question or from similar engines is required for best results.

Regrettably, only a passing reference is made in the paper to a difficulty that has often plagued the designers and operators of engines with ported cylinders. I refer to the fouling of the ports with carbon deposits. Has any significant difficulty in this regard been experienced in the loop-scavenged engines cited in this paper? What would the author consider to be an acceptable or normal period of operation between the required cleaning of ports?

Fig. 11 in the paper shows the optimum heights of the exhaust and inlet ports in terms of per cent of stroke. However, the basis of selection is not clear. Are the optimum heights based on power output, fuel economy, airflow characteristics, or other considerations?

In referring to Fig. 43, the author concludes that, from the standpoint of mechanical loads, there is no definite limit to the degree of supercharge. This seems questionable. Whether the plotted curves continue to drop or whether they flatten out, the fact remains that the maximum cylinder pressure increases at a rate five to eight times faster than the brake mean effective pressure. Therefore, extremely high cylinder pressures can result if the bmep continues to be increased. While the point at which the pressure must be limited is debatable, there obviously are limits.

The fuel economy data shown in the paper for the several

European engines are excellent. The fuel consumption of comparable American engines is not usually that good, at least as judged from my own experience. Can the author suggest any reasons for this apparent fuel economy superiority of the European engines, that is, are there differences in test procedure, engine equipment, and cylinder pressure?

The turbocharging possibilities of the high-speed loop-scavenged diesel engines, as analyzed in this paper, appear to offer considerable promise. The simplicity of the engine, the prospects of high unit output, and the relative compactness seem attractive. However, the examples of turbocharged loop-scavenged engines cited in the paper are of only moderate output, 100- to 115-psi bmep. Does the author know of any European developments of really high output turbocharged loop-scavenged engines, of perhaps 150- to 160-psi bmep?

Note: The opinions or assertions made above are those of the author and are not to be construed as official or reflecting the views of the Navy Department or the naval service at large.

ORAL DISCUSSION

Reported by G. Flynn, Jr.

General Motors Corp.

Mr. Burrows, White Motor Co.: The successful operation of the White loop-scavenged engine pleases us, particularly as regards lubricating oil consumption. Unlike most of the engines mentioned in the paper the White engine utilized a ductile cast-iron piston for high strength and ring grove protection. The decreased peak bearing loads are due to inertia unloading of the iron pistons, and these higher strength materials would be required when turbocharging and higher engine outputs were achieved. A 40% increase in output of the White engine has been reached by turbocharging with satisfactory exhaust temperatures.

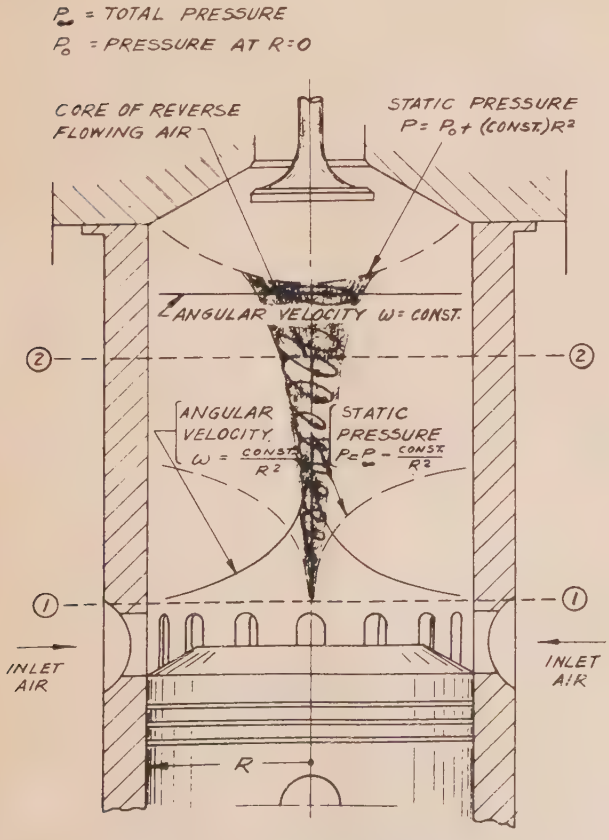


Fig. D—Schematic diagram of reverse flow theory

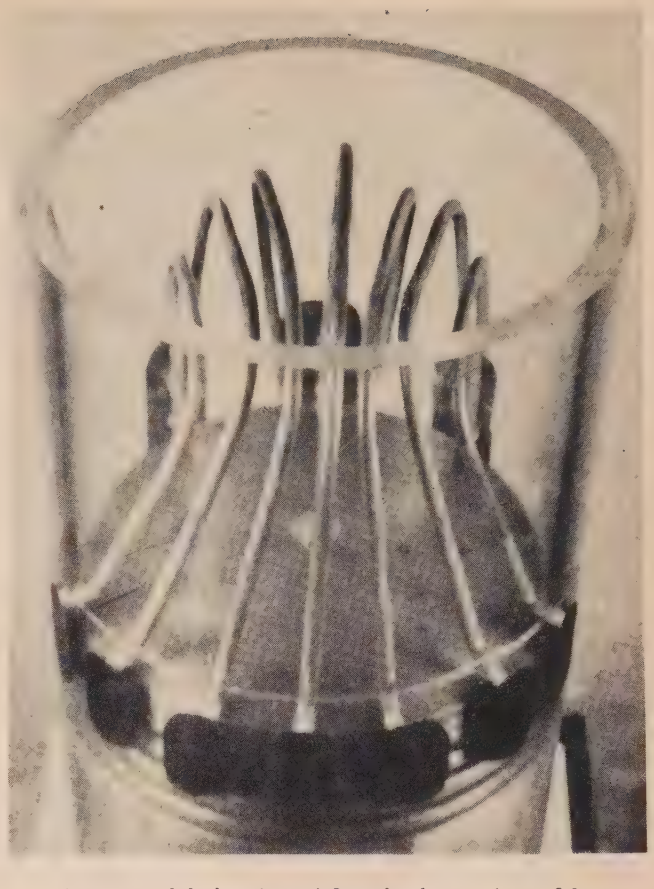


Fig. E—Model showing airflow by bent wires of loop-scavenged engine

C. F. Taylor, MIT: Other factors besides porting have a great effect on the relationship of air retained to air supplied in 2-cycle engines. Engine speed, load, operating condition, and the like were mentioned.

J. E. Witzky, Studebaker-Packard Corp.: Why did one engine mentioned in the paper have a turbulent combustion chamber while all the rest were direct injection?

Dr. List: The Graf and Stift engine chamber has been modified and is now a direct injection type. The turbulent chamber was originally used with the knowledge of its higher heat transfer to provide more quiet engine operation.

Author's Closure To Discussion

THE port heights which were shown in Fig. 11 had been calculated on the assumption that the engine was to deliver the highest obtainable bmep. It is possible also, from a similar relationship, to calculate port heights for minimum specific fuel consumption. In the latter case one arrives at greater port height and, consequently, lower scavenging pressures.

Fig. F shows the relationship which exists between blow-back, output, and preexhaust (difference between inlet and exhaust port heights). These curves were calculated for one specific case, and they show that bmep increases with decreasing port height difference up to a certain point. If the port height difference is further reduced bmep begins to decrease also. In general it is advisable to choose the preexhaust somewhat greater than that which corresponds to maximum bmep.

Due to the unavoidable short-circuiting of some of the scavenging air, the exhaust gas is mixed with air so that the exhaust temperatures of loop-scavenged, 2-cycle engines are substantially lower than those of 4-cycle machines. The thermal load imposed on the turbocharger, therefore, is considerably lower than in the case of 4-cycle engines so that, in this type of engine, the turbocharger does not limit the obtainable output. The degree of

supercharge, and therefore, the output obtainable from the engine is defined by the thermal load of the surfaces forming the combustion chamber. As the shape of the cylinder head is very simple, the thermal load to be carried by the piston, therefore, usually establishes the output limitation.

The piston temperatures were measured on uncooled 4-cycle and 2-cycle pistons. It can be concluded, therefore, that the small difference in piston temperature is largely due to piston cooling provided by the scavenging air.

The temperature in the center of the piston crown provides a measure for the thermal condition of the whole piston. To evaluate completely the heat condition of the piston it is, however, also necessary to measure the temperatures in the ring belt. The points of a loop-scavenged engine which are subject to critical thermal loading are shown in Fig. G.

In small high-speed automotive engines we usually use divided combustion chambers because they are less sensitive to fuel characteristics than open-chamber and because they permit keeping combustion noise within acceptable limits. Furthermore, single-hole or pintle nozzles can be used with this system, thus avoiding extremely small nozzle holes. These considerations made us decide on the use of a turbulence chamber in most of the small engines, including the Ford engine. When using this type of combustion system it is usually advisable to place a high percentage of the total volume in the turbulence chamber. The chamber of the Ford engine contains 71% of the total volume.

Pischinger, in his experiments to establish the heat load on the nozzle, measured the nozzle temperature with a thermocouple. By heating the nozzle electrically and simultaneously injecting fuel at the same temperatures as previously measured in operating engines, the heat absorbed by the nozzle, was measured. The squish velocities (v_{max}) which were shown in Fig. 24 of the paper, were determined mathematically; ϵ is the compression ratio of the engine. The squish velocity is a function of the chamber entrance diameter, the clearance between piston and cylinder head, the compression ratio, and the mean piston speed. It was shown that, within the limits investigated, the heat transfer to the nozzle depended solely on the squish velocity.

Port deposits largely depend on the lubricating oil consumption of the engine, the quality of the lubricating oil, the quality of combustion, and the temperature of the port walls. We used to consider it satisfactory if cleaning of parts became necessary after approximately 40,000 miles of road service. Present experience, however, shows that 80,000-mile periods are normally obtained.

Separate cylinder lubricating devices can, of course, not be used in small engines as they would not be compatible with the required simplicity of design. Thus, it is necessary to transfer a sufficient but not excessive amount of lubricating oil to the upper part of the cylinder. The oil is transferred along the port bridges and then along the cylinder surface up to the piston rings. The tenacity with which a thin oil film adheres to a surface prevents the port edges from stripping the oil from the piston. For proper lubrication and small wear, it is necessary not only to transfer a sufficient quantity of oil as far up as the top piston ring but also to distribute it evenly over the whole circumference.

The fuel consumption data, shown in the paper, were measured on engines equipped with all accessories with the sole exception of the cooling fan. The satisfactory specific fuel consumption obtained with the direct-injection engines is largely due to the shape of the combustion chamber which, due to the absence of valves, can be designed specifically for efficient combustion.

Development work on truly high-output, 2-cycle engines has only just begun in Europe. Present production loop-scavenged, 2-cycle engines with supercharging are generally rated at approximately 100-psi bmep. Some Deutz and British Polar engines reach 125 psi maximum. The information concerning the Polar engine was taken from the October, 1955, issue of *The Motorship*. This engine shows an excellently designed scavenging system. It works with a low product of bore times rpm so that the conditions for scavenging and thermal loading are very good. The high-speed Napier Nomad has reached 205 psi. I am convinced that, within the relatively near future, commercial engines will reach 140 to 160 bmep and will operate reliably at this rating. A great deal of development work, particularly on cooling and on parts which are highly loaded thermally, remains to be done.

The advantage of supercharging consists in the fact that maximum pressure does not increase in direct proportion to the general pressure level of the working cycle. The higher pressures and temperatures which exist in a supercharged engine tend to reduce ignition delay so that the combustion process becomes "softer." For the same reason, the highly supercharged engine is less sensitive to fuel characteristics than an engine in which combuston takes place under conditions which cause greater ignition delay. It has been mentioned before that the favorable $p_{z\,max}$/bmep ratio results in the highly supercharged engine having a somewhat lower weight per horsepower than the naturally-aspirated engine. This statement holds good for engines designed for equal mechanical stresses. It is a matter of course that the physical size of the crank drive has to be increased. Mechanical stresses in the crank drive do not limit the output obtainable from an engine, but it is to be expected that the mechanical stresses imposed upon the piston rings and the cylinder head gaskets will cause some difficulties.

I would like to repeat that, assuming suitable mechanical design, the output limitation is defined by thermal stresses occurring in parts which are in contact with the hot gases. This refers particularly to the pistons. In the development of high-output engines

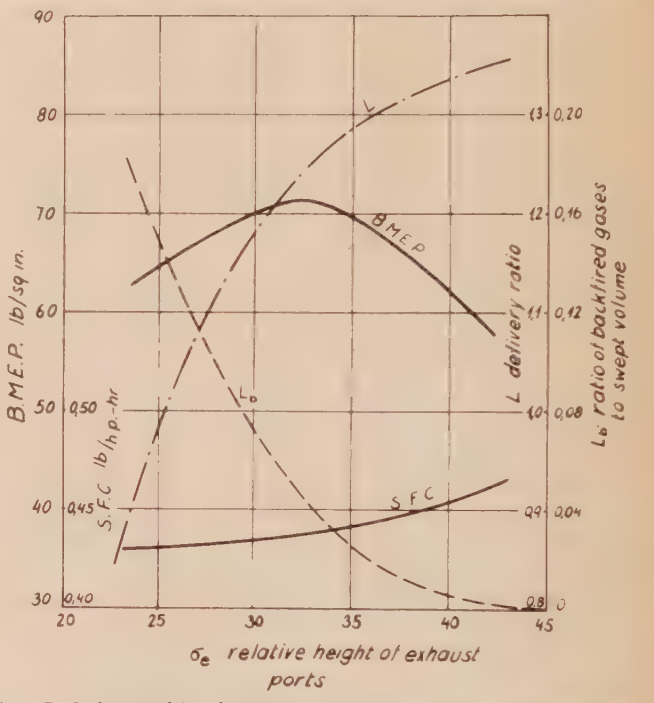


Fig. F—Relationship between blow-back, output, and pre-exhaust

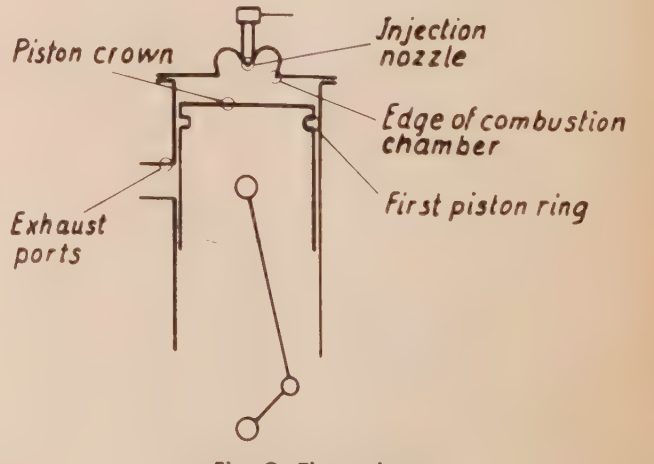


Fig. G—Thermal stress

of this type the main effort therefore, has to be directed at controlling thermal stresses.

With reference to some specific questions asked by discussers I would like to say the following:

The purity of the charge depends on the ratio between the volume to be scavenged to the volume of the entering air. To calculate the purity it is therefore necesary to use the mean cylinder volume during scavenging as the basis.

Generally speaking, very little is left, at the end of the compression stroke, of the turbulence produced by a symmetrical Schnürle arrangement. In some cases a slightly asymmetrical arrangement of the inlet ports proves effective. This design creates a rotation around the cylinder axis which improves mixing and combustion. Unfortunately, this arrangement also reduces the scavenging efficiency and, thus can only be employed to a very limited degree.

We have done a great deal of work in connection with tuned exhaust systems. It appears to us that a worthwhile result can only be obtained with an exhaust system which is hardly applicable. Thus, we have reached the conclusion that an increase in scavenge volume, compared to an open exhaust, cannot be obtained in practice. In fact, the best that can be hoped for is that the production exhaust system will not reduce scavenging efficiency.

Fig. 37 of the paper showed that, in accordance with our test results, the rotary valve in the exhaust is better than that in the intake so far as bmep is concerned. We were unable to ascertain an improvement due to a rotary valve in the intake.

It is most important that the rotary valve, with a given closing point, produce a maximum scavenging-time area. This necessitates the most rapid closing possible. Numerical data can, of course, only be supplied for certain stated conditions.

We have had unsatisfactory experience with pinned piston rings as they are prone to sticking. Rounded port edges appear to be absolutely necessary for satisfactory piston ring performance.

We have successfully used port width equal to 0.2 of the bore.

Fig. 1 of the paper showed the 2-cycle engines actually are quite favorable in weight per horsepower and that they are actually better than 4-cycle in this respect. It is to be said, with reference to Mr. Miller's discussion that high-speed, 2-cycle engines, even those with symetrical timing diagrams, work with a slight degree of supercharge as can be seen from Fig. 4; that high-speed, loop-scavenged, 2-cycle engines can operate with the same piston speeds as those used in 4-cycle engines; that exact tests have proven that the excess air requirement (with reference to the intake air volume) can be somewhat smaller when exhaust gas is mixed with the charge. We have done a great deal of work on the effect of exhaust gas content in the cylinder, experiments which were conducted on 4-cycle engines and have established that the smoke limit is a function only of the total air content of the charge.

I might add the following with respect to the process of engine research:

The overall performance depends on output and specific fuel consumption. To determine which are the most promising fields for improvement, a functional cycle analysis is performed. The complete cycle is divided into a sequence of events, and input and result is determined for each event. It is best to start with an assumed, conventional and "perfect" engine and to investigate the relative losses which would result from each deviation from the ideal cycle. We call this our "loss analysis" and have used it effectively for the evalution of existing engines as well as for the development of new designs. The insight into the events occurring in an engine which is provided by this method of analysis, permits us to predict what can be expected from anticipated modifications and, thus, points up the direction in which development work should proceed. Figs. H and I show the result of a "loss analysis" for a 4-cycle diesel engine. When performing an analysis of this type, the various reference data must be established in a manner which clearly shows the input into and result produced by each event.

For this reason it seems best, in studying charge change events, to use volumetric parameters rather than those based on weight The values can be rendered dimensionless by referring them to cylinder displacement. The relationship to the volume of air necessary to burn unit quantity of fuel can easily be established by the introduction of a certain reference condition, usually the ambient. Three parameters are used to define the cylinder c... the total charge contained in the cylinder, the "purity" of ... charge and the content of fresh air. The last is the product of the first two.

The investigation of turbocharged engines requires that the energy levels utilized by each of the two components, the piston engine and the turbocharger, be analyzed with reference to the "ideal" cycle. Thus, the energy utilization in each component can be established, providing the basis for establishing the efficiency for each individual event. Space limitations prevent further discussion here of the proper selection of reference parameters.

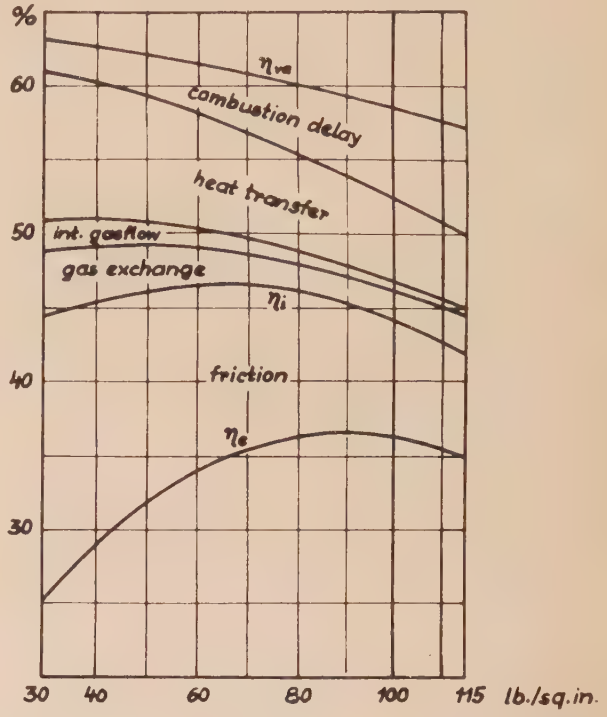


Fig. I—Result of "loss analysis" for 4-cycle diesel engine

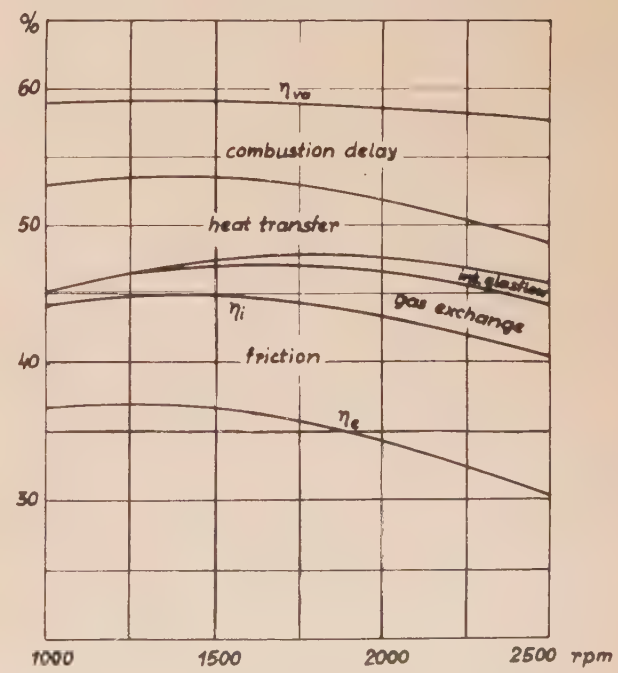


Fig. H—Result of "loss analysis" for 4-cycle diesel engine

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